Guide

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HEATING VENTILATING
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GUIDE 1959

Applied Science Dept.

HEATING VENTILATING AIR CONDITIONING GUIDE 1959

An Instrument of Service
Prepared for the Profession
containing

A TECHNICAL DATA SECTION OF REFERENCE MATERIAL ON THE DESIGN AND SPECIFICATION OF HEATING, VENTILATING, AND AIR-CONDITIONING SYSTEMS BASED ON—THE TRANSACTIONS—THE INVESTIGATIONS OF THE RESEARCH LABORATORY AND COOPERATING INSTITUTIONS—AND THE PRACTICE OF THE MEMBERS AND FRIENDS OF THE SOCIETY; A MANUFACTURERS' CATALOG DATA SECTION CONTAINING ESSENTIAL AND RELIABLE INFORMATION CONCERNING MODERN EQUIPMENT; COMPLETE INDEXES TO TECHNICAL AND CATALOG DATA SECTIONS.

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PREFACE TO THE 37th EDITION

The HEATING VENTILATING AIR CONDITIONING GUIDE, 1959, has been enlarged to 8½ x 11 inch page size for the purpose of providing additional space for new technical text and for enlargement of many illustrations, working charts, and tables. The use of larger type and open spacing in tables will be of particular advantage, for instance, in selecting heat transfer coefficients for various types of building construction or finding degree days for various cities.

This edition has 286 new or revised diagrams and charts. In most cases the enlarged page size has made it possible to place illustrations very close to their first mention in the text.

Five new chapters have been added on the following subjects: (1) high-temperature water systems, (2) the heat pump, (3) evaporative apparatus for heat rejection, (4) evaporative air cooling and humidification, and (5) snow melting.

Outstanding improvements made in chapters continued from the previous edition include (1) new outdoor winter design temperatures based on probability of occurrence of various temperatures, (2) new steam pipe sizing charts and tables based on Moody friction data for steam flow, (3) a revised table on gages and construction of air ducts including low-, medium-, and high-pressure ducts, (4) revised information and illustrations for unit ventilators and their application, (5) a rewritten chapter on automatic control with new illustrations of control applications, (6) general revision of the information on sorbents, including their use at elevated pressures, (7) data on heat transfer coefficients for ventilated attics, (8) revised information on sound absorption in ducts and plenums, noise levels, and diffuser noise, (9) up-to-date values for maximum allowable concentration of contaminants in occupied spaces, (10) a new section on the absorption refrigeration system, (11) a new section on the gas-fired year-round residential air conditioner, (12) new data on duct construction for residences, (13) a revised discussion of fuel utilization, efficiency, and consumption, (14) a rewritten chapter on school systems, and (15) additions and changes in the list of codes and standards.

More detailed information regarding the changes in the various chapters:

Chapter 7—Air Contaminants. Additions and revisions have been made in the tables of maximum allowable concentration of toxic gases, vapors, dusts, fumes, and mists, in order to have these values conform to those adopted by the American Conference of Governmental Industrial Hygienists in 1958.

Chapter 9—Heat Transmission Coefficients of Building Materials. A section on ventilated attic coefficients, with examples, has been added. The section on the practical use of overall coefficients has been revised. The tables of heat transmission coefficients for various building constructions have been enlarged to facilitate their use. All wall and roof constructions have been numbered to aid in identifying and referring to the coefficient for a particular construction.

Chapter 10—Moisture in Building Construction. The information on moisture migration and its calculation has been revised. The discussion of condensation in cooled structures has been extended.

Chapter 12—Heating Load. The data on winter outdoor temperatures have been revised to show as design values for the United States the minimum average daily temperatures that have the probability of occurring once in 40, 20, 13, 10, or 5 years as determined from U. S. Weather Bureau records. The discussion of design temperatures has been rewritten to explain the use of these new values. The tables have been enlarged in type and size for easier readability.

Chapter 15—Unit Ventilators and Unit Heaters. The discussion of unit ventilators has been enlarged and thirteen new illustrations have been added. The illustrations of piping connections to unit heaters have also been revised.

Chapter 16—Unitary Air-Conditioning Equipment. A section discussing the features, design considerations, selection, and installation of room air conditioners has been added.

Chapter 17—Electric Heating. This chapter was rewritten to include information on present heating units and methods of use.

Chapter 18—Warm Air Heating Systems. The information on ducts has been revised to include recommended gages for residential systems.

Chapter 20—Air Distribution. The sections on principles of air distribution and on ventilating jets have been rewritten and simplified.

Chapter 21—Air Duct Design. A revised section on construction includes data on high-, medium-, and low-pressure ductwork, coordinated with the recommendations of the Sheet Metal and Air Conditioning Contractors National Association. The charts for friction loss in ducts and the table of equivalent sizes of round and rectangular ducts have been enlarged.

Chapter 25—Sound Control. Data on noise levels in offices have been added. The section on noise from axial-flow fans has been enlarged. New information is given on diffuser noise. The sections on sound attenuation in ducts and plenum absorption have been revised.

Chapter 26—Steam Heating Systems. The information on steam pipe sizing has been rewritten to include a new basic chart (0 to 200 psig) based on the Moody friction data which take into account the Reynolds number and its effect on the friction factor, and to include also new charts for 30, 50, 100, and 150 psig and tables for 3.5 and 12 psig. (The new information, charts, and tables were prepared by the ASHAE Research Laboratory under the direction of the Technical Advisory Committee on Hot Water and Steam Heating.) The section on pressure-reducing valves has been enlarged by the addition of information formerly given in the chapter on District Heating.

Chapter 28—Hot Water Heating Systems. A new chart for available gravity head has been added.

Chapter 29—High-Temperature Water Systems. A new chapter has been added giving a general idea of the principles and practices that apply to high-temperature systems and discussing significant considerations for designing these systems. Many new illustrations showing piping arrangements were used.

Chapter 34—Automatic Fuel-burning Equipment. There has been a general revision of this chapter to bring the information up-to-date. A new chart for calculating friction drop or capacity for gas flow in piping has been added.

Chapter 35—Heating Boilers, Furnaces, and Space Heaters. The data on steel boilers have been improved and enlarged.

Chapter 37—Estimating Fuel Consumption for Space Heating. The former section on seasonal efficiency has been replaced by a new section on Efficiency of Utilization and the section on Calculated Heat Loss Method has been rewritten.

Chapter 38—Refrigeration. The information on absorption systems has been revised and new illustrations have been added.

Chapter 39—The Heat Pump. This is a new chapter giving information on the fundamentals, types, components, performance, design, and selection of heat pumps.

Chapter 40—Evaporative Apparatus for Heat Rejection. This is a new chapter giving information on water-cooling methods and equipment. Cooling tower theory, design, selection, operation, and maintenance are discussed. New data on spray performance are included.

Chapter 41—Evaporative Air Cooling and Humidification. This is a new presentation of the operation of air washers and their use for humidification and dehumidification. New information is given on evaporative cooling, types of equipment, and design methods. Several new illustrations were used.

Chapter 42—Dehumidification by Sorbent Materials. This chapter has been rewritten to include current information on solid and liquid sorption systems. A new section has been added on use of sorbents at elevated pressures. New illustrations were included.

Chapter 43—Automatic Control. This chapter was rewritten and enlarged to include 18 new cuts showing application of controls for various operations. The description of control types and operation has been extended.

Chapter 44—Instruments and Measurements. This chapter has been brought up to date by extensive revision.

Chapter 45—Motors and Motor Controls. The section on classification of motors has been revised to indicate the nomenclature adopted by the National Electric Manufacturers Association.

Chapter 46—Residential Summer Air Conditioning. A new section on gas-fired year-round air conditioners has been added.

Chapter 47—School Systems. This chapter has been rewritten with emphasis on factors affecting the heating and cooling loads. Various types of heating, cooling, and ventilating systems are discussed.

Chapter 48—Transportation Air Conditioning. The section on control of passenger bus air conditioning has been revised.

Chapter 49—Snow Melting. A new chapter has been added giving a method of designing snow-melting systems. New tables and data are included.

Chapter 55—Corrosion and Water-formed Deposits, Causes and Prevention. The section on use of resistant metals has been improved.

Chapter 56—Water Services. The charts for friction loss have been enlarged. The section on electric water heating has been transferred to this chapter from the chapter on electric heating.

Chapter 57—Codes and Standards. Several new standards have been added and the correct editions of listed standards have been indicated.

In addition to the changes in specific chapters outlined, numerous improvements have been made in both text and illustrations.

A complete index to Technical Data is contained in a 16-page section at the front of THE GUIDE.

For convenient reference to equipment to be selected for various types of installations, this book contains a Catalog Data Section showing products of 309 manufacturers of heating, ventilating, cooling, and air-conditioning equipment. This section is preceded by a 16-page index listing equipment available under numerous headings for most convenient use.

The Guide Committee is pleased to give credit for Preparation of Guide material for this edition to the ASHAE Technical Advisory Committees, particularly those on Evaporative Cooling, Heat Pump, Heating and Air-Conditioning Loads, Hot Water and Steam Heating, Insulation, and Sorption. The staff of the ASHAE Research Laboratory has rendered valuable assistance to these committees. Many trade associations, engineering societies, and other organizations mentioned in the text, in addition to individual engineers, both Society members and nonmembers, have contributed new material or reviewed present material to establish its authenticity. While it is impossible to recognize all sources of information and help, the Guide Committee is pleased to mention the following, who have been very active in preparation of this volume:

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The Guide Committee takes pleasure in presenting to the Society this 1959 volume in its new format, which should make the book more valuable than any preceding edition to its users.

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CHAPTER 1

TERMINOLOGY

Glossary of Physical and Heating, Ventilating, Refrigerating and Air Conditioning Terms Used in the Text

Absolute Zero: The zero from which absolute temperature is reckoned. Approximately −273.15 C or −459.67 F. These values were established by the Tenth General Conference on Weights and Measures (1954).

Absorbent: A sorbent which changes physically or chemically, or both, during the sorption process.

Absorption: The action of a material in extracting one or more substances present in an atmosphere or mixture of gases or liquids; accompanied by physical change, chemical change, or both, of the sorbent.

Acceleration: The time rate of change of velocity, i.e., the derivative of velocity with respect to time. In the cgs system the unit of acceleration is the centimeter per (second) (second); in the fps system the unit is the foot per (second) (second),

$$a = \frac{dv}{dt}$$

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body the value of which varies with latitude and elevation. The international gravity standard has the value of 980.665 cm per (sec) (sec) or 32.174 ft per (sec) (sec) which is the actual value of this acceleration at sea level and about 45 deg latitude.

Adiabatic: An adjective descriptive of a process such that no heat is added to, or taken from, a substance or system undergoing the process.

Adsorbent: A sorbent which does not change physically or chemically during the sorption process.

Adsorption: The action, associated with surface adherence, of a material in extracting one or more substances present in an atmosphere or mixture of gases and liquids, unaccompanied by physical or chemical change. Commercial adsorbent materials have enormous internal surfaces.

Aerosol: An assemblage of small particles, solid or liquid, suspended in air. The diameters of the particles may vary from 100 microns down to 0.01 micron or less, e.g., dust, fog, smoke.

Air Cleaner: A device designed for the purpose of removing airborne impurities such as dusts, gases, vapors, fumes, and smokes. (Air cleaners include air washers, air filters, electrostatic precipitators, and charcoal filters.)

Air Conditioning: The process of treating air so as to control simultaneously its temperature, humidity, cleanliness, and distribution to meet the requirements of the conditioned space.

Air, Dry: In psychrometry, air unmixed with, or containing no, water vapor.

Air, Saturated: A mixture of dry air and saturated water vapor, all at the same dry-bulb temperature.

Air, Standard: Air with a density of 0.075 lb per cu ft and an absolute viscosity of 0.0379×10^{-4} lb mass per (ft) (sec). This is substantially equivalent to dry air at 70 F and 29.92 in. Hg barometric pressure.

Air Washer: An enclosure in which air is drawn or forced through a spray of water in order to cleanse, humidify, or dehumidify the air.

Anemometer: An instrument for measuring the velocity of a fluid.

Aspect Ratio: In air distribution outlets, the ratio of the length of the core of a grille, face, or register to the width.
In rectangular ducts, the ratio of the width to the depth.

Atmospheric Pressure: The pressure of the atmosphere to be measured by means of the barometer. *Standard Atmospheric Pressure* or *Standard Atmosphere* is exactly 1,013,250 dynes per sq cm by recent action of the Tenth General Conference on Weights and Measures (1954). It is approximately the pressure exerted by a column of mercury 76 cm high at standard gravitational acceleration, 980.665 cm per (sec) (sec), the mercury having a density of 13.5951 g per cu cm. It is equivalent to 14.696 psi or 29.921 in. Hg at 32 F.

Baffle: A surface used for deflecting fluids, usually in the form of a plate or wall.

Blast Heater: A set of heat transfer coils or sections used to heat air which is drawn or forced through it by a fan.

Blow (throw): In air distribution, the distance an air stream travels from an outlet to a position at which air motion along the axis reduces to a velocity of 50 fpm.
For unit heaters, the distance an air stream travels from a heater without a perceptible rise due to temperature difference and loss of velocity.

Boiler Heating Surface: All of the surface of the heat-transfer apparatus in contact on one side with the water or wet steam being heated and on the other side with the gas or refractory being cooled, in which the fluid being heated forms part of the circulating system, this surface to be measured on the side receiving heat. (Stationary Steam-Generating Units, *ASME Power Test Codes*, PTC 4-1946.)
Direct Heating Surface is generally understood to be the *boiler heating surface* subject to direct radiation from the surface of the grate, or from the surfaces of oil or gas burners.
Indirect Heating Surface is the *boiler heating surface* within the flues.

Boiler Horsepower: The equivalent evaporation of 34.5 lb of water per hr from and at 212 F. This is equal to a heat output of 970.3 × 34.5 = 33,475 Btuh.

British Thermal Unit (Btu): The Btu is defined as 778.177 foot-pounds if it is related to the IT calorie in such a way that 1 IT calorie per (kg) (C deg) = 1 Btu per (lb) (F deg), with 1 lb = 453.5924 g. The Ninth General Conference on Weights and Measures (1948) recommended the absolute joule as the fundamental unit of energy thereby, in effect, liquidating the old international electrical units. This calls for a redefinition of the International Tables calorie (IT cal) according to the following conversion factor: 4.18684 joule per IT calorie.

Calorie (Gram Calorie): The IT gram calorie is equivalent to 0.003968 Btu. (See definition of Btu.)

Central Fan System: A mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and is conveyed to and from the rooms by means of a fan and a system of distributing ducts. (See Chapter 19.)

Chimney Effect: The tendency of air or gas in a duct or other vertical passage to rise when heated due to its lower density compared with that of the surrounding air or gas. In buildings, the tendency toward displacement (caused by the difference in temperature) of internal heated air by unheated outside air due to the difference in density of outside and inside air.

Coefficient of Performance (Heating): The ratio of the total *instantaneous* useful heating effect produced by a heat pump system at stated conditions, to the heat equivalent of the total energy input rate required to drive or operate the system.

Coefficient of Performance (Cooling): The ratio of the *instantaneous* refrigeration effect produced by a heat pump

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system at stated conditions, to the heat equivalent of the total energy input rate required to drive or operate the system.

Comfort Air Conditioning: The process of treating air so as to control simultaneously its temperature, humidity, cleanliness, and distribution to meet the comfort requirements of the occupants of the conditioned space. (See also general definition of *Air Conditioning* on page 1 of this chapter.)

Comfort Line: A line on the comfort chart showing relation between the effective temperature and the percentage of adults feeling comfortable.

Condensate: The liquid formed by condensation of a vapor. In steam heating, water condensed from steam; in air conditioning, water extracted from air, as by condensation on the cooling coil of a refrigeration machine.

Condensation: The process of changing a vapor into liquid by the extraction of heat. Condensation of steam or water vapor is effected in either steam condensers or in dehumidifying coils and the resulting water is called *condensate*.

Conductance, Surface (*Unit*): The amount of heat transferred by radiation, conduction, and convection from unit area of a surface to the air or other fluid in contact with it, or vice versa, in unit time for a unit difference in temperature between the surface and the fluid. The common unit is: Btu per (hour) (square foot) (Fahrenheit degree). Symbol f. The temperature of the fluid should be taken in a plane sufficiently far from the surface that it will not be affected by the temperature of the surface.

Conductance, Thermal: The time rate of heat flow through unit area of a body, of given size and shape, per unit temperature difference. Common unit is: Btu per (hour) (square foot) (Fahrenheit degree). Symbol C.

Conduction, Thermal: The process of heat transfer through a material medium in which kinetic energy is transmitted by the particles of the material from particle to particle without gross displacement of the particles.

Conductivity, Thermal: The time rate of heat flow through unit area of a homogeneous substance under the influence of a unit temperature gradient. Common units are: Btu per (hour) (square foot) (Fahrenheit degree per inch). Symbol k.

Conductor, Thermal: A material which readily transmits heat by means of conduction.

Convection: The motion resulting in a fluid from the differences in density and the action of gravity. In heat transmission this meaning has been extended to include both *forced* and *natural* motion or circulation.

Convective Heat Transfer: The transmission of heat by either natural or forced motion of a fluid (liquid or gas).

Convector: An agency of convection. In heat transfer, a surface designed to transfer its heat to a surrounding fluid largely or wholly by convection. The heated fluid may be removed mechanically or by gravity (Gravity Convector). Such a surface may or may not be enclosed or concealed. When concealed and enclosed the resulting device is sometimes referred to as a concealed radiator. (See also definition of *Radiator*.) (See also Chapter 14.)

Decibel: A unit used to express the relation between two amounts of power. By definition the difference in decibels between two powers P_1 and P_2, P_2 being the larger, is: db difference $= 10 \log_{10} P_2/P_1$.

In acoustics the threshold of hearing at 1,000 cycles per sec has been standardized at 10^{-16} watts per sq cm. If P_2 is the power in watts per square centimeter of a measured sound, then $10 \log_{10} P_2/10^{-16}$ is the db difference above the threshold and is known as the *intensity level*. This is a definite recognized way of describing the intensity of a sound.

Declination of Sun: The angle above or below equatorial plane. It is plus if north of the plane, and minus if below. Celestial objects are located by declination.

Degree Day: A unit, based upon temperature difference and time, used in estimating fuel consumption and specifying nominal heating load of a building in winter. For any one day, when the mean temperature is less than 65 F, there exist as many degree days as there are Fahrenheit degrees difference in temperature between the mean temperature for the day and 65 F.

Dehumidify: To reduce by any process, the quantity of water vapor within a given space.

Dehydrate: To remove water in all forms from matter. Liquid water, hygroscopic water, and water of crystallization or water of hydration are included.

Density: The ratio of the mass of a specimen of a substance to the volume of the specimen. The mass of a unit volume of a substance. When weight can be used without confusion, as synonymous with mass, density is the weight per unit volume.

Dew Point: See *Temperature, Dew Point*.

Diffusivity, Thermal: Thermal conductivity divided by product of density and specific heat $= k/pc$. Expressed in units of square feet per hour.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct-Return System (*Hot Water*): A hot water system in which the water, after it has passed through a heating unit is returned to the boiler along a direct path so that the total distance traveled by the water is the shortest feasible, and so that there are considerable differences in the lengths of the several circuits composing the system.

Down-Feed One-Pipe Riser (*Steam*): A pipe which carries steam downward to the heating units and into which the condensate from the heating units drains.

Down-Feed System (*Steam*): A steam heating system in which the supply mains are above the level of the heating units which they serve.

Draft: A current of air. Usually refers to the pressure difference which causes a current of air or gases to flow through a flue, chimney, heater, or space.

Draft Head (*Side Outlet Enclosure*): The height of a gravity convector between the bottom of the heating unit and the bottom of the air outlet opening. (*Top Outlet Enclosure*): The height of a gravity convector between the bottom of the heating unit and the top of the enclosure.

Drip: A pipe, or a steam trap and a pipe considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Dry: To separate or remove a liquid or vapor from another substance. The liquid may be water, but the term is also used for the removal of liquid or vapor forms of other substances.

Dust: An air suspension (aerosol) of solid particles of any material. (See also Chapter 7.)

Enthalpy: A term used in lieu of *total heat* or *heat content*. Expressible in Btu per pound. Mathematically defined as $h = u + pv/J$. When a change occurs at constant pressure, as when water is boiled, the change in enthalpy is equal to the heat added, in this case latent heat.

Enthalpy, Free: A thermodynamic property which serves as a measure of the available energy of a system with respect to surroundings at the same temperature and same pressure as that of the system. No process involving an increase in available energy can occur spontaneously.

Enthalpy, Specific: A term sometimes applied to enthalpy per unit weight, the English unit being Btu per pound.

Entropy: The ratio of the heat added to a substance to the absolute temperature at which it is added. Mathematically, for a reversible process, $dS = dQ/T$ or $S = \int dQ/T$.

These formulas are applicable when temperature is not constant. During a reversible adiabatic change, entropy is constant. During a reversible isothermal change, the heat absorbed by the substance is equal to the product of the absolute temperature of the substance and its change of entropy.

Entropy, Specific: A term sometimes applied to entropy per unit weight, the English unit being Btu per (Fahrenheit degree, absolute) (pound).

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at the same temperature and corresponding atmospheric pressure.

Fan Furnace System: See *Warm Air Heating System*.

Fog: Suspended liquid droplets generated by condensation from the gaseous to the liquid state, or by breaking up a liquid into a dispersed state, such as by splashing, foaming, and atomizing. (See also Chapter 7.)

Force: The action on a body which tends to change its relative condition as to rest or motion.

Fumes: Smoke; aromatic smoke; odor emitted, as of flowers; a smoky or vaporous exhalation, usually odorous, as that from concentrated nitric acid. The word *fumes* is so broad and inclusive that its usefulness as a technical term is very limited. Its principal definitive characteristic is that it implies an odor. The terms vapor, smoke, fog, etc., which can be more strictly defined, should be used whenever possible.

Also defined as solid particles generated by condensation from the gaseous state, generally, after volatilization from molten metals, etc., and often accompanied by a chemical reaction such as oxidation. Fumes flocculate and sometimes coalesce. (See also Chapter 7.)

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also a complete heating unit for transferring heat from fuel being burned to the air supplied to a heating system.

Furnace Volume (*Total*): The total furnace volume for horizontal-return tubular boilers and water-tube boilers is the cubical contents of the furnace between the grate and the first plane of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal-return tubular boiler settings, unless manifestly ineffective (i.e., no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally-fired boilers, it is the cubical contents of the furnace, flues, and combustion chamber, up to the plane of first entry into the tubes. (*ASME Power Test Codes*, Series 1929.)

Grate Area: The area of the grate surface, measured in square feet, to be used in estimating the rate of burning fuel. This area is construed to mean the area measured in the plane of the top surface of the grate, except that with special furnaces, such as those having magazine feed, or special shapes, the grate area shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down-draft boilers, the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as previously defined.

Gravity, Specific: The ratio of the mass of a unit volume of a substance to the mass of the same volume of a standard substance at a standard temperature. Water at 39.2 F is the standard substance usually referred to. For gases, dry air at the same temperature and pressure as the gas, is often taken as the standard substance.

Gravity Warm Air Heating System: See *Warm Air Heating System*.

Head, Dynamic: Same as Total Pressure expressed in height of liquid.

Heat: *That form of energy which transfers from one system* to another, by virtue of the temperature difference which exists between the two systems, when they are placed in communication.

Heat, Humid: Ratio of increase of enthalpy per pound of dry air to rise of temperature under conditions of constant pressure and constant humidity ratio.

Heat, Latent: A term used to express the energy involved in a change of state.

Heat, Sensible: A term used in heating and cooling to indicate any portion of heat which changes only the temperature of the substances involved.

Heat of the Liquid: The increase in enthalpy per unit weight of a saturated liquid as its temperature increases from a chosen base temperature. For water, the base temperature is usually taken as 32 F.

Heat Pump: In year-round air conditioning, refrigeration equipment used in such manner that heat is taken from a *heat source* and given up to the conditioned space when heating service is wanted, and is removed from the space and discharged to a *heat sink* when cooling and dehumidification are desired.

Heat, Specific: The heat absorbed (or given up) by a unit mass of a substance when its temperature is increased (or decreased) by 1 deg. Common Units: Btu per (pound) (Fahrenheit degree), calories per (gram) (Centigrade degree). For gases, both specific heat at constant pressure (c_p) and specific heat at constant volume (c_v) are frequently used. In air conditioning, c_p is usually used.

Heat Stress Index: A numerical index of the relative degree of heat stress imposed by various combinations of metabolic heat production with environmental radiant heat, air temperature, air velocity, and moisture content of the air. A young man in good physical condition and acclimatized to heat can withstand a total heat load having an HSI = 100 for eight hours a day of simple physical work without suffering undue physiological strain. Older men and others with physiological impairments and men engaged in more exacting work require lower *HSI* values.

Heat, Total: See *Enthalpy*.

Heat Transmission, Coefficient: Any one of a number of coefficients used in the calculation of heat transmission by conduction, convection, and radiation, through various materials and structures. (See thermal conductance, thermal conductivity, thermal resistance, thermal resistivity, thermal transmittance, etc.)

Heating Element, Electric: A unit assembly consisting of a resistor, insulated supports, and terminals for connecting the resistor to electric power.

Heating Unit, Electric: A structure containing one or more heating elements, electrical terminals or leads, electric insulation, and a frame or casing, all assembled together in one unit.

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried from the boiler to the heating units.

Humidify: To increase, by any process, the density of water vapor within a given space.

Humidistat: A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.

Humidity: Water vapor within a given space.

Humidity, Absolute: The weight of water vapor per unit volume, pounds per cubic foot, or grams per cubic centimeter.

Humidity, Relative: The ratio of the mol fraction of water vapor in a mixture to the mol fraction of water vapor in saturated air at the same dry-bulb temperature and barometric pressure. (See discussion in Chapter 3.) Also defined in various texts as the ratio of the actual partial pressure of the water vapor in a space to the saturation pressure of pure water at the same temperature.

Humidity Ratio: In a mixture of water vapor and air, the weight of water vapor per pound of dry air. Symbol W. Also called *Specific Humidity*.

Humidity, Specific: See *Humidity Ratio*.

Hygrostat: Same as *Humidistat*.

Inch of Water: A unit of pressure equal to the pressure exerted by a column of liquid water 1 in. high at a standard temperature. The standard temperature is sometimes taken as 0 C and sometimes as 62 F. One inch of water at 62 F = 5.197 lb per sq ft.

Insulation (*Thermal*): A material having a relatively high resistance to heat flow, and used principally to retard the flow of heat.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Load, Estimated Design: In a heating or cooling system, the sum of the useful heat transfer, plus heat transfer from or to the connected piping, plus heat transfer occurring in any auxiliary apparatus connected to the system. The units are Btu per hour or, in heating, equivalent direct radiation (EDR).

Load, Estimated Maximum: In a heating or cooling system, the calculated maximum heat transfer that the system will be called upon to provide.

Manometer: An instrument for measuring pressures; essentially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so constructed that the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: A measure of the inertia of a body. It also measures the quantity of matter in a body. Since the only general property of a given portion of matter that cannot be changed is its inertia, it is this property by which quantities of matter are defined. Two bodies which have equal inertias are said to have equal masses, or to contain equal quantities of matter. (This definition fails at velocities approaching the velocity of light.) The mass of a body is numerically equal to the ratio of the force required to give the body a given acceleration, to the acceleration. $m = F/a$. The common units of mass are the gram and the pound.

Mechanical Equivalent of Heat: The quantity of mechanical energy equal to one unit of heat. $J = 778.177$ ft-lb per Btu $= 4.1858 \times 10^7$ ergs per g-cal.

Medium, Heating: A substance such as water, steam, air, or furnace gas used to convey heat from the boiler, furnace, or other source of heat or energy to the heating unit from which the heat is dissipated.

Micron: A unit of length, the thousandth part of 1 mm or the millionth of a meter.

Millimeter of Mercury: A unit of pressure equal to the pressure exerted by a column of mercury 1 mm high at a temperature of 0 C. One millimeter of mercury at 0 C $= 1.934 \times 10^{-2}$ psi.

Mol: A weight of a substance numerically equal to its molecular weight. If the weight is in pounds the unit is a *Pound Mol*, in grams the unit is a *Gram Mol*. For perfect gases the volume of 1 mol is constant for all gases at the same temperature and pressure. For real gases this is approximately true at moderate pressures. At 32 F and zero-pressure the value of the product, pressure times specific volume, is 359.045 ± 0.006 atmosphere cubic feet (atm ft³), *for 1 mol of any gas*. For dry air at 32 F and standard atmospheric pressure, the specific volume is 358.83 cu ft per mol (ft³ per mol).

One-Pipe Supply Riser (*Steam*): A pipe which carries steam vertically to a heating unit, and which also carries the condensate from the heating unit. In an up-feed system, steam and condensate flow in opposite directions; in an overhead or down-feed system, they flow in the same direction.

One-Pipe System (*Steam*): A steam heating system in which a single main serves the dual purpose of supplying steam to the heating unit and conveying condensate from it. Ordinarily to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used. (*Hot Water*)—A hot water system in which the cooled water from the heating units is returned to the supply main. Consequently, the heating units farthest from the boiler are supplied with cooler water than those near the boiler in the same circuit.

Overhead System: Any steam or hot water system in which the supply main is above the heating unit. In a steam system the return must be below the heating units; in a water system the return may be above or below the heating units.

Panel Heating: A heating system in which heat is transmitted by both radiation and convection from panel surfaces to both air and surrounding surfaces.

Panel Radiator: A heating unit placed on or flush with a flat wall surface, and intended to function essentially as a radiator.

Perimeter System: See *Warm Air Heating System*.

Permeability: Water vapor permeability is a property of a substance which permits passage of water vapor, and is equal to the permeance of a 1 in. thickness of the substance. When permeability varies with psychrometric conditions, the *Spot* or *Specific Permeability* defines the property at a specific condition. Permeability is measured in perm-inches.

Permeance: The water vapor permeance of a sheet of any thickness (or assembly between parallel surfaces) is the ratio of water vapor flow to the vapor pressure difference between the surfaces. Permeance is measured in perms.

Perm: The unit of permeance. A perm is equal to 1 grain per (sq ft) (hr) (inch of mercury vapor pressure difference).

Plenum Chamber: An air compartment maintained under pressure, and connected to one or more distributing ducts.

Potentiometer: An instrument for comparing small electromotive forces, or for measuring small electromotive forces by comparison with a known electromotive force. Its principal advantage is that during the measurement, no current flows through the source of electromotive force.

Power: The rate of performing work. Common units are horsepower, Btu per hour, and watts.

Pressure: Force per unit area. Common units are pounds per square inch, gram per square centimeter, inch of water, millimeter of mercury.

Pressure, Absolute: The sum of the gage pressure and the barometric pressure.

Pressure, Dynamic: Same as *Total Pressure*.

Pressure, Gage: Pressure measured from atmospheric pressure as a base. Gage pressure may be indicated by a manometer which has one leg connected to the pressure source and the other exposed to atmospheric pressure.

Pressure, Saturation: The saturation pressure for a pure substance for any given temperature is that pressure at which vapor and liquid, or vapor and solid, can coexist in stable equilibrium.

Pressure, Static: The normal force per unit area that would be exerted by a moving fluid on a small body immersed in it if the body were carried along with the fluid. Practically, it is the normal force per unit area at a small hole in a wall of the duct through which the fluid flows (piezometer) or on the surface of a stationary tube at a point where the disturbances, created by inserting the tube, cancel. It is supposed that the thermodynamic properties of a moving fluid depend on static pressure in exactly the same manner as those of the same fluid at rest depend upon its uniform hydrostatic pressure.

Pressure, Total: In the theory of the flow of fluids; the sum of the static pressure and the velocity pressure at the point of measurement.

Pressure, Vapor: The pressure exerted by a vapor. If a vapor is kept in confinement over its liquid so that the vapor can accumulate above the liquid, the temperature being held constant, the vapor pressure approaches a fixed limit called the maximum, or saturated, vapor pressure, dependent only on the temperature and the liquid. The term *vapor pressure* is sometimes used as synonymous with *saturated vapor pressure*.

Pressure, Velocity: In a moving fluid, the pressure capable of causing an equivalent velocity, if applied to move the same fluid through an orifice such that all pressure energy expended is converted into kinetic energy.

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere.

Psychrometric: Pertaining to psychrometry or the state of the atmosphere with reference to moisture.

Psychrometric Chart: A graphical representation of the thermodynamic properties of moist air.

Psychrometry: The branch of physics relating to the measurement or determination of atmospheric conditions, particularly regarding the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiant Heating: A heating system in which only the heat radiated from panels is effective in providing the heating requirements. The term *Radiant Heating* is frequently used to include both *Panel and Radiant Heating*.

Radiation: The transmission of energy by means of electromagnetic waves.

Radiation, Thermal (Heat) Radiation: The transmission of energy by means of electromagnetic waves of very long wavelength. Radiant energy of any wavelength may, when absorbed, become thermal energy and result in an increase in the temperature of the absorbing body.

Radiation, Equivalent Direct (EDR): A unit of heat delivery of 240 Btu per hr. It does not imply 144 sq in. of surface.

Radiator: A heating unit exposed to view within the room or space to be heated. A radiator transfers heat by radiation to objects within visible range, and by conduction to the surrounding air which in turn is circulated by natural convection; a so-called radiator is also a convector, but the term radiator has been established by long usage.

Radiator, Concealed: A heating device located within, adjacent to, or exterior to the room being heated, but so covered or enclosed or concealed that the heat transfer surface of the device, which may be either a radiator or a convector, is not visible from the room. Such a device transfers its heat to the room largely by convection air currents.

Radiator, Direct: Same as *Radiator*.

Radiator, Recessed: A heating unit set back into a wall recess, but not enclosed.

Radiator, Tube or Tubular: A heating unit used as a radiator in which the heat transfer surfaces are principally tubes.

Refrigerant: A substance which produces a refrigerating effect by its absorption of heat while expanding or vaporizing.

Refrigeration, Ton of: The removal of heat at a rate of 200 Btu per min, 12,000 Btu per hr, or 288,000 Btu per 24 hr.

Resistance, Thermal: The reciprocal of thermal conductance. Symbol *R*.

Resistivity, Thermal: The reciprocal of thermal conductivity. Symbol *r*.

Resistor, Electric: A material used to produce heat by passing an electric current through it.

Return, Dry: A return pipe in a steam heating system which carries both water of condensation and air. The dry return is above the level of the water line in the boiler in a gravity system. (See *Return, Wet*.)

Return, Wet: That part of a return main of a steam heating system which is filled with water of condensation. The wet return usually is below the level of the water line in the boiler, although not necessarily so. (See *Return, Dry*.)

Return Mains: Pipes or conduits which return the heating or cooling medium from the heat transfer unit to the source of heat or refrigeration.

Reversed-Return System: A system in which the heating or cooling medium from several heat transfer units is returned along paths arranged so that all circuits composing the system or composing a major sub-division of it are of practically equal length.

Sabin: A unit of equivalent sound absorption equal to the equivalent absorption of one square foot of a surface of unit absorptivity (i.e., of one square foot of surface which absorbs all incident sound energy).

Saturation: The condition for co-existence in stable equilibrium of a vapor and liquid or a vapor and solid phase of the same substance. Example: Steam over the water from which it is being generated.

Saturation, Degree of: The ratio of the weight of water vapor associated with a pound of dry air to the weight of water vapor associated with a pound of dry air saturated at the same temperature.

Smoke: An air suspension (aerosol) of particles, usually but not necessarily solid, often originating in a solid nucleus, formed from combustion or sublimation. Also defined as carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion, and thereby to reduce the smoke produced.

Solar Constant: The solar intensity incident on a normal surface located outside the earth's atmosphere at a distance from the sun equal to the mean distance between the earth and the sun. Its value is 415, 445, or 430 Btu per (hr) (sq ft) as the July, January, or mean value, respectively. At sea level in July the solar intensity value is about 300 Btu per (sq ft) (hr) since about 28 percent is absorbed in the earth's atmosphere.

Sorbent: A material which extracts one or more substances present in an atmosphere or mixture of gases or liquids with which it is in contact, due to an affinity for such substances.

Sorption: Adsorption or absorption.

Split System: A system in which the heating is accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point. Ventilation may be provided by the same system.

Square Foot of Heating Surface (*Equivalent*): This term is synonymous with Equivalent Direct Radiation (EDR).

Stack Height: The height of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Steam: Water in the vapor phase. *Dry Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing no water in suspension. *Wet Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing water particles in suspension. *Superheated Steam* is steam at a temperature higher than the saturation temperature corresponding to the pressure.

Steam Heating System: A heating system in which heat is transferred from the boiler or other source of heat to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate, or air and condensate, and preventing the passage of steam.

Supply Mains: The pipes through which the heating medium flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface, Heating: The exterior surface of a heating unit: *Extended Heating Surface (or Extended Surface)*: Heating surface consisting of fins, pins, or ribs which receive heat by conduction from the prime surface. *Prime Surface*: Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also *Boiler Heating Surface*.)

Temperature: The thermal state of matter with reference to its tendency to communicate heat to matter in contact with it. If no heat flows upon contact, there is no difference in temperature.

Temperature, Absolute: Temperature expressed in degrees above absolute zero.

Temperature, Dew-Point: The temperature at which the condensation of water vapor in a space begins for a given state of humidity and pressure as the temperature of the vapor is reduced. The temperature corresponding to saturation (100 percent relative humidity) for a given absolute humidity at constant pressure.

Temperature, Dry-Bulb: The temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

Temperature, Effective: An arbitrary index which combines into a single value the effect of temperature, humidity, and air movement on the sensation of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation.

Temperature, Mean Radiant (MRT): The temperature of a uniform black enclosure in which a solid body or occupant would exchange the same amount of radiant heat as in the existing non-uniform environment.

Temperature Scales: Three temperature scales, Centigrade, Fahrenheit, and Reaumur, derive their degree values by dividing the difference between the ice point and steam points of water as follows: Centigrade 100, Fahrenheit 180, and Reaumur 80. Only the first two are important in the United States. The value of a Fahrenheit degree is therefore 5/9 of a Centigrade degree.

The Kelvin scale, sometimes called Centigrade absolute, has its zero point at the lowest attainable temperature 273.16 Centigrade degrees below the triple-point temperature of water and exactly 273.15 below the zero of the Centigrade scale.

The Rankine scale, sometimes called Fahrenheit absolute, has its zero at the lowest attainable temperature 459.688 Fahrenheit degrees below the triple-point temperature of water and exactly 459.67 F deg below the zero of the Fahrenheit scale.

To convert Centigrade to Kelvin temperature (generally designated K) add 273.15 deg.

To convert Fahrenheit to Rankine temperature (generally designated R) add 459.67 deg.

Temperature, Wet-Bulb: Thermodynamic wet-bulb temperature is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Wet-bulb temperature (without qualification) is the temperature indicated by a wet-bulb psychrometer constructed and used according to specifications. (*ASME Power Test Codes*, Series 1932, Instruments and Apparatus, Part 18.)

Therm: A quantity of heat equivalent to 100,000 Btu.

Thermodynamics, Laws of: Two laws upon which rest the classical theory of thermodynamics. These laws have been stated in many different, but equivalent ways. *The First Law:* (1) When work is expended in generating heat, the quantity of heat produced is proportional to the work expended; and conversely, when heat is employed in the performance of work, the quantity of heat which disappears is proportional to the work done. (Joule)[a] (G.P.)[b]; (2) If a system is caused to change from an initial state to a final state by adiabatic means only, the work done is the same for all adiabatic paths connecting the two states. (Zemansky); (3) In any power cycle or refrigeration cycle the net heat absorbed by the working substance is exactly equal to the net work done. *The Second Law:* (1) It is impossible for a self-acting machine, unaided by any external agency, to convey heat from a body of lower to one of higher temperature. (Clausius) (G.P.); (2) It is impossible to derive mechanical work from heat taken from a body unless there is available a body of lower temperature into which the residue not so used may be discharged. (Kelvin) (G.P.); (3) It is impossible to construct an engine that, operating in a cycle, will produce no effect other than the extraction of heat from a reservoir and the performance of an equivalent amount of work (Zemansky).

Thermostat: An instrument which responds to changes in temperature, and which directly or indirectly controls temperature.

Transmittance, Thermal: The time rate of heat flow, from the fluid on the warm side to the fluid on the cold side, per (square foot) (degree temperature difference between the two fluids). Sometimes called *Overall Coefficient of Heat Transfer*.

Common unit is Btu per (hour) (square foot) (Fahrenheit degree). Symbol *U*.

Two-Pipe System (*Steam or Water*): A heating system in which one pipe is used for the supply of the heating medium to the heating unit, and another for the return of the heating medium to the source of heat supply. The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium, which medium cannot have served a preceding heating unit.

Up-Feed System: A heating system in which the supply *mains are below the level of the heating units which they* serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vane Ratio: In air distributing devices the ratio of depth of vane to shortest opening width between two adjacent grille bars.

Vapor: The gaseous form of substances which are normally in the solid or liquid state, and which can be changed to these states either by increasing the pressure or decreasing the temperature. Vapors diffuse. (ASA definition.)

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensate to the boiler or receiver by gravity. Vapor systems have thermostatic traps or other means of resistance on the return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry-return.

Velocity: A vector quantity which denotes at once the time rate and the direction of a linear motion. $V = \frac{ds}{dt}$. For uniform linear motion $V = \frac{s}{t}$. Common units are: feet per second.

Ventilation: The process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning*.)

Viscosity: That property of semi-fluids, fluids, and gases *by virtue of which they resist an instantaneous change of* shape or arrangement of parts. It is the cause of fluid friction whenever adjacent layers of fluid move with relation to each other.

The coefficient of viscosity is the resistance offered by a layer of the fluid of unit area to the motion parallel to this area of another layer of the fluid at unit distance moving with unit velocity relative to the first layer. This coefficient is known as the *absolute viscosity*, and in cgs units is the force in dynes per square centimeter at a velocity of 1 cm per second at a distance of 1 cm. This unit of absolute viscosity is the *poise*. Values of absolute viscosity are frequently listed in centipoises, a centipoise being 1/100 of 1 *poise*.

In many formulas *absolute viscosity* is expressed in pounds per foot second or pounds per foot hour. Conversion from centipoises may be made as follows: viscosity in centipoises × 0.000672 = viscosity in pounds per foot second, or viscosity in centipoises × 2.42 = viscosity in pounds per foot hour.

Kinematic viscosity is the ratio of *absolute viscosity* to the density of a fluid. The unit is the stoke which equals one poise per cu cm per gram, or 1 sq cm per sec. For conversion to English units: kinematic viscosity in stokes × 0.001076 = kinematic viscosity in square feet per second.

Volume, Specific: The volume of a substance per unit mass; the reciprocal of density. Units: cubic feet per pound, cubic centimeters per gram, etc.

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to various rooms of the building through ducts.

Warm Air Heating System, Forced: A warm air heating system in which circulation of air is effected by a fan. Such a system may include air cleaning devices.

Warm Air Heating System, Gravity: A warm air heating system in which the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing.

Warm Air Heating System, Perimeter: A warm air heating system of the combination panel and convection type. Warm air ducts embedded in the concrete slab of a basementless house, around the perimeter, received heated air from a furnace and deliver it to the heated space through registers *placed in or near the floor. Air is returned to the furnace from* registers near the ceiling.

[a] Names of authors who first stated laws are given in parentheses.
[b] From *Glossary of Physics*, by LeRoy Dougherty Weld (McGraw-Hill, 1937).

THE GREEK ALPHABET

A α	Alpha	H η	Eta	N ν	Nu	T τ	Tau
B β	Beta	Θ ϑ θ	Theta	Ξ ξ	Xi	Υ υ	Upsilon
Γ γ	Gamma	I ι	Iota	O o	Omicron	Φ φ ϕ	Phi
Δ δ	Delta	K κ	Kappa	Π π	Pi	X χ	Chi
E ϵ	Epsilon	Λ λ	Lambda	P ρ	Rho	Ψ ψ	Psi
Z ζ	Zeta	M μ	Mu	Σ σ s	Sigma	Ω ω	Omega

CHAPTER 2

ABBREVIATIONS, SYMBOLS, CONVERSION FACTORS

Standard Abbreviations; Standard Symbols; Conversion Equations; Graphical Symbols for Piping, Ductwork, Heating and Ventilating, Refrigerating; Identification of Piping by Color

THIS chapter contains information regarding abbreviations, symbols, and conversion equations, which are of particular interest to the engineer engaged in heating, ventilating, and air conditioning.

ABBREVIATIONS

Abbreviations are shortened forms of names and expressions employed in texts and tabulations, and should not generally be used as symbols in equations. Most of the following abbreviations have been compiled from a list of approved standards.[1] In general, the period has been omitted in all abbreviations, except where the omission results in the formation of an English word.

Absolute	abs
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiling point	bp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu
British thermal units per hour	Btuh
Calorie	cal
Centigram	cg
Centimeter	cm
Centimeter-gram-second (system)	cgs
Cubic	cu
Cubic centimeter	cu cm or cc
Cubic foot	cu ft
Cubic feet per minute	cfm
Cubic feet per second	cfs
Decibel	db
Degree[2]	deg or °
Degree, Centigrade	C
Degree, Fahrenheit	F
Degree, Kelvin	K
Degree, Rankine	R
Diameter	dia
Direct-current (as adjective)	d-c
Electromotive Force	emf
Feet per minute	fpm
Feet per second	fps
Foot	ft
Foot-pound	*ft-lb*
Foot-pound-second (system)	fps
Freezing point	fp
Gallon	gal
Gallons per minute	gpm
Gallons per second	gps
Gram	g
Gram-calorie	g-cal
Horsepower	hp
Horsepower-hour	hp-hr
Hour	hr
Inch	in.
Inch-pound	in.-lb
Indicated horsepower	ihp
Indicated horsepower-hour	ihp-hr
Kilogram	kg
Kilowatt	kw
Kilowatt-hour	kwhr
Mass	mass
Melting point	mp
Meter	m
Micron	μ (mu)
Miles per hour	mph
Millimeter	mm
Minute	min
Molecular weight	mol wt
Mol	mol
Ounce	os
Pound	lb
Pounds per square inch	psi
Pounds per square inch, gage	psig
Pounds per square inch, absolute	psia
Revolutions per minute	rpm
Revolutions per second	rps
Second	sec
Specific gravity	sp gr
Specific heat	*sp ht*
Square foot	sq ft
Square inch	sq in.
Watt	w
Watthour	whr

[1] Abbreviations for Scientific and Engineering Terms, Z10.1-1941 (American Standards Association).
[2] It is recommended that the abbreviation for the temperature scale, F, C, K, R, be included in expressions for numerical temperatures but, wherever feasible, the abbreviation for *degree* be omitted *e.g.*, 68 F.

SYMBOLS

A letter symbol is a single character, with subscript or superscript if required, used to designate a physical magnitude in mathematical equations and expressions. Two or more symbols together always represent a product. The following have been compiled from a selected list of approved standards.[3] Additional symbols and variations in the standard symbols found necessary in the individual chapters will be found in a list at the end of Chapters 3, 4, 13, and 53.

Acceleration, due to gravity	*g*
Acceleration, linear	*a*

[3] Letter Symbols for Mechanics of Solid Bodies, Z10.3-1948, and Letter Symbols for Heat and Thermodynamics, Y10.4-1957 (American Standards Association).

Area ..... A
Change in specific volume during vaporization ..... v_{fg}
Density, Weight per unit volume, Specific weight. d or ρ (rho)

$$d = \frac{1}{v}$$

Distance, linear ..... s
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, vapor in contact with liquid ..... *Subscript g*
Efficiency ..... η
Elevation above some datum ..... z, Z
Emissivity ..... ϵ
Energy in general; work, total; work, molal ..... E
Entropy. (The capital should be used for any weight, and the small letter for unit weight) ..... S or s
Force, total load ..... F
Gas Constant, in equation $pV = nRT$ ..... R
Head ..... H or h
Heat content, Total heat, Enthalpy. (The capital should be used for any weight and the small letter for unit weight) ..... H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid ..... h_f
Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor ..... h_g
Heat of vaporization at constant pressure ..... L or h_{fg}
Hydraulic radius ..... R_H
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight) ..... U or u
Length of path of heat flow, thickness L
Load, total ..... W
Mechanical efficiency ..... e_m
Mechanical equivalent of heat ..... J
Power, Horsepower, Work per unit time ..... P
Pressure, Absolute pressure, Gage pressure, Force per unit area ..... p
Quantity (total) of fluid, water, gas, heat; Quantity by volume; Total quantity of heat transferred ..... Q
Quality of steam, Pounds of dry steam per pound of mixture x
Reynolds Number ..... N_{Re}
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor ..... *Subscript f*
Specific heat c
Specific heat at constant pressure ..... c_p
Specific heat at constant volume ..... c_v
Specific volume, Volume per unit weight, Volume per unit mass v
Temperature (ordinary) F or C. (*Theta* is used preferably only when t is used for Time in the same discussion) ..... t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature) ..... T or Θ (*capital theta*)
Thermal conductance: heat transferred per (unit time) (degree) C

$$C = \frac{1}{R} = \frac{kA}{L} = \frac{q}{t_1 - t_2}$$

Thermal conductance per unit area, Unit conductance: heat transferred per (unit time) (unit area) (degree) ..... C_o

$$C_o = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}$$

[4] Terms ending *ivity* designate properties independent of size or shape, sometimes called *specific properties*. Examples: conductivity, resistivity. Terms ending *ance* designate quantities depending not only on the material, but also upon size and shape, sometimes called *total quantities*. Examples: conductance, transmittance. Terms ending *ion* designate rate of heat transfer. Examples: conduction, transmission.

Thermal conductivity: heat transferred per (unit time) (unit area) (degree per unit length) ..... k

$$k = \frac{\frac{q}{A}}{\frac{(t_1 - t_2)}{L}}$$

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer: heat transferred per (unit time) (unit area) (degree) ..... f

$$f = \frac{\frac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film.)
Overall coefficient of heat transfer, Thermal transmittance per unit area: heat transferred per (unit time) (unit area) (degree overall) ..... U

$$U = \frac{\frac{q}{A}}{t_1 - t_2}$$

Thermal transmission (heat transferred per unit time) ..... q

$$q = \frac{Q}{t}$$

Thermal resistance (degree per unit of heat transferred per unit time) ..... R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity ..... $1/k$
Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure ..... *Subscript fg*
Velocity V
Viscosity, absolute ..... μ
Viscosity, kinematic ..... μ/ρ
Volume (total) V
Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight ..... q
Weight of a major item, Total weight ..... W
Weight rate, Weight per unit of power, Weight per unit time of ..... w
Work (total) ..... W

CONVERSION EQUATIONS[5]

Heat, Power and Work

1 ton refrigeration = { 12,000 Btu per hour; 200 Btu per minute }
Latent heat of ice = 143.4 Btu per pound
1 Btu = { 778.3 ft-lb; 0.2930 Int. whr; 252.0 I.T. calorie }
1 Int. watthour = { 2656 ft-lb; 3.413 Btu; 3600 Int. joules; 860 I.T. calories }
1 Int. kilowatthour = { 3,413 Btu; 3.517 lb water evaporated from and at 212 F }

[5] Checked in 1944 by National Bureau of Standards. Abbreviations *Int.* and *I.T.* refer to *International* and *International (Steam) Table*, respectively.

1 Int. kilowatt (1000 watts) = { 1.341 hp; 56.88 Btu per minute; 44,267 ft-lb per minute }
1000 I.T. calories / 1 I.T. kilocalorie = { 3.968 Btu; 3088 ft-lb; 1.1628 Int. whr }
1 horsepower = { 0.7455 Int. kw; 42.40 Btu per minute; 33,000 ft-lb per minute; 550 ft-lb per second }
1 boiler horsepower = { 33,475 Btu per hour; 9.809 Int. kw }

Weight and Volume

1 gal (U. S.) = { 231 cu in.; 0.1337 cu ft }
1 British or Imperial gallon = 277.42 cu in.
1 cu ft = { 7.481 gal; 1728 cu in. }
1 cu ft water at 60 F (in vacuo) = 62.37 lb
1 cu ft water at 212 F (" ") = 59.83 lb
1 gal water at 60 F (" ") = 8.338 lb
1 gal water at 212 F (" ") = 7.998 lb
1 lb (avdp) = { 16 oz; 7000 grains }
1 bushel = 1.244 cu ft
1 short ton = 2000 lb

Pressure

1 lb per square inch = { 144 lb per square foot; 2.0360 in. mercury at 32 F; 2.0422 in. mercury at 62 F; 2.309 ft water at 62 F; 27.71 in. water at 62 F }
1 oz per square inch = { 0.1276 in. mercury at 62 F; 1.732 in. water at 62 F }
1 atmosphere = { 14.696 lb per square inch; 2116 lb per square foot; 33.94 ft water at 62 F; 30.01 in. mercury at 62 F; 29.921 in. mercury at 32 F }
1 in. water at 62 F (in vacuo) = { 0.03609 lb per square inch; 0.5774 oz per square inch; 5.197 lb per square foot }
1 ft water at 62 F (in vacuo) = { 0.4330 lb per square inch; 62.37 lb per square foot }
1 in. mercury at 62 F (in vacuo) = { 0.4897 lb per square inch; 7.835 oz per square inch; 1.131 ft water at 62 F; 13.57 in. water at 62 F }
1 in. mercury at 32 F (in vacuo) = 0.49115 lb per square inch

Metric Units

1 cm = 0.3937 in. = 0.0328 ft
1 in. = 2.540 cm
1 m = 3.281 ft
1 ft = 0.3048 m
1 sq cm = 0.1550 sq in.
1 sq in. = 6.452 sq cm
1 sq m = 10.76 sq ft
1 sq ft = 0.09290 sq m
1 cu cm = 0.06102 cu in.
1 cu in. = 16.39 cu cm
1 cu m = 35.31 cu ft
1 cu ft = 0.02832 cu m
1 liter = 1000 cu cm = 0.2642 gal
1 kg = 2.205 lb (avdp)
1 lb = 0.4536 kg
1 metric ton = 2205 lb (avdp)
1 gram = 0.002205 lb (avdp)
1 kilometer per hour = 0.6214 mph
1 gram per square centimeter = { 0.02905 in. mercury at 62 F; 0.3942 in. water at 62 F }
1 kg per sq cm (metric atmosphere) = 14.22 lb per square inch
1 gram per cubic centimeter = { 0.03613 lb per cubic inch; 62.43 lb per cubic foot }
1 dyne = 0.00007233 poundals
1 absolute joule = { 10,000,000 ergs; 0.7376 ft-lb }
1 Int. joule = 0.7378 ft-lb
1 metric horsepower = { 75 kg-m per second; 0.986 hp (U. S.) }
1 I.T. kilocalorie per kilogram = 1.8 Btu per pound
1 I.T. calorie per square centimeter = 3.687 Btu per square foot
1 I.T. calorie per (second) (square centimeter) for a temperature gradient of 1 C deg per centimeter = { 2903 Btu per (hour) (square foot) for a temperature gradient of 1 F deg per inch of thickness. }

GRAPHICAL SYMBOLS FOR DRAWINGS[6]

Piping

Heating

1. High-Pressure Steam
2. Medium-Pressure Steam
3. Low-Pressure Steam
4. High-Pressure Return
5. Medium-Pressure Return
6. Low-Pressure Return
7. Boiler Blow Off
8. Condensate or Vacuum Pump Discharge
9. Feedwater Pump Discharge
10. Make-Up Water
11. Air Relief Line
12. Fuel Oil Flow — F
13. Fuel Oil Return — FOR
14. Fuel Oil Tank Vent — FOV
15. Compressed Air — A
16. Hot Water Heating Supply
17. Hot Water Heating Return

Air Conditioning

18. Refrigerant Discharge — RD
19. Refrigerant Suction — RS
20. Condenser Water Flow — C
21. Condenser Water Return — CR
22. Circulating Chilled or Hot Water Flow — CH

[6] Extracted from: American Standard Graphical Symbols for Pipe Fittings Valves, and Piping (*ASA* Z32.2.3-1949) and American Standard Graphical Symbols for Heating, Ventilating, and Air Conditioning (*ASA* Z32.2.4-1949) with the permission of the publisher, The American Society of Mechanical Engineers, 29 West 39th St., New York 18, N. Y.

23. Circulating Chilled or Hot Water Return — CHR
24. Make-Up Water
25. Humidification Line — H
26. Drain — D
27. Brine Supply — B
28. Brine Return — BR

Plumbing

29. Soil, Waste, or Leader (Above Grade)
30. Soil, Waste, or Leader (Below Grade)
31. Vent
32. Cold Water
33. Hot Water
34. Hot Water Return
35. Fire Line — F
36. Gas — G
37. Acid Waste — ACID
38. Drinking Water Flow
39. Drinking Water Return
40. Vacuum Cleaning — V
41. Compressed Air — A

Sprinklers

42. Main Supplies — S
43. Branch and Head
44. Drain — S

Heating

1. Air Eliminator
2. Anchor — PA
3. Expansion Joint
4. Hanger or Support — H
5. Heat Exchanger
6. Heat Transfer Surface, Plan (Indicate Type Such as convector)
7. Pump (Indicate Type Such as Vacuum) — M
8. Strainer
9. Tank (Designate Type) — REC
10. Thermometer
11. Thermostat — T
12. Traps
    - 12.1 Boiler Return
    - 12.2 Blast Thermostatic
    - 12.3 Float — F
    - 12.4 Float and Thermostatic
    - 12.5 Thermostatic
13. Unit Heater (Centrifugal Fan), Plan
14. Unit Heater (Propeller), Plan
15. Unit Ventilator, Plan
16. Valves
    - 16.1 Check
    - 16.2 Diaphragm
    - 16.3 Gate
    - 16.4 Globe
    - 16.5 Lock and Shield
    - 16.6 Motor Operated — M
    - 16.7 Reducing Pressure
    - 16.8 Relief (Either Pressure or Vacuum) — R
17. Vent Point — VENT

Ventilating

18. Access Door — AD
19. Adjustable Blank Off — TR 20X12
20. Adjustable Plaque — P-20X12-700Cfm / P-20"ϕ-700 Cfm

Ventilating

21. Automatic Dampers — M
22. Canvas Connections
23. Deflecting Damper
24. Direction of Flow
25. Duct (1st Figure, Side Shown; 2nd Side Not Shown) — 12 X 20
26. Duct Section (Exhaust or Return) — (E OR R 20X12)
27. Duct Section (Supply) — (S 20X12)
28. Exhaust Inlet Ceiling (Indicate Type) — CR 20X12 - 700 Cfm / CG 20X12 - 700 Cfm
29. Exhaust Inlet Wall (Indicate Type) — TR-12X8 700Cfm
30. Fan and Motor With Belt Guard
31. Inclined Drop in Respect to Air Flow — D
32. Inclined Rise in Respect to Air Flow — R
33. Intake Louvers on Screen
34. Louver Opening — L 20X12-700 Cfm
35. Supply Outlet Ceiling (Indicate Type) — 20" DIAM. 1000 Cfm
36. Supply Outlet Wall (Indicate Type) — TR-12X8 700 Cfm
37. Vanes
38. Volume Damper

Air Conditioning

39. Capillary Tube
40. Compressor
41. Compressor, Enclosed, Crankcase, Rotary, Belted
42. Compressor, Open Crankcase, Reciprocating, Belted
43. Compressor, Open Crankcase, Reciprocating, Direct Drive
44. Condenser, Air Cooled, Finned, Forced Air
45. Condenser, Air Cooled, Finned, Static
46. Condenser, Water Cooled, Concentric Tube in a Tube
47. Condenser, Water Cooled, Shell and Coil
48. Condenser, Water Cooled, Shell and Tube
49. Condensing Unit, Air Cooled
50. Condensing Unit, Water Cooled
51. Cooling Tower
52. Dryer
53. Evaporative Condenser
54. Evaporator, Circular, Ceiling Type, Finned
55. Evaporator, Manifolded, Bare Tube, Gravity Air
56. Evaporator Manifolded, Finned, Forced Air
57. Evaporator Manifolded, Finned, Gravity Air
58. Evaporator, Plate Coils, Headered or Manifold
59. Filter, Line
60. Filter and Strainer, Line
61. Finned Type Cooling Unit, Natural Convection
62. Forced Convection Cooling Unit

63. Gage
64. High Side Float
65. Immersion Cooling Unit
66. Low Side Float
67. Motor-Compressor, Enclosed Crankcase, Reciprocating, Direct Connected
68. Motor-Compressor, Enclosed Crankcase, Rotary, Direct Connected
69. Motor-Compressor, Sealed Crankcase, Reciprocating
70. Motor-Compressor, Sealed Crankcase, Rotary
71. Pressurestat
72. Pressure Switch
73. Pressure Switch With High Pressure Cut-Out
74. Receiver, Horizontal
75. Receiver, Vertical
76. Scale Trap
77. Spray Pond
78. Thermal Bulb
79. Thermostat (Remote Bulb)
80. Valves
    - 80.1 Automatic Expansion
    - 80.2 Compressor Suction Pressure Limiting, Throttling Type (Compressor Side)
    - 80.3 Constant Pressure, Suction
    - 80.4 Evaporator Pressure Regulating, Snap Action
    - 80.5 Evaporator Pressure Regulating, Thermostatic Throttling Type
    - 80.6 Evaporator Pressure Regulating, Throttling Type (Evaporator Side)
    - 80.7 Hand Expansion
    - 80.8 Magnetic Stop
    - 80.9 Snap Action
    - 80.10 Suction Vapor Regulating
    - 80.11 Thermo Suction
    - 80.12 Thermostatic Expansion
    - 80.13 Water
81. Vibration Absorber, Line

IDENTIFICATION OF PIPING SYSTEMS BY COLOR

The color scheme for identification of piping systems, based on material carried, as listed in the following table and shown in Fig. 1, is reprinted from Part V, Fourth Edition, of the Engineering Standards of the Heating, Piping and Air Conditioning Contractors National Association.[1]

Class	Color
F—Fire-protection	Red
D—Dangerous materials	Yellow or Orange
S—Safe materials	Green (or the achromatic colors, white, black, gray or aluminum)
and, when required	
P—Protective materials	Bright blue
V—Extra valuable materials	Deep purple

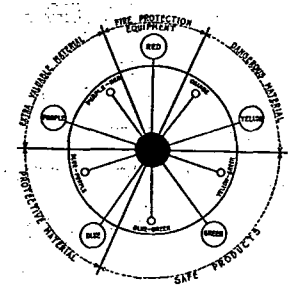


Fig. 1.... Main Classification by Color

[1] See Scheme for Identification of Piping Systems, A13.1-1956, American Standards Association.

CHAPTER 3

THERMODYNAMICS

Mass and Energy Balances, Thermodynamic Properties of Moist Air, Formulas and Tables; Thermodynamic Properties of Water, Formulas and Tables; Degree of Saturation; ASHAE PSYCHROMETRIC CHART; Solution of Air Conditioning Problems by Use of Tables and Psychrometric Chart; U. S. Standard Atmospheres

THERMODYNAMICS is that branch of natural science which deals with energy and its transformations into various forms. In this chapter the discussion will be limited to thermodynamics as it affects the arts of heating and air conditioning. This will necessarily presume some knowledge of the fundamentals of the science on the part of the reader who may also find it desirable to refer to a standard text on the subject, preferably one published after 1930.

MASS AND ENERGY BALANCES

The First Law of Thermodynamics is a statement of the Principle of Conservation of Energy. It may be stated as follows: *The energy added to a system is equal to the increase or decrease of the energy stored in the system, plus the energy which leaves the system.* For a completely contained, or nonflow system, this may be restated as: The heat added to a nonflow system is equal to the change in the internal energy of the system, plus the work done by the system.

$$ {}_1q_2 = U_2 - U_1 + w \qquad (1) $$

For a constant pressure process

$$ {}_1q_2 = H_1 - H_2 \qquad (2) $$

where

${}_1q_2$ = energy added between points 1 and 2.
U = internal energy of system.
w = work done by system.
H = enthalpy of system.

Subscripts 1 and 2 refer to sections of the system between which a change takes place.

For engineering problems, a more important application of the First Law is its use in cases in which, in addition to energy, one or more fluids are crossing the boundaries of the system. The most simple of these is the steady flow system, in which the rates of energy and mass flow across the boundaries of the system are constant, and no mass or energy is stored or released by the system.

Consider a system as illustrated in Fig. 1. The fluid crossing the boundaries of the system carries with it potential energy by reason of its elevation above some convenient datum, kinetic energy by reason of its velocity, and energy in the form of enthalpy. Additional energy may cross the boundaries of the system in the forms of heat or work. The various forms of energy crossing the boundaries between the sections under consideration may be equated by applying the First Law of Thermodynamics:

$$ PE_1 + KE_1 + H_1 + {}_1q_2 = PE_2 + KE_2 + H_2 + w \qquad (3) $$

where

PE = potential energy, Btu per pound dry air.
KE = kinetic energy, Btu per pound dry air.
H = enthalpy, Btu per pound dry air.
${}_1q_2$ = heat added between sections 1 and 2, Btu per pound dry air.
w = shaft work withdrawn between sections 1 and 2, Btu per pound of dry air.

For most psychrometric problems, since the change in the potential energy and kinetic energy terms is negligible compared to the enthalpy change, Equation 3 may be written

$$ H_1 + {}_1q_2 = H_2 + w \qquad (4) $$

where

H = enthalpy of flowing medium, Btu per pound of dry air.

The enthalpy of the entire system may be broken down into constituent parts, thus:

$$ GH = Gh + Lh_{wl} + Sh_{ws} \qquad (5) $$

where

h = enthalpy of *moist* air, Btu per pound of dry air.
h_{wl} = enthalpy of liquid water, Btu per pound.
h_{ws} = enthalpy of solid water, Btu per pound.
L = flow rate of liquid water, pounds per hour.
S = flow rate of solid water, pounds per hour.
G = flow rate of dry air, pounds per hour.

Similarly, an equation expressing the conservation of mass may be written thus:

$$ \sum_{in} [G(1 + W) + L + S] = \sum_{out} [G(1 + W) + L + S] \qquad (6) $$

where

W = humidity ratio, pounds of water vapor per pound of dry air.

THERMODYNAMIC PROPERTIES OF MOIST AIR

The working substance of the air-conditioning engineer is moist air. Air is actually a mixture of oxygen, nitrogen, carbon dioxide, water vapor, and traces of other gases.

The mixture consisting of the components other than water vapor is known as *dry air*. Its composition remains essentially constant under all conditions. In moist air the amount of

water vapor varies considerably. To allow for this variation the specific properties of moist air are developed in terms of the relative amounts of water vapor and dry air. Accepted air-conditioning practice is to express this in terms of the amount of water vapor per pound of dry air.

Terms frequently used in describing the condition of a mixture of air and water vapor are *humidity ratio, relative humidity, degree of saturation, dry-bulb temperature, thermodynamic wet-bulb temperature,* and *dew-point temperature.* These terms are defined in following paragraphs.

Humidity Ratio. Weight of water vapor associated with unit weight of dry air, pounds of water vapor per pound of dry air. Humidity ratio has also been called *specific humidity,* and this term is still used in many places.

Relative Humidity. Ratio of the mol fraction of water vapor in the actual mixture to the mol fraction of water vapor in

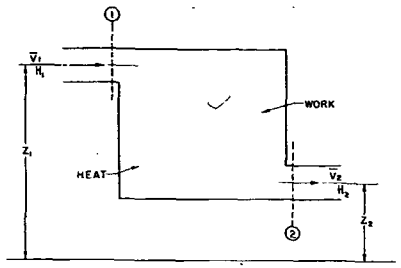


Fig. 1 . . . . Energy Change between Two Sections of a System

saturated air at the same dry-bulb temperature and barometric pressure.

Degree of Saturation. Ratio of the actual humidity ratio to the humidity ratio of saturated air at the same dry-bulb temperature and barometric pressure.

Relative humidity and degree of saturation are related according to the identity:

$$\phi = \frac{\mu}{\left[1 - (1 - \mu) f_s \frac{P_s}{P_a}\right]} \tag{7}$$

where

ϕ = relative humidity, expressed as a decimal.
μ = degree of saturation, expressed as a decimal.
P_a = observed (or barometric) pressure of the moist air.
P_s = saturation pressure of pure water at the prevailing temperature, expressed in the same units as P_a.
f_s = a dimensionless factor which may be regarded as accounting for influences arising when air and water are intermixed. Magnitudes of f_s have been reported by Goff and Gratch[1] and by Goff.[2] Table 1 gives values of f_s for a limited range of conditions.

Dry-Bulb Temperature. The temperature indicated by any type of thermometer or thermocouple not affected by the water-vapor content of the air, or by radiation.

Table 1 . . . . Magnitudes of f_s for the Range 0 to 125 F
(Standard Barometric Pressure, 29.921 in. Hg)

Temp. F	f_s	Temp. F	f_s
0	1.0048	70	1.0045
10	1.0046	80	1.0047
20	1.0046	90	1.0048
30	1.0045	100	1.0050
40	1.0044	110	1.0053
50	1.0044	120	1.0055
60	1.0044	125	1.0057

Note: The original source gives f_s to seven significant figures over the temperature range −208 F to +202 F and over the pressure range 20 to 35 in. Hg.

Thermodynamic Wet-Bulb Temperature. The temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature.

Consider an adiabatic system as shown in Fig. 2. Unsaturated air at the state h_1, W_1, enters the system at Section 1, and saturated air at the state h_1,* W_1,* leaves the system at Section 2. Liquid water at the state h_{w_2},* corresponding to the temperature of the saturated air leaving the system is supplied. Then, since no work is done and the system is strictly adiabatic, the energy equation becomes

$$h_1 + (W^* - W_1)h_{w^*} = h^* \tag{8}$$

where

* indicates condition at thermodynamic wet-bulb temperature.

The temperature corresponding to h^* for given values of h_1 and W_1 is called the thermodynamic wet-bulb temperature, or the temperature of adiabatic saturation.

The temperature indicated by an ordinary wet-bulb thermometer is affected by a number of factors not accounted for in Equation 8, and hence, may be quite different from the theoretical temperature obtained from its use. The measured wet-bulb temperature is influenced by (*a*) radiation from the *surroundings* to the wick; (*b*) *conduction* of heat along the stem of the thermometer; and (*c*) impact of the air on the wick or bulb of the thermometer. Arnold[3] has developed a theory which makes possible the calculation of the true thermodynamic wet-bulb temperature from observed data through the use of suitable corrections to be applied to the

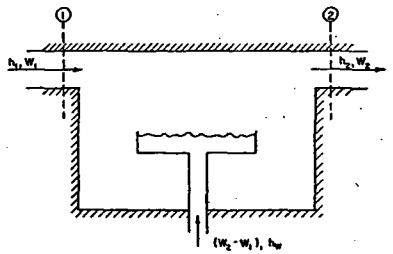


Fig. 2 . . . . Illustration of Adiabatic Saturation

readings of the wet-bulb thermometer. However, unless extreme precision is required, the observed temperature may be taken equal to the theoretical temperature for most engineering problems, if no attempt is made to shield the wick from radiation and the air velocity past the wick is about 1000 fpm.

Dew-Point Temperature. The saturation temperature corresponding to a given combination of humidity ratio W and barometric pressure is called the dew-point temperature. It is the lowest temperature at which the given humidity ratio can exist at the corresponding barometric pressure. At this temperature condensation will first start to form when moist air is cooled.

Perfect Gas Relationships

Hypotheses, based on experimental observation of the physical behavior of gases, which were advanced by Boyle, Charles, Gay-Lussac, Dalton, Gibbs, Joule, Kelvin, and others, were reduced to reasonably simple mathematical expressions and came to be regarded as physical laws. However, as scientific knowledge increased, and as more precise methods of measurement were developed, it became apparent that these simple equations did not describe the behavior of the mixtures of actual gases and vapors accurately.

The original statements have been found to be useful tools, nevertheless, in many cases. For example, the behavior of common diatomic and triatomic gases at low pressures follows these equations closely enough so that they may be used for some types of engineering problems. Many useful engineering works have been constructed through the use of approximations. The degree of approximation, however, which may be tolerated in engineering design must be decided by the engineer, based upon his study and experience in the field.

Boyle's Law. One of the original observations of the physical behavior of gases was made by Robert Boyle who noted that, if a constant weight of gas is compressed with the temperature held constant, the volume V varied inversely as the absolute pressure P. Stated mathematically,

$$PV = \text{constant} \quad \text{(temperature constant)} \tag{9}$$

Charles' Law. Experiments made independently by Charles and Gay-Lussac led to the formulation of what is now known as Charles' Law: If a constant weight of gas is heated or cooled at constant volume, the absolute pressure P varies as the absolute temperature T; if a constant weight of gas is heated or cooled at constant pressure, the volume V varies as the absolute temperature T. Stated mathematically,

$$\frac{P}{T} = \text{constant} \quad \text{(volume constant)} \tag{10}$$

$$\frac{V}{T} = \text{constant} \quad \text{(pressure constant)} \tag{11}$$

Boyle's Law and Charles' Law may be combined to form the equation of state for the ideal or perfect gas,

$$PV = RT \tag{12}$$

where

R is a constant whose value depends on the units selected for P, V, and T.

Dalton's Rule. Dalton stated that each gas in a mixture occupies the total volume of the mixture just as though the other gases were not present. Gibbs later expanded this statement for perfect gases into the following principles:

1. The pressure of a mixture of gases is the sum of the partial pressures of the individual gases when they exist at the total volume and temperature of the mixture.

2. The internal energy, enthalpy, and entropy of a mixture of gases are respectively equal to the sums of the individual internal energies, enthalpies, and entropies of the components when they exist at the total volume and temperature of the mixture.

While these relationships do not hold exactly for all systems of real gases, they may be used with a good degree of precision for many engineering applications at low pressures. Moreover, since water vapor very closely follows the perfect gas relationships in the range usually encountered in air conditioning, the Gibbs-Dalton Rule may frequently be applied to mixtures of dry air and water vapor.

Thus,
$$V_m = V_a = V_w \tag{13}$$
$$T_m = T_a = T_w \tag{14}$$
$$p_m = p_a + p_w \tag{15}$$
$$m_m h = m_a h_a + m_w h_w \tag{16}$$

where

Subscript m denotes mixture; subscript a denotes dry air; subscript w denotes water vapor.
Symbol m = weight of dry air crossing any duct section, pounds per minute.

Using Equations 12, 13, 14, and 15, the relation is obtained as follows:

$$v_T = \frac{n_a RT}{p_a} = \frac{n_w RT}{p_w} = \frac{(n_a + n_w)RT}{p} \tag{17}$$

where

v_T = total volume, cubic feet.
n_a = number of mols of dry air.
n_w = number of mols of water vapor.
R = universal gas constant, 1545 foot-pounds per (Fahrenheit degree) (mol).
p_a = partial pressure of dry air.
p_w = partial pressure of water vapor.
T = absolute temperature, Fahrenheit degrees.

The partial pressure of water vapor in the mixture is then

$$p_w = \frac{n_w}{n_a + n_w} p \tag{18}$$

or, the partial pressure of the water vapor in moist air is equal to the product of the mol fraction of the water vapor and the observed pressure of the mixture. A similar expression is obtained for the dry air.

Assuming that the perfect gas laws can be applied to water vapor at saturation at low pressures, the partial pressure of water vapor in a *saturated* mixture may be written as

$$p_s = \frac{n_s}{n_a + n_s} p \tag{19}$$

where

n_s = number of mols of water vapor at saturation.

The relative humidity may be obtained by combining

Equations 18 and 19 and solving for the ratio of mol fractions. Thus, using perfect gas relationships,

$$\phi = \frac{p_w}{p_a} \tag{20}$$

The humidity ratio W may be obtained from Equation 17:

$$W = \frac{18.016}{28.961}\frac{p_w}{p_a} = 0.622\,\frac{p_w}{p - p_w} \tag{21}$$

where 18.016 and 28.966 are the molecular weights of water and dry air, respectively.

Equation 16 may be rewritten as

$$h = h_a + Wh_w \tag{22}$$

where

h = enthalpy, Btu per pound of dry air.

In relating the enthalpy to the state of the moist air, the fact that in all applications only differences in enthalpy are involved, allows the arbitrary selection of a datum or zero enthalpy point. Accordingly, from perfect gas relationships, it is possible to write for any temperature t (Fahrenheit) greater than 0 F

$$h_a = 0.24\,t \tag{23}$$

where it is assumed that the same arbitrary datum of 0 F is used as in determining the properties of moist air in Table 2.

Tables of Thermodynamic Properties of Moist Air

Research work conducted at the University of Pennsylvania and at other institutions has shown that the Gibbs-Dalton Rule is inaccurate in varying degrees, depending on temperature, pressure, and the amount of water vapor present. This inaccuracy is probably due to the effect of:

1. Chemical solution of gas molecules in the water vapor.
2. The finite size of the molecules causing interference with the free passage of other molecules toward the boundaries of the system.
3. Intermolecular forces of attraction and repulsion.

Many attempts have been made to develop an equation of state which would predict the true states of real gases and vapors. The Van der Waal, Maxwell, and Beattie-Bridgman equations are probably the best known. Unfortunately, these expressions rapidly become much too complicated to be used in everyday calculations and, therefore, engineers find it more convenient to use tables of thermodynamic properties for specific working substances, as these can be prepared by physicists using the best laboratory equipment and all the refinements of mathematics.

Mechanical engineers have long been familiar with such tables for the properties of steam. Tables of the properties of moist-air, as prepared by Goodenough and others, have been available for some time, but the latest and most precise of such tables are those which have resulted from a cooperative research agreement between the American Society of Heating and Air-Conditioning Engineers and the Towne Scientific School of the University of Pennsylvania. These properties are published herein as Table 2, and are taken from a research report by Goff and Gratch.[4] Table 2, which experimentally and mathematically takes into account deviations from perfect gas behavior, such as those listed above, makes the application of the Gibbs-Dalton Rule a less frequent necessity.

In Table 2 there are 15 columns of figures, each column being headed by a suitable symbol. In the following subparagraphs brief explanations are given of the data in Table 2 under the appropriate column headings.

t(F) = Fahrenheit temperature defined in terms of absolute temperature T by the relation,

$$T = t + 459.67 \tag{24}$$

W_s = humidity ratio at *saturation*. Saturation is the condition at which the vapor phase (moist air) may exist in equilibrium with a condensed phase (liquid or solid) at the given temperature and pressure (standard atmospheric pressure in the case of Table 2). At given values of temperature and pressure, the humidity ratio W can have any value from zero to W_s.

v_a = specific volume of dry air, cubic feet per pound.

$v_{aa} = v_s - v_a$, the difference between the volume of moist air *at saturation*, per pound of dry air, and the specific volume of the dry air itself, cubic feet per pound of dry air.

v_s = specific volume of moist air *at saturation* per pound of dry air, cubic feet per pound of dry air.

h_a = specific enthalpy of dry air, Btu per pound of dry air. The specific enthalpy of dry air has been assigned the value zero at 0 F, standard atmospheric pressure. The energy unit Btu is related to the foot-pound by definition, as follows: 1 Btu = 778.3 ft-lb.

$h_{aa} = h_s - h_a$, the difference between the enthalpy of moist air *at saturation*, per pound of dry air, and the specific enthalpy of the dry air itself, Btu per pound of dry air.

h_s = enthalpy of moist air *at saturation* per pound of dry air, Btu per pound of dry air.

s_a = specific entropy of dry air, Btu per (pound) (Fahrenheit degree). It will be noticed that the specific entropy of dry air has been assigned the value zero at 0 F and standard atmospheric pressure.

$s_{aa} = s_s - s_a$, the difference between the entropy of moist air *at saturation*, per pound of dry air, and the specific entropy of the dry air itself, Btu per (pound of dry air) (Fahrenheit degree).

s_s = entropy of moist air *at saturation* per pound of dry air, Btu per (pound of dry air) (Fahrenheit degree).

h_w = specific enthalpy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per pound of water. The specific enthalpy of liquid water has been assigned the value zero at 32 F, saturation pressure (0.088586 psia).

s_w = specific entropy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per (pound of water) (Fahrenheit degree). The specific entropy of liquid water has been assigned the value zero at 32 F, saturation pressure (0.088586 psia).

p_s = saturation pressure of pure water vapor, pounds per square inch or inches of mercury (absolute pressure). At a given pressure, moist air can be saturated at any temperature, though this requires that it have a definite humidity ratio W, and that the coexisting condensed phase contain a definite, but very small, quantity of dissolved air. On the other hand, pure water vapor (steam) below the critical temperature, can be saturated at only one temperature for a given pressure. The values of saturation pressure listed in Table 2 have been computed from the formulas of Goff and Gratch.

THERMODYNAMIC PROPERTIES OF WATER AT SATURATION

Since water vapor at low pressures acts almost as a perfect gas, the enthalpy of water vapor should also be a function only of the temperature within these limits. Therefore, the enthalpy of the water vapor may be expressed as approximately equal to the enthalpy of saturated vapor at the dry-bulb temperature of the mixture. Substituting these values in Equation 22, the enthalpy of the mixture becomes

$$h = 0.24\,t + Wh_s \tag{25}$$

where h_s is the value of the enthalpy of saturated vapor at the temperature t, and is obtained from Table 3.

Table 3 offers revisions to existing steam table data with extensions downward to −160 F. These revisions and extensions were a necessary preliminary to the construction of Table 2. A detailed explanation of the methods employed in the construction of Table 3 is given in a paper by John A. Goff and S. Gratch.[5]

As in Table 2, the temperature scale used as argument in Table 3 is the Fahrenheit scale defined in terms of absolute temperature T by Equation 24. The symbols used as column headings in Table 3 are the same as those used in steam tables, and have the same meanings.

Properties of water above 212 F from Keenan and Keyes[6] are given in Table 4.

DEGREE OF SATURATION

Degree of saturation has previously been defined as the ratio of the actual humidity ratio to the humidity ratio of saturated air at the same dry-bulb temperature and barometric pressure. This may be stated mathematically as

$$\mu = \frac{W}{W_s} \tag{26}$$

Obviously the degree of saturation μ can have any value from zero (dry air) to unity (moist air at saturation). The degree of saturation is conveniently used to interpolate values for the enthalpy, specific volume, and entropy of moist air from the data of Table 2. Within the estimated precision of the data of Table 2, at temperatures below 150 F, the volume v, enthalpy h, and entropy s, of moist air *per pound of dry air* at any degree of saturation μ may be computed from the simple relations:

$$v = v_a + \mu v_{aa} \tag{27}$$

$$h = h_a + \mu h_{aa} \tag{28}$$

$$s = s_a + \mu s_{aa} \tag{29}$$

Thus, the degree of saturation is used in conjunction with Table 2 in the same manner as the quality is used with tables of the thermodynamic properties of steam.

Correction of Table 2 for Temperatures Above 150 F

The simple relations expressed in Equations 27, 28, and 29 give the properties of unsaturated air with satisfactory precision for most engineering design problems. Above 150 F, when greater precision than that obtained by Table 2 is required, these simple relations can be adjusted by the addition of supplementary terms.

To correct the volume, it is necessary to add a correction term $\bar{v}$ which is defined as

$$\bar{v} = \frac{\mu(1-\mu)A}{1 + aW_s\mu} \tag{30}$$

where a denotes the ratio of the apparent molecular weight of dry air (28.966) to the molecular weight of water (18.016), and is equal to 1.6078. Table 5 gives the values of the coefficient A for several higher temperatures, the value of μ at which the correction term $\bar{v}$ attains its maximum value, and the maximum value of $\bar{v}$ term there attained.

The correction term for the enthalpy is

$$\bar{h} = \frac{\mu(1-\mu)B}{1 + aW_s\mu} \tag{31}$$

Table 5 gives the values of the coefficient B and maximum values of $\bar{h}$, the maximum values occurring at the same degree of saturation as $\bar{v}$.

Corrections for the entropy consist of two terms: $\bar{s}$ which is defined as

$$\bar{s} = \frac{\mu(1-\mu)C}{1 + aW_s\mu} \tag{32}$$

and s, the so-called *mixing entropy*, which contributes the larger part of the error. The *mixing entropy* is defined as

$$s = 0.1579\,[(1 + \mu aW_s)\log_{10}(1 + \mu aW_s) - \mu aW_s\log_{10}(\mu) - \mu(1 + aW_s)\log_{10}(1 + aW_s)] \tag{33}$$

Table 5 lists the values of the coefficient C, the maximum values of $\bar{s}$ and $\bar{\bar{s}}$ and the values of μ at which they occur. The maximum $\bar{s}$ occurs at the same degree of saturation as the maximum values of $\bar{v}$ and $\bar{h}$.

THE ASHAE PSYCHROMETRIC CHART

A psychrometric chart is a graphical representation of the thermodynamic properties of moist air. To be of real value in the solution of engineering problems, it must have distinctive features which aid in problem analysis.

An examination of psychrometric mass and energy balances shows that no properties other than *enthalpy* and *humidity ratio* are required for the solution of problems. It is logical, then, to use these properties as the coordinates of a psychrometric chart. This was initially done by Mollier in 1923,[7, 8] and is the arrangement followed in the chart included with The Guide. The ASHAE Psychrometric Chart is based on the best thermodynamic data available today, namely, those of Goff and Gratch, as given in Table 2. The chart is plotted on oblique coordinates of enthalpy and humidity ratio; the enthalpy axis making an angle of approximately 40 deg with the humidity ratio axis. This is shown in Fig. 3.

For practical use, the inclusion of dry-bulb and wet-bulb temperatures, volumes, and indexes of the condition of the

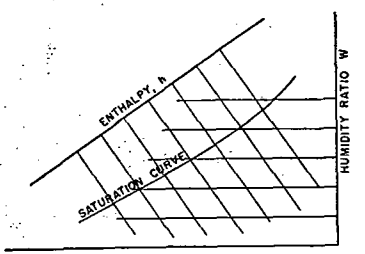


Fig. 3 .... Basic Coordinates of ASHAE Psychrometric Chart

Table 2 .... Thermodynamic Properties of *MOIST AIR*[a] (Standard Atmospheric Pressure, 29.921 in. Hg)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^4$	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per(°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/lb h_w	Entropy Btu per (°F)(Lb) s_w	Vap. Press In. Hg $p_s \times 10^3$	Fahr. Temp. t(F)
−160	0.0002120	7.520	0.000	7.520	−38.504	0.000	−38.504	−0.10300	0.00000	−0.10300	−223.00	−0.4907	0.0001009	−160
−155	0.0003869	7.647	0.000	7.647	−37.296	0.000	−37.296	−0.09901	0.00000	−0.09901	−220.40	−0.4863	0.0001842	−155
−150	0.0006932	7.775	0.000	7.775	−36.088	0.000	−36.088	−0.09508	0.00000	−0.09508	−218.77	−0.4800	0.0003301	−150
−145	0.001219	7.902	0.000	7.902	−34.881	0.000	−34.881	−0.09121	0.00000	−0.09121	−217.12	−0.4747	0.0005807	−145
−140	0.002109	8.029	0.000	8.029	−33.674	0.000	−33.674	−0.08740	0.00000	−0.08740	−215.44	−0.4695	0.001004	−140
−135	0.003586	8.156	0.000	8.156	−32.468	0.000	−32.468	−0.08365	0.00000	−0.08365	−213.73	−0.4643	0.001707	−135
−130	0.006000	8.283	0.000	8.283	−31.262	0.000	−31.262	−0.07997	0.00000	−0.07997	−212.03	−0.4590	0.002858	−130
−125	0.009857	8.411	0.000	8.411	−30.057	0.000	−30.057	−0.07634	0.00000	−0.07634	−210.28	−0.4538	0.004710	−125
−120	0.01606	8.537	0.000	8.537	−28.852	0.000	−28.852	−0.07277	0.00000	−0.07277	−208.52	−0.4485	0.007653	−120
−115	0.02571	8.664	0.000	8.664	−27.648	0.000	−27.648	−0.06924	0.00000	−0.06924	−206.72	−0.4433	0.01226	−115
−110	0.04063	8.792	0.000	8.792	−26.444	0.000	−26.444	−0.06577	0.00000	−0.06577	−204.93	−0.4381	0.01939	−110
−105	0.06340	8.919	0.000	8.919	−25.240	0.001	−25.239	−0.06234	0.00000	−0.06234	−203.09	−0.4329	0.03028	−105
−100	0.09772	9.046	0.000	9.046	−24.037	0.001	−24.036	−0.05897	0.00000	−0.05897	−201.23	−0.4277	0.04666	−100
−95	0.1489	9.173	0.000	9.173	−22.835	0.002	−22.833	−0.05565	0.00000	−0.05565	−199.35	−0.4225	0.07111	−95
−90	0.2242	9.300	0.000	9.300	−21.631	0.002	−21.629	−0.05237	0.00001	−0.05236	−197.44	−0.4173	0.1071	−90
−85	0.3341	9.426	0.000	9.426	−20.428	0.003	−20.425	−0.04913	0.00001	−0.04912	−195.51	−0.4121	0.1597	−85
−80	0.4930	9.553	0.000	9.553	−19.225	0.005	−19.220	−0.04595	0.00001	−0.04594	−193.55	−0.4069	0.2356	−80
−75	0.7198	9.680	0.000	9.680	−18.022	0.007	−18.015	−0.04280	0.00002	−0.04278	−191.57	−0.4017	0.3441	−75
−70	1.040	9.806	0.000	9.806	−16.820	0.011	−16.809	−0.03969	0.00003	−0.03966	−189.56	−0.3965	0.4976	−70
−65	1.491	9.933	0.000	9.932	−15.617	0.016	−15.602	−0.03663	0.00005	−0.03658	−187.53	−0.3912	0.7130	−65
−60	2.118	10.059	0.000	10.059	−14.416	0.022	−14.394	−0.03360	0.00006	−0.03354	−185.47	−0.3861	1.0127	−60
−55	2.982	10.186	0.000	10.186	−13.214	0.031	−13.183	−0.03061	0.00009	−0.03053	−183.39	−0.3810	1.4258	−55
−50	4.163	10.313	0.001	10.314	−12.012	0.043	−11.969	−0.02766	0.00012	−0.02754	−181.29	−0.3758	1.9910	−50
−45	4.747	10.364	0.001	10.365	−11.532	0.049	−11.483	−0.02649	0.00013	−0.02636	−180.44	−0.3738	2.2702	−48
−46	5.406	10.414	0.001	10.415	−11.051	0.056	−10.995	−0.02533	0.00014	−0.02518	−179.59	−0.3717	2.5854	−46
−44	6.149	10.465	0.001	10.466	−10.571	0.064	−10.507	−0.02418	0.00016	−0.02400	−178.73	−0.3696	2.9405	−44
−42	6.985	10.516	0.001	10.517	−10.090	0.073	−10.017	−0.02301	0.00019	−0.02282	−177.87	−0.3676	3.3408	−42
−40	7.925	10.566	0.001	10.567	−9.609	0.083	−9.526	−0.02186	0.00021	−0.02165	−177.01	−0.3653	3.7906	−40
−38	8.980	10.617	0.002	10.619	−9.129	0.094	−9.035	−0.02072	0.00024	−0.02048	−176.14	−0.3634	4.2958	−38
−36	10.16	10.668	0.002	10.670	−8.648	0.106	−8.543	−0.01958	0.00028	−0.01933	−175.27	−0.3614	4.8626	−36
−34	11.49	10.718	0.002	10.720	−8.168	0.121	−8.047	−0.01845	0.00030	−0.01815	−174.40	−0.3593	5.4980	−34
−32	12.98	10.769	0.002	10.771	−7.687	0.136	−7.551	−0.01733	0.00034	−0.01699	−173.52	−0.3573	6.2093	−32

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^4$	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/lb h_w	Entropy Btu per (°F)(Lb) s_w	Vap. Press In. Hg $p_s \times 10^3$	Fahr. Temp. t(F)
−30	1.464	10.820	0.002	10.823	−7.207	0.154	−7.053	−0.01631	0.00033	−0.01598	−172.64	−0.3552	0.70046	−30
−28	1.649	10.870	0.003	10.873	−6.726	0.173	−6.553	−0.01509	0.00043	−0.01466	−171.75	−0.3531	0.78925	−28
−26	1.856	10.921	0.003	10.924	−6.246	0.196	−6.050	−0.01398	0.00045	−0.01350	−170.88	−0.3511	0.88838	−26
−24	2.087	10.972	0.004	10.976	−5.765	0.219	−5.546	−0.01287	0.00054	−0.01233	−169.97	−0.3490	0.99885	−24
−22	2.344	11.022	0.004	11.026	−5.285	0.246	−5.039	−0.01177	0.00061	−0.01116	−169.07	−0.3469	1.1219	−22
−20	2.630	11.073	0.005	11.078	−4.804	0.277	−4.527	−0.01067	0.00068	−0.00999	−168.17	−0.3449	1.2587	−20
−18	2.755	11.098	0.005	11.103	−4.564	0.293	−4.271	−0.01012	0.00073	−0.00940	−167.72	−0.3438	1.3327	−19
−18	2.948	11.124	0.005	11.129	−4.324	0.310	−4.014	−0.00958	0.00078	−0.00880	−167.28	−0.3428	1.4107	−18
−17	3.120	11.149	0.006	11.155	−4.083	0.328	−3.755	−0.00904	0.00080	−0.00824	−166.81	−0.3418	1.4929	−17
−16	3.301	11.174	0.006	11.180	−3.843	0.348	−3.495	−0.00850	0.00084	−0.00766	−166.35	−0.3408	1.5795	−16
−15	3.491	11.200	0.006	11.206	−3.603	0.368	−3.235	−0.00796	0.00089	−0.00707	−165.90	−0.3398	1.6706	−15
−14	3.692	11.225	0.007	11.232	−3.363	0.389	−2.974	−0.00743	0.00094	−0.00649	−165.44	−0.3387	1.7666	−14
−13	3.903	11.250	0.007	11.257	−3.123	0.413	−2.711	−0.00689	0.00099	−0.00590	−164.98	−0.3377	1.8677	−13
−12	4.125	11.276	0.008	11.283	−2.883	0.436	−2.446	−0.00636	0.00104	−0.00533	−164.52	−0.3367	1.9740	−12
−11	4.359	11.301	0.008	11.309	−2.642	0.461	−2.181	−0.00583	0.00109	−0.00473	−164.06	−0.3357	2.0859	−11
−10	4.606	11.326	0.008	11.334	−2.402	0.487	−1.915	−0.00529	0.00113	−0.00414	−163.60	−0.3346	2.2035	−10
−9	4.865	11.351	0.008	11.359	−2.162	0.514	−1.648	−0.00475	0.00121	−0.00354	−163.14	−0.3336	2.3272	−9
−8	5.137	11.376	0.009	11.385	−1.923	0.543	−1.379	−0.00422	0.00125	−0.00296	−162.67	−0.3326	2.4573	−8
−7	5.423	11.401	0.010	11.411	−1.681	0.574	−1.107	−0.00369	0.00135	−0.00234	−162.21	−0.3316	2.5940	−7
−6	5.724	11.427	0.010	11.437	−1.441	0.606	−0.835	−0.00316	0.00143	−0.00174	−161.74	−0.3305	2.7377	−6
−5	6.040	11.452	0.011	11.463	−1.201	0.639	−0.563	−0.00263	0.00149	−0.00114	−161.28	−0.3295	2.8886	−5
−4	6.371	11.477	0.012	11.489	−0.961	0.675	−0.286	−0.00210	0.00157	−0.00053	−160.81	−0.3285	3.0472	−4
−3	6.720	11.502	0.013	11.515	−0.721	0.712	−0.009	−0.00157	0.00163	0.00006	−160.34	−0.3275	3.2137	−3
−2	7.085	11.528	0.013	11.541	−0.480	0.751	0.271	−0.00105	0.00174	0.00069	−159.87	−0.3264	3.3885	−2
−1	7.469	11.553	0.014	11.567	−0.240	0.792	0.552	−0.00053	0.00183	0.00131	−159.40	−0.3254	3.5720	−1
0	7.873	11.578	0.015	11.593	0.000	0.835	0.835	0.00000	0.00192	0.00192	−158.93	−0.3244	3.7645	0
1	8.295	11.604	0.015	11.619	0.240	0.880	1.120	0.00053	0.00203	0.00254	−158.46	−0.3234	3.9668	1
2	8.739	11.629	0.016	11.645	0.480	0.928	1.408	0.00104	0.00212	0.00316	−157.99	−0.3223	4.1785	2
3	9.204	11.654	0.017	11.671	0.721	0.977	1.698	0.00156	0.00223	0.00379	−157.52	−0.3213	4.4007	3
4	9.692	11.679	0.018	11.697	0.961	1.030	1.991	0.00208	0.00234	0.00442	−157.04	−0.3203	4.6337	4
5	10.20	11.705	0.019	11.724	1.201	1.085	2.286	0.00260	0.00248	0.00506	−156.57	−0.3193	4.8779	5
6	10.74	11.730	0.020	11.750	1.441	1.142	2.583	0.00313	0.00258	0.00570	−156.09	−0.3182	5.1339	6

[a] Compiled by John A. Goff and S. Gratch.

Table 2 .... Thermodynamic Properties of *MOIST AIR*[a] (Standard Atmospheric Pressure, 29.921 in. Hg) (*Continued*)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^3$	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/Lb h_w	Entropy Btu per (°F)(Lb) s_w	Vap. Press In. Hg $p_s \times 10^2$	Fahr. Temp. t(F)
7	1.130	11.756	0.021	11.777	1.681	1.202	2.883	0.00384	0.00271	0.00635	−155.61	−0.3172	5.4022	7
8	1.189	11.781	0.022	11.803	1.922	1.266	3.188	0.00415	0.00285	0.00700	−155.13	−0.3162	5.6823	8
9	1.251	11.806	0.024	11.830	2.162	1.332	3.494	0.00457	0.00299	0.00756	−154.65	−0.3152	5.9776	9
10	1.315	11.831	0.025	11.856	2.402	1.401	3.803	0.00518	0.00314	0.00832	−154.17	−0.3141	6.2888	10
11	1.383	11.857	0.026	11.883	2.642	1.474	4.116	0.00569	0.00330	0.00899	−153.69	−0.3131	6.6065	11
12	1.454	11.883	0.028	11.910	2.882	1.550	4.432	0.00620	0.00346	0.00966	−153.21	−0.3121	6.9463	12
13	1.528	11.907	0.029	11.936	3.123	1.630	4.753	0.00671	0.00363	0.01034	−152.73	−0.3111	7.2927	13
14	1.606	11.933	0.030	11.963	3.363	1.713	5.076	0.00721	0.00380	0.01101	−152.24	−0.3100	7.6696	14
15	1.687	11.958	0.032	11.990	3.603	1.800	5.403	0.00772	0.00399	0.01171	−151.76	−0.3090	8.0565	15
16	1.773	11.983	0.034	12.017	3.843	1.892	5.735	0.00822	0.00418	0.01240	−151.27	−0.3080	8.4612	16
17	1.861	12.009	0.035	12.044	4.083	1.988	6.071	0.00873	0.00438	0.01311	−150.78	−0.3070	8.8843	17
18	1.953	12.034	0.038	12.072	4.324	2.088	6.412	0.00923	0.00459	0.01383	−150.29	0.3059	9.3257	18
19	2.051	12.059	0.040	12.099	4.564	2.192	6.756	0.00973	0.00481	0.01454	−149.80	−0.3049	9.7859	19
20	2.152	12.084	0.042	12.126	4.804	2.302	7.106	0.01023	0.00504	0.01527	−149.31	−0.3039	10.275	20
21	2.258	12.110	0.044	12.154	5.044	2.416	7.460	0.01073	0.00528	0.01601	−148.82	−0.3029	10.777	21
22	2.369	12.135	0.046	12.181	5.284	2.536	7.820	0.01123	0.00553	0.01676	−148.33	−0.3018	11.305	22
23	2.485	12.160	0.049	12.209	5.525	2.661	8.186	0.01173	0.00579	0.01752	−147.84	−0.3008	11.856	23
24	2.605	12.186	0.051	12.237	5.765	2.792	8.557	0.01223	0.00607	0.01830	−147.34	−0.2998	12.431	24
25	2.733	12.211	0.054	12.265	6.005	2.929	8.934	0.01272	0.00635	0.01908	−146.85	−0.2988	13.032	25
26	2.865	12.236	0.057	12.293	6.245	3.072	9.317	0.01323	0.00665	0.01987	−146.35	−0.2977	13.659	26
27	3.003	12.262	0.059	12.321	6.485	3.221	9.706	0.01373	0.00696	0.02068	−145.85	−0.2967	14.313	27
28	3.147	12.287	0.062	12.349	6.726	3.377	10.103	0.01421	0.00728	0.02149	−145.36	−0.2957	14.995	28
29	3.297	12.313	0.065	12.377	6.966	3.540	10.506	0.01470	0.00761	0.02231	−144.86	−0.2947	15.709	29
30	3.454	12.338	0.068	12.406	7.206	3.709	10.915	0.01519	0.00796	0.02315	−144.36	−0.2936	16.452	30
31	3.617	12.363	0.071	12.434	7.446	3.887	11.333	0.01568	0.00832	0.02400	−143.86	−0.2926	17.227	31
32	3.788	12.388	0.074	12.463	7.686	4.072	11.758	0.01617	0.00870	0.02487	−143.36	−0.2916	18.035	32
32*	3.788	12.388	0.075	12.463	7.686	4.072	11.758	0.01617	0.00870	0.02487	0.04	0.0000	18.037	32*
33	3.944	12.413	0.079	12.492	7.927	4.342	12.269	0.01666	0.00904	0.02570	1.03	0.0030	18.778	33
34	4.107	12.438	0.083	12.520	8.167	4.618	12.785	0.01715	0.00940	0.02655	2.06	0.0061	19.546	34
35	4.275	12.464	0.085	12.549	8.407	4.901	13.308	0.01764	0.00977	0.02741	3.06	0.0091	20.342	35
36	4.450	12.489	0.089	12.578	8.647	4.791	13.438	0.01812	0.01016	0.02828	4.07	0.0081	21.166	36
37	4.631	12.514	0.093	12.607	8.887	4.987	13.874	0.01861	0.01056	0.02917	5.07	0.0102	22.020	37

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^3$	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/Lb h_w	Entropy Btu per (°F)(Lb) s_w	Vap. Press In. Hg p_s	Fahr. Temp. t(F)
38	4.818	12.540	0.097	12.637	9.128	5.191	14.319	0.01909	0.01097	0.03006	6.08	0.0122	0.27204	38
39	5.012	12.565	0.101	12.666	9.368	5.403	14.771	0.01957	0.01139	0.03096	7.08	0.0142	0.33819	39
40	5.213	12.590	0.105	12.695	9.608	5.622	15.230	0.02005	0.01183	0.03188	8.09	0.0162	0.34787	40
41	5.421	12.616	0.109	12.725	9.848	5.849	15.697	0.02053	0.01228	0.03281	9.09	0.0182	0.35748	41
42	5.635	12.641	0.114	12.755	10.088	6.084	16.173	0.02101	0.01275	0.03376	10.09	0.0202	0.36763	42
43	5.850	12.666	0.119	12.785	10.329	6.328	16.657	0.02149	0.01323	0.03472	11.10	0.0222	0.37813	43
44	6.091	12.691	0.124	12.815	10.569	6.580	17.149	0.02197	0.01373	0.03570	12.10	0.0242	0.38899	44
45	6.331	12.717	0.129	12.846	10.809	6.841	17.650	0.02245	0.01425	0.03670	13.10	0.0262	0.40023	45
46	6.578	12.742	0.134	12.876	11.049	7.112	18.161	0.02293	0.01478	0.03771	14.10	0.0282	0.41185	46
47	6.835	12.767	0.140	12.907	11.289	7.391	18.680	0.02340	0.01534	0.03874	15.11	0.0302	0.42386	47
48	7.100	12.792	0.145	12.938	11.530	7.681	19.211	0.02387	0.01591	0.03978	16.11	0.0321	0.43628	48
49	7.374	12.818	0.151	12.969	11.770	7.981	19.751	0.02434	0.01650	0.04084	17.11	0.0341	0.34913	49
50	7.658	12.843	0.158	13.001	12.010	8.291	20.301	0.02481	0.01711	0.04192	18.11	0.0361	0.36240	50
51	7.952	12.868	0.164	13.033	12.250	8.612	20.862	0.02528	0.01774	0.04303	19.11	0.0381	0.37611	51
52	8.256	12.894	0.170	13.064	12.491	8.945	21.436	0.02575	0.01839	0.04414	20.11	0.0400	0.39028	52
53	8.569	12.919	0.178	13.097	12.731	9.289	22.020	0.02622	0.01906	0.04528	21.12	0.0420	0.40492	53
54	8.894	12.944	0.185	13.129	12.971	9.644	22.615	0.02669	0.01976	0.04645	22.12	0.0439	0.42004	54
55	9.229	12.970	0.192	13.162	13.211	10.01	23.22	0.02716	0.02047	0.04763	23.12	0.0459	0.43565	55
56	9.575	12.995	0.200	13.195	13.452	10.39	23.84	0.02762	0.02121	0.04883	24.12	0.0478	0.45178	56
57	9.934	13.020	0.208	13.228	13.692	10.79	24.48	0.02809	0.02197	0.05006	25.12	0.0497	0.46840	57
58	10.30	13.045	0.216	13.261	13.932	11.19	25.12	0.02855	0.02276	0.05131	26.12	0.0517	0.48558	58
59	10.69	13.071	0.224	13.295	14.173	11.61	25.78	0.02903	0.02357	0.05259	27.13	0.0536	0.50330	59
60	11.08	13.096	0.233	13.329	14.413	12.05	26.46	0.02948	0.02441	0.05389	28.13	0.0555	0.52159	60
61	11.49	13.121	0.243	13.363	14.653	12.50	27.15	0.02994	0.02527	0.05521	29.13	0.0574	0.54047	61
62	11.91	13.147	0.251	13.398	14.893	12.96	27.85	0.03040	0.02616	0.05656	30.13	0.0594	0.55994	62
63	12.35	13.173	0.261	13.433	15.134	13.44	28.57	0.03086	0.02708	0.05794	31.12	0.0613	0.58002	63
64	12.80	13.197	0.271	13.468	15.374	13.94	29.31	0.03133	0.02803	0.05935	32.12	0.0633	0.60073	64
65	13.28	13.223	0.282	13.504	15.614	14.45	30.06	0.03177	0.02901	0.06078	33.11	0.0651	0.62209	65
66	13.74	13.247	0.292	13.539	15.855	14.98	30.83	0.03223	0.03002	0.06225	34.11	0.0670	0.64411	66
67	14.24	13.273	0.303	13.576	16.095	15.53	31.62	0.03269	0.03106	0.06375	35.11	0.0689	0.66681	67
68	14.75	13.298	0.315	13.613	16.335	16.09	32.43	0.03314	0.03213	0.06527	36.11	0.0708	0.69019	68
69	15.28	13.323	0.327	13.650	16.576	16.67	33.25	0.03360	0.03323	0.06683	37.11	0.0727	0.71430	69

[a] Compiled by John A. Goff and S. Gratch.

* Extrapolated to represent metastable equilibrium with undercooled liquid.

Table 2 .... Thermodynamic Properties of *MOIST AIR*[a] (Standard Atmospheric Pressure, 29.921 in. Hg) (*Continued*)

Fahr. Temp. t(F)	Humidity Ratio W, × 10³	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu per (°F)(Lb) s_w	Vap. Press In. Hg p_s	
70	1.582	13.348	0.339	13.687	16.816	17.27	34.09	0.03405	0.03437	0.06842	38.11	0.0766	0.73915	70
71	1.639	13.373	0.351	13.724	17.056	17.89	34.95	0.03450	0.03554	0.07004	39.11	0.0785	0.76475	71
72	1.697	13.398	0.364	13.762	17.297	18.53	35.83	0.03495	0.03675	0.07170	40.11	0.0804	0.79113	72
73	1.757	13.424	0.377	13.801	17.537	19.20	36.74	0.03540	0.03800	0.07340	41.11	0.0823	0.81828	73
74	1.819	13.449	0.392	13.841	17.778	19.88	37.66	0.03585	0.03928	0.07513	42.10	0.0821	0.84624	74
75	1.882	13.474	0.407	13.881	18.018	20.59	38.61	0.03630	0.04060	0.07690	43.10	0.0840	0.87504	75
76	1.948	13.499	0.422	13.921	18.259	21.31	39.57	0.03675	0.04197	0.07872	44.10	0.0859	0.90470	76
77	2.016	13.525	0.437	13.962	18.499	22.07	40.57	0.03720	0.04337	0.08057	45.10	0.0877	0.93523	77
78	2.086	13.550	0.453	14.003	18.740	22.84	41.58	0.03765	0.04482	0.08247	46.10	0.0896	0.96663	78
79	2.158	13.575	0.470	14.045	18.980	23.64	42.62	0.03810	0.04631	0.08441	47.10	0.0914	0.99899	79
80	2.233	13.601	0.486	14.087	19.221	24.47	43.69	0.03854	0.04784	0.08638	48.10	0.0933	1.0323	80
81	2.310	13.626	0.504	14.130	19.461	25.32	44.78	0.03899	0.04942	0.08841	49.09	0.0952	1.0665	81
82	2.389	13.651	0.523	14.174	19.702	26.20	45.90	0.03943	0.05105	0.09048	50.09	0.0970	1.1017	82
83	2.471	13.676	0.542	14.218	19.942	27.10	47.04	0.03987	0.05273	0.09260	51.09	0.0989	1.1379	83
84	2.556	13.702	0.560	14.262	20.183	28.04	48.22	0.04031	0.05446	0.09477	52.09	0.1007	1.1752	84
85	2.642	13.727	0.581	14.308	20.423	29.01	49.43	0.04075	0.05624	0.09699	53.09	0.1025	1.2135	85
86	2.731	13.752	0.602	14.354	20.663	30.00	50.66	0.04119	0.05807	0.09926	54.08	0.1043	1.2529	86
87	2.824	13.777	0.624	14.401	20.904	31.03	51.93	0.04163	0.05995	0.10158	55.08	0.1062	1.2934	87
88	2.919	13.803	0.645	14.448	21.144	32.09	53.23	0.04207	0.06189	0.10396	56.08	0.1080	1.3351	88
89	3.017	13.828	0.668	14.496	21.385	33.18	54.56	0.04251	0.06389	0.10640	57.08	0.1098	1.3779	89
90	3.118	13.853	0.692	14.545	21.625	34.31	55.93	0.04295	0.06596	0.10890	58.08	0.1116	1.4219	90
91	3.223	13.879	0.716	14.595	21.865	35.47	57.33	0.04339	0.06807	0.11146	59.07	0.1135	1.4671	91
92	3.330	13.904	0.741	14.645	22.106	36.67	58.78	0.04382	0.07025	0.11407	60.07	0.1153	1.5135	92
93	3.441	13.929	0.768	14.697	22.346	37.90	60.25	0.04426	0.07249	0.11675	61.07	0.1171	1.5612	93
94	3.556	13.954	0.795	14.749	23.587	39.18	61.77	0.04469	0.07480	0.11949	62.07	0.1188	1.6102	94
95	3.673	13.980	0.822	14.802	23.827	40.49	63.32	0.04513	0.07718	0.12231	63.07	0.1206	1.6606	95
96	3.795	14.005	0.851	14.856	23.068	41.85	64.92	0.04556	0.07963	0.12519	64.06	0.1224	1.7123	96
97	3.920	14.030	0.881	14.911	23.308	43.24	66.55	0.04600	0.08215	0.12815	65.06	0.1242	1.7654	97
98	4.049	14.056	0.911	14.967	23.548	44.68	68.23	0.04643	0.08474	0.13117	66.06	0.1260	1.8199	98
99	4.182	14.081	0.942	15.023	23.789	46.17	69.96	0.04688	0.08741	0.13427	67.06	0.1278	1.8759	99
100	4.319	14.106	0.975	15.081	24.029	47.70	71.73	0.04729	0.09016	0.13745	68.06	0.1296	1.9333	100
101	4.460	14.131	1.009	15.140	24.270	49.28	73.55	0.04772	0.09299	0.14071	69.05	0.1314	1.9923	101
102	4.606	14.157	1.043	15.200	24.510	50.91	75.42	0.04815	0.09591	0.14406	70.05	0.1332	2.0528	102
103	4.756	14.182	1.079	15.261	24.751	53.59	77.34	0.04858	0.09891	0.14749	71.05	0.1350	2.1149	103
104	4.911	14.207	1.117	15.324	24.991	54.33	79.31	0.04900	0.1020	0.1510	72.05	0.1367	2.1786	104

Fahr. Temp. t(F)	Humidity Ratio W, × 10	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu per (°F)(Lb) s_w	Vap. Press In. Hg p_s	
105	0.5070	14.232	1.155	15.387	25.232	56.11	81.34	0.04943	0.1052	0.1546	73.04	0.1385	2.2439	105
106	0.5234	14.258	1.194	15.452	25.473	57.95	83.43	0.04985	0.1085	0.1584	74.04	0.1403	2.3109	106
107	0.5404	14.283	1.235	15.518	25.713	59.85	85.56	0.05028	0.1119	0.1621	75.04	0.1421	2.3797	107
108	0.5578	14.308	1.278	15.586	25.953	61.80	87.76	0.05070	0.1153	0.1660	76.04	0.1438	2.4502	108
109	0.5758	14.333	1.321	15.654	26.194	63.82	90.03	0.05113	0.1189	0.1700	77.04	0.1456	2.5225	109
110	0.5944	14.359	1.365	15.724	26.434	65.91	92.34	0.05155	0.1228	0.1742	78.03	0.1474	2.5966	110
111	0.6135	14.384	1.412	15.796	26.675	68.05	94.72	0.05197	0.1264	0.1784	79.03	0.1491	2.6726	111
112	0.6332	14.409	1.460	15.869	26.915	70.27	97.18	0.05239	0.1302	0.1826	80.03	0.1509	2.7505	112
113	0.6536	14.435	1.509	15.944	27.156	72.55	99.71	0.05281	0.1342	0.1870	81.03	0.1525	2.8304	113
114	0.6746	14.460	1.560	16.020	27.397	74.91	102.31	0.05323	0.1384	0.1916	82.03	0.1543	2.9123	114
115	0.6962	14.485	1.613	16.098	27.637	77.34	104.98	0.05365	0.1426	0.1962	83.02	0.1560	2.9963	115
116	0.7185	14.510	1.668	16.178	27.878	79.85	107.73	0.05407	0.1470	0.2011	84.02	0.1577	3.0821	116
117	0.7415	14.536	1.723	16.259	28.119	82.43	110.55	0.05449	0.1515	0.2060	85.02	0.1595	3.1701	117
118	0.7652	14.561	1.782	16.343	28.359	85.10	113.46	0.05490	0.1563	0.2111	86.02	0.1612	3.2603	118
119	0.7897	14.586	1.843	16.428	28.600	87.86	116.46	0.05532	0.1610	0.2163	87.02	0.1629	3.3527	119
120	0.8149	14.611	1.905	16.516	28.841	90.70	119.54	0.05573	0.1659	0.2216	88.01	0.1646	3.4474	120
121	0.8410	14.637	1.968	16.605	29.082	93.64	122.72	0.05615	0.1710	0.2272	89.01	0.1664	3.5443	121
122	0.8678	14.662	2.034	16.696	29.322	96.66	123.98	0.05656	0.1763	0.2332	90.01	0.1681	3.6436	122
123	0.8955	14.687	2.103	16.790	29.563	99.79	129.35	0.05698	0.1817	0.2387	91.01	0.1698	3.7452	123
124	0.9242	14.713	2.174	16.886	29.804	103.0	132.8	0.05739	0.1872	0.2445	92.01	0.1715	3.8493	124
125	0.9537	14.738	2.247	16.985	30.044	106.4	136.4	0.05780	0.1930	0.2508	93.01	0.1732	3.9558	125
126	0.9841	14.763	2.323	17.086	30.285	109.8	140.1	0.05821	0.1989	0.2571	94.01	0.1749	4.0649	126
127	1.016	14.788	2.401	17.189	30.525	113.4	143.9	0.05862	0.2050	0.2636	95.00	0.1766	4.1765	127
128	1.048	14.813	2.482	17.295	30.766	117.0	147.8	0.05903	0.2113	0.2703	96.00	0.1782	4.2907	128
129	1.082	14.839	2.565	17.404	31.007	120.8	151.8	0.05944	0.2178	0.2772	97.00	0.1800	4.4076	129
130	1.116	14.864	2.652	17.516	31.248	124.7	155.9	0.05985	0.2245	0.2844	98.00	0.1817	4.5272	130
131	1.152	14.889	2.742	17.631	31.488	128.6	160.1	0.06026	0.2314	0.2917	99.00	0.1834	4.6495	131
132	1.189	14.916	2.834	17.749	31.729	133.0	164.7	0.06067	0.2386	0.2993	100.00	0.1851	4.7747	132
133	1.227	14.940	2.930	17.870	31.970	137.3	169.3	0.06108	0.2459	0.3070	101.00	0.1868	4.9028	133
134	1.267	14.965	3.029	17.994	32.211	141.8	174.0	0.06148	0.2536	0.3151	102.00	0.1885	5.0337	134

[a] Compiled by John A. Goff and S. Gratch.

Table 2 .... Thermodynamic Properties of *MOIST AIR*[a] (Standard Atmospheric Pressure, 29.921 in. Hg) (*Concluded*)

Fahr. Temp. t(F)	Humidity Ratio W_s	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu per (°F)(Lb) s_w	Vap. Press In. Hg p_s	
135	0.1308	14.990	2.132	18.122	32.452	146.4	178.9	0.06189	0.2614	0.3233	103.00	0.1902	5.1676	135
136	0.1350	15.016	2.227	18.243	32.692	151.3	183.9	0.06229	0.2693	0.3318	104.00	0.1918	5.3046	136
137	0.1393	15.041	2.328	18.369	32.933	156.1	189.0	0.06270	0.2778	0.3405	105.00	0.1935	5.4446	137
138	0.1439	15.066	2.462	18.528	33.174	161.3	194.4	0.06310	0.2865	0.3496	106.00	0.1952	5.5878	138
139	0.1485	15.091	2.580	18.671	33.414	166.5	199.9	0.06350	0.2954	0.3589	107.00	0.1969	5.7342	139
140	0.1534	15.117	2.702	18.819	33.655	172.0	205.7	0.06390	0.3047	0.3686	107.99	0.1985	5.8838	140
141	0.1584	15.142	2.829	18.971	33.896	177.7	211.6	0.06430	0.3142	0.3785	108.99	0.2002	6.0367	141
142	0.1636	15.167	3.061	19.128	34.136	183.6	217.7	0.06470	0.3241	0.3888	109.99	0.2018	6.1930	142
143	0.1689	15.192	4.098	19.290	34.377	189.7	224.1	0.06510	0.3343	0.3994	110.99	0.2035	6.3527	143
144	0.1745	15.218	4.339	19.457	34.618	196.0	230.6	0.06549	0.3449	0.4104	111.99	0.2051	6.5160	144
145	0.1803	15.243	4.586	19.629	34.859	202.5	237.4	0.06589	0.3559	0.4218	113.99	0.2068	6.6828	145
146	0.1862	15.268	4.839	19.807	35.099	209.3	244.4	0.06629	0.3672	0.4335	113.99	0.2084	6.8532	146
147	0.1924	15.293	4.868	19.991	35.340	216.4	251.7	0.06668	0.3790	0.4457	114.99	0.2101	7.0273	147
148	0.1989	15.319	4.863	20.181	35.581	223.7	259.3	0.06708	0.3911	0.4583	115.99	0.2117	7.2051	148
149	0.2055	15.344	5.033	20.377	35.822	231.3	267.1	0.06748	0.4038	0.4713	116.99	0.2134	7.3867	149
150	0.2125	15.369	5.211	20.580	36.063	239.2	275.3	0.06787	0.4169	0.4848	117.99	0.2150	7.5722	150
151	0.2197	15.394	5.396	20.790	36.304	247.5	283.8	0.06827	0.4304	0.4987	118.99	0.2167	7.7616	151
152	0.2271	15.420	5.587	21.007	36.545	255.9	292.4	0.06866	0.4445	0.5132	119.99	0.2183	7.9550	152
153	0.2349	15.445	5.788	21.233	36.785	264.7	301.5	0.06906	0.4591	0.5282	120.99	0.2200	8.1525	153
154	0.2430	15.470	5.998	21.468	37.026	273.9	310.9	0.06945	0.4743	0.5438	121.99	0.2216	8.3541	154
155	0.2514	15.496	6.218	21.709	37.267	283.6	320.8	0.06984	0.4901	0.5598	122.99	0.2232	8.5599	155
156	0.2602	15.521	6.439	21.960	37.508	293.5	331.0	0.07023	0.5066	0.5765	123.99	0.2248	8.7701	156
157	0.2693	15.548	6.675	22.221	37.749	303.9	341.7	0.07062	0.5237	0.5943	124.99	0.2265	8.9846	157
158	0.2788	15.571	6.922	22.493	37.990	314.7	352.7	0.07101	0.5415	0.6125	125.99	0.2281	9.2036	158
159	0.2887	15.597	7.178	22.775	38.231	326.0	364.2	0.07140	0.5600	0.6314	127.00	0.2297	9.4271	159
160	0.2990	15.622	7.446	23.068	38.472	337.8	376.3	0.07179	0.5793	0.6511	128.00	0.2313	9.6556	160
161	0.3098	15.647	7.727	23.374	38.713	350.1	388.8	0.07218	0.5994	0.6716	129.00	0.2329	9.8876	161
162	0.3211	15.672	8.020	23.692	38.954	363.0	402.0	0.07257	0.6204	0.6930	130.00	0.2345	10.125	162
163	0.3329	15.698	8.326	24.024	39.195	376.5	415.7	0.07296	0.6423	0.7153	131.00	0.2361	10.367	163
164	0.3452	15.723	8.648	24.371	39.436	390.5	429.9	0.07334	0.6652	0.7385	132.00	0.2377	10.614	164
165	0.3581	15.748	8.985	24.733	39.677	405.3	445.0	0.07373	0.6892	0.7629	133.00	0.2393	10.866	165
166	0.3716	15.773	9.339	25.112	39.918	420.8	460.7	0.07411	0.7143	0.7883	134.00	0.2409	11.123	166
167	0.3858	15.799	9.708	25.507	40.159	437.0	477.2	0.07450	0.7405	0.8150	135.01	0.2426	11.385	167
168	0.4007	15.824	10.098	25.922	40.400	454.0	494.4	0.07488	0.7680	0.8429	135.01	0.2441	11.653	168
169	0.4163	15.849	10.508	26.357	40.641	471.8	512.4	0.07527	0.7969	0.8722	137.01	0.2457	11.925	169

Fahr. Temp. t(F)	Humidity Ratio W_s	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. (Ft)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Enthropy Btu/(LB) (°F) s_w	Vap. Press In. Hg p_s	
170	0.4327	15.874	10.935	26.812	40.882	490.6	531.5	0.07565	0.8273	0.9030	133.01	0.2473	12.203	170
171	0.4500	15.900	11.391	27.291	41.123	510.4	551.5	0.07603	0.8593	0.9353	139.01	0.2489	12.486	171
172	0.4682	15.925	11.870	27.795	41.364	531.3	572.7	0.07641	0.8927	0.9691	140.01	0.2505	12.775	172
173	0.4873	15.950	12.376	28.326	41.604	553.3	594.9	0.07680	0.9281	1.0049	141.01	0.2521	13.069	173
174	0.5078	15.975	13.911	28.886	41.845	576.5	618.3	0.07718	0.9654	1.0428	142.02	0.2537	13.369	174
175	0.5292	16.001	13.476	29.476	42.087	601.1	643.2	0.07756	1.005	1.083	142.02	0.2553	13.675	175
176	0.5519	16.026	14.074	30.100	42.328	627.1	669.4	0.07794	1.047	1.125	144.02	0.2568	13.987	176
177	0.5760	16.051	14.710	30.761	42.569	654.7	697.3	0.07832	1.091	1.169	145.02	0.2584	14.304	177
178	0.6016	16.076	15.388	31.463	42.810	684.1	726.9	0.07870	1.137	1.216	146.03	0.2600	14.628	178
179	0.6288	16.102	16.104	32.206	43.051	715.2	758.3	0.07908	1.187	1.266	147.03	0.2616	14.958	179
180	0.6578	16.127	16.870	32.997	43.292	748.5	791.8	0.07946	1.240	1.319	148.03	0.2631	15.294	180
181	0.6887	16.152	17.689	33.841	43.534	783.9	827.4	0.07984	1.298	1.378	149.03	0.2647	15.636	181
182	0.7218	16.177	18.563	34.740	43.775	821.9	865.7	0.08021	1.361	1.437	150.04	0.2662	15.985	182
183	0.7572	16.203	19.504	35.707	44.016	862.5	906.5	0.08059	1.421	1.502	151.04	0.2678	16.340	183
184	0.7953	16.228	20.513	36.741	44.257	906.2	950.5	0.08096	1.490	1.571	153.04	0.2693	16.702	184
185	0.8383	16.253	21.601	37.854	44.498	953.2	997.7	0.08134	1.565	1.646	153.05	0.2709	17.071	185
186	0.8805	16.278	22.775	39.053	44.740	1004	1049	0.08171	1.645	1.727	154.05	0.2724	17.448	186
187	0.9283	16.304	24.047	40.351	44.981	1059	1104	0.08208	1.731	1.813	155.05	0.2740	17.833	187
188	0.9809	16.329	25.427	41.756	45.222	1119	1164	0.08245	1.825	1.907	156.06	0.2755	18.217	188
189	1.037	16.354	26.934	43.288	45.463	1184	1229	0.08283	1.928	2.011	157.06	0.2771	18.614	189
190	1.099	16.379	28.580	44.959	45.704	1255	1301	0.08320	2.039	2.122	158.07	0.2786	19.017	190
191	1.166	16.405	30.385	46.790	45.946	1332	1378	0.08357	2.161	2.245	159.07	0.2802	19.427	191
192	1.241	16.430	32.375	48.805	46.187	1418	1464	0.08394	2.296	2.380	160.07	0.2817	19.845	192
193	1.324	16.455	34.581	51.036	46.428	1513	1559	0.08431	2.446	2.528	161.08	0.2833	20.271	193
194	1.416	16.480	37.028	53.518	46.670	1619	1666	0.08468	2.609	2.694	162.08	0.2848	20.704	194
195	1.519	16.506	39.785	56.291	46.911	1737	1784	0.08505	2.794	2.879	163.09	0.2864	21.145	195
196	1.633	16.531	42.885	59.416	47.153	1871	1918	0.08543	3.002	3.087	164.09	0.2879	21.594	196
197	1.757	16.556	46.402	62.958	47.394	2023	2069	0.08579	3.235	3.324	165.10	0.2895	22.050	197
198	1.917	16.581	50.426	67.007	47.636	2195	2243	0.08616	3.507	3.593	166.10	0.2910	22.514	198
199	2.091	16.607	55.074	71.681	47.877	2395	2443	0.08653	3.817	3.904	167.11	0.2925	22.987	199
200	2.293	16.632	60.510	77.143	48.119	2629	2677	0.08689	4.179	4.266	168.11	0.2940	23.468	200

[a] Compiled by John A. Goff and S. Gratch.

Table 3 .... Thermodynamic Properties of WATER at Saturation[a]

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^2$ Lb/Sq In.	Absolute Pressure $p_s \times 10^2$ In. Hg	Specific Volume, cu ft per lb Sat. Solid v_i	Specific Volume, cu ft per lb Evap. $v_{ie} \times 10^{-4}$	Specific Volume, cu ft per lb Sat. Vapor $v_g \times 10^{-4}$	Enthalpy, Btu per lb h_i	Enthalpy, Btu per lb Evap. h_{ie}	Enthalpy, Btu per lb Sat. Vapor h_g	Entropy, Btu per (Lb) (°F) Sat. Solid s_i	Entropy, Btu per (Lb) (°F) Evap. s_{ie}	Entropy, Btu per (Lb) (°F) Sat. Vapor s_g	Fahr. Temp. t(F)
−160	0.004949	0.01008	0.01722	36070	36070	−222.05	1312.43	990.38	−0.4907	4.0458	3.5549	−160
−155	0.009040	0.01840	0.01723	20080	20080	−220.44	1313.02	992.58	−0.4854	3.9512	3.4658	−155
−150	0.01630	0.03298	0.01723	11390	11390	−218.82	1313.62	994.80	−0.4801	3.8588	3.4787	−150
−145	0.02850	0.05803	0.01724	6577	6577	−217.17	1314.17	997.00	−0.4745	3.8583	3.3835	−145
−140	0.04928	0.1003	0.01724	3864	3864	−215.49	1314.70	999.21	−0.4695	3.7996	3.3301	−140
−135	0.08380	0.1706	0.01725	2306	2306	−213.80	1315.22	1001.43	−0.4643	3.7438	3.2795	−135
−130	0.1403	0.2856	0.01725	1400	1400	−212.08	1315.71	1003.63	−0.4590	3.6874	3.2284	−130
−125	0.2312	0.4708	0.01726	862.2	862.2	−210.34	1316.18	1005.84	−0.4538	3.6338	3.1800	−125
−120	0.3767	0.7669	0.01726	538.6	538.6	−208.58	1316.63	1008.05	−0.4485	3.5815	3.1330	−120
−115	0.6019	1.226	0.01727	341.1	341.1	−206.79	1317.05	1010.26	−0.4433	3.5308	3.0875	−115
−110	0.9517	1.938	0.01728	218.9	218.9	−204.98	1317.45	1012.47	−0.4381	3.4815	3.0434	−110
−105	1.486	3.025	0.01728	143.2	143.2	−203.14	1317.83	1014.68	−0.4329	3.4335	3.0006	−105
−100	2.291	4.664	0.01729	93.53	93.53	−201.28	1318.17	1016.89	−0.4277	3.3868	2.9591	−100
−95	3.491	7.108	0.01729	63.23	63.23	−199.40	1318.50	1019.10	−0.4225	3.3412	2.9187	−95
−90	5.260	10.71	0.01730	41.86	41.86	−197.49	1318.80	1021.31	−0.4173	3.2969	2.8796	−90
−85	7.841	15.96	0.01730	28.46	28.46	−195.56	1319.08	1023.52	−0.4121	3.2538	2.8415	−85
−80	11.57	23.55	0.01731	19.55	19.55	−193.60	1319.33	1025.73	−0.4069	3.2114	2.8045	−80
−75	16.89	34.39	0.01732	13.56	13.56	−191.63	1319.56	1027.94	−0.4017	3.1702	2.7685	−75
−70	24.43	49.74	0.01732	9.501	9.501	−189.61	1319.76	1030.15	−0.3965	3.1301	2.7336	−70
−65	35.01	71.28	0.01733	6.715	6.715	−187.56	1319.94	1032.36	−0.3914	3.0910	2.6996	−65
−60	49.72	101.2	0.01734	4.788	4.788	−185.52	1320.10	1034.58	−0.3863	3.0525	2.6664	−60
−55	70.01	142.6	0.01734	3.443	3.443	−183.44	1320.23	1036.79	−0.3810	3.0152	2.6342	−55
−50	97.76	199.0	0.01735	2.496	2.496	−181.34	1320.34	1039.00	−0.3758	2.9786	2.6028	−50
−45	111.5	227.0	0.01736	2.200	2.200	−180.49	1320.37	1039.88	−0.3738	2.9643	2.5905	−45
−46	127.0	258.5	0.01736	1.941	1.941	−178.64	1320.40	1040.76	−0.3717	2.9501	2.5784	−46
−44	144.4	294.0	0.01736	1.715	1.715	−178.78	1320.43	1041.65	−0.3696	2.9359	2.5663	−44
−42	164.1	334.0	0.01736	1.516	1.516	−177.92	1320.45	1042.53	−0.3676	2.9219	2.5543	−42
−40	186.1	379.0	0.01737	1.343	1.343	−177.06	1320.48	1043.42	−0.3655	2.9080	2.5425	−40
−38	211.0	429.5	0.01737	1.191	1.191	−176.19	1320.49	1044.30	−0.3634	2.8942	2.5308	−38
−36	238.8	486.2	0.01737	1.057	1.057	−175.32	1320.51	1045.19	−0.3614	2.8807	2.5193	−36
−34	270.0	549.7	0.01737	0.9391	0.9391	−174.45	1320.52	1046.07	−0.3593	2.8671	2.5078	−34
−32	304.9	620.8	0.01738	0.8355	0.8355	−173.57	1320.53	1046.95	−0.3573	2.8538	2.4965	−32

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^2$ Lb/Sq In.	Absolute Pressure $p_s \times 10^2$ In. Hg	Specific Volume, cu ft per lb Sat. Solid v_i	Specific Volume, cu ft per lb Evap. $v_{ie} \times 10^{-4}$	Specific Volume, cu ft per lb Sat. Vapor $v_g \times 10^{-4}$	Enthalpy, Btu per lb Sat. Solid h_i	Enthalpy, Btu per lb Evap. h_{ie}	Enthalpy, Btu per lb Sat. Vapor h_g	Entropy, Btu per (Lb) (°F) Sat. Solid s_i	Entropy, Btu per (Lb) (°F) Evap. s_{ie}	Entropy, Btu per (Lb) (°F) Sat. Vapor s_g	Fahr. Temp. t(F)
−30	0.3440	0.7003	0.01738	7.441	7.441	−172.68	1320.52	1047.84	−0.3552	2.8405	2.4853	−30
−28	0.3876	0.7891	0.01738	6.634	6.634	−171.80	1320.53	1048.72	−0.3532	2.8274	2.4742	−28
−26	0.4363	0.8882	0.01738	5.921	5.921	−170.91	1320.51	1049.60	−0.3511	2.8143	2.4632	−26
−24	0.4905	0.9987	0.01739	5.290	5.290	−170.01	1320.50	1050.49	−0.3490	2.8013	2.4523	−24
−22	0.5509	1.122	0.01739	4.732	4.732	−169.13	1320.49	1051.37	−0.3470	2.7885	2.4415	−22
−20	0.6181	1.259	0.01739	4.237	4.237	−168.21	1320.47	1052.26	−0.3449	2.7757	2.4308	−20
−18	0.6545	1.333	0.01739	4.011	4.011	−167.76	1320.46	1052.70	−0.3439	2.7695	2.4256	−18
−18	0.6928	1.410	0.01740	3.797	3.797	−167.31	1320.45	1053.14	−0.3429	2.7632	2.4203	−18
−17	0.7323	1.493	0.01740	3.596	3.596	−166.85	1320.43	1053.58	−0.3418	2.7568	2.4150	−17
−16	0.7757	1.578	0.01740	3.407	3.407	−166.40	1320.42	1054.02	−0.3408	2.7506	2.4098	−16
−15	0.8204	1.670	0.01740	3.228	3.228	−165.94	1320.41	1054.47	−0.3398	2.7444	2.4046	−15
−14	0.8676	1.766	0.01740	3.060	3.060	−165.48	1320.39	1054.91	−0.3388	2.7383	2.3995	−14
−13	0.9172	1.867	0.01740	2.901	2.901	−165.03	1320.38	1055.35	−0.3377	2.7320	2.3943	−13
−12	0.9694	1.974	0.01740	2.750	2.750	−164.57	1320.36	1055.79	−0.3367	2.7259	2.3892	−12
−11	1.024	2.086	0.01740	2.609	2.609	−164.11	1320.34	1056.23	−0.3357	2.7198	2.3841	−11
−10	1.082	2.203	0.01741	2.475	2.475	−163.65	1320.32	1056.67	−0.3347	2.7138	2.3791	−10
−9	1.143	2.327	0.01741	2.349	2.349	−163.18	1320.30	1057.13	−0.3328	2.7078	2.3740	−9
−8	1.207	2.457	0.01741	2.229	2.229	−162.72	1320.28	1057.56	−0.3326	2.7016	2.3690	−8
−7	1.274	2.594	0.01741	2.116	2.116	−162.26	1320.26	1058.00	−0.3316	2.6956	2.3640	−7
−6	1.344	2.737	0.01741	2.010	2.010	−161.79	1320.23	1058.44	−0.3306	2.6896	2.3590	−6
−5	1.419	2.888	0.01741	1.909	1.909	−161.33	1320.21	1058.88	−0.3295	2.6836	2.3541	−5
−4	1.496	3.047	0.01742	1.814	1.814	−160.86	1320.18	1059.32	−0.3285	2.6777	2.3492	−4
−3	1.578	3.213	0.01742	1.723	1.723	−160.39	1320.15	1059.76	−0.3275	2.6718	2.3443	−3
−2	1.664	3.388	0.01742	1.638	1.638	−159.92	1320.13	1060.21	−0.3264	2.6658	2.3394	−2
−1	1.754	3.572	0.01742	1.557	1.557	−159.45	1320.10	1060.65	−0.3254	2.6600	2.3346	−1
0	1.849	3.764	0.01743	1.481	1.481	−158.98	1320.07	1061.09	−0.3244	2.6541	2.3297	0
1	1.948	3.966	0.01743	1.408	1.408	−158.51	1320.04	1061.53	−0.3234	2.6483	2.3249	1
2	2.052	4.178	0.01743	1.340	1.340	−158.04	1320.01	1061.97	−0.3224	2.6425	2.3201	2
3	2.161	4.400	0.01743	1.275	1.257	−157.56	1319.97	1062.41	−0.3213	2.6367	2.3154	3
4	2.276	4.633	0.01743	1.214	1.214	−157.09	1319.94	1062.85	−0.3203	2.6309	2.3106	4
5	2.396	4.878	0.01743	1.155	1.155	−156.61	1319.90	1063.29	−0.3193	2.6253	2.3059	5
6	2.521	5.134	0.01743	1.100	1.100	−156.14	1319.88	1063.74	−0.3183	2.6194	2.3012	6

[a] Compiled by John A. Goff and S. Gratch.

Table 3 .... Thermodynamic Properties of WATER at Saturation[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s Lb/Sq In.	Absolute Pressure p_s In. Hg	Specific Volume, cu ft per lb Sat. Solid v_i	Specific Volume, cu ft per lb Evap. v_{ie}	Specific Volume, cu ft per lb Sat. Vapor v_g	Enthalpy, Btu per lb Sat. Solid h_i	Enthalpy, Btu per lb Evap. h_{ie}	Enthalpy, Btu per lb Sat. Vapor h_g	Entropy, Btu per (Lb) (°F) Sat. Solid s_i	Entropy, Btu per (Lb) (°F) Evap. s_{ie}	Entropy, Btu per (Lb) (°F) Sat. Vapor s_g	Fahr. Temp. t(F)
7	0.02653	0.05402	0.01743	10480	10480	−155.66	1319.84	1064.18	−0.3173	2.6138	2.2966	7
8	0.02791	0.05683	0.01743	9979	9979	−155.18	1319.80	1064.62	−0.3163	2.6081	2.2918	8
9	0.02936	0.05977	0.01744	9507	9507	−154.70	1319.76	1065.06	−0.3152	2.6023	2.2872	9
10	0.03087	0.06286	0.01744	9060	9060	−154.23	1319.72	1065.50	−0.3142	2.5969	2.2827	10
11	0.03246	0.06608	0.01744	8636	8636	−153.74	1319.68	1065.94	−0.3131	2.5913	2.2781	11
12	0.03412	0.06948	0.01744	8234	8234	−153.26	1319.64	1066.38	−0.3121	2.5857	2.2736	12
13	0.03585	0.07300	0.01744	7851	7851	−152.77	1319.59	1066.82	−0.3111	2.5801	2.2690	13
14	0.03767	0.07669	0.01744	7489	7489	−152.29	1319.55	1067.26	−0.3101	2.5748	2.2645	14
15	0.03957	0.08056	0.01744	7144	7144	−151.80	1319.50	1067.70	−0.3090	2.5690	2.2600	15
16	0.04156	0.08461	0.01745	6817	6817	−151.33	1319.46	1068.14	−0.3080	2.5635	2.2555	16
17	0.04363	0.08884	0.01745	6505	6505	−150.83	1319.41	1068.58	−0.3070	2.5581	2.2511	17
18	0.04581	0.09326	0.01745	6210	6210	−150.34	1319.36	1069.02	−0.3060	2.5526	2.2466	18
19	0.04808	0.09789	0.01745	5929	5929	−149.85	1319.31	1069.46	−0.3049	2.5471	2.2422	19
20	0.05045	0.1027	0.01745	5663	5663	−149.36	1319.26	1069.90	−0.3039	2.5417	2.2378	20
21	0.05293	0.1078	0.01745	5408	5408	−148.87	1319.21	1070.34	−0.3029	2.5364	2.2335	21
22	0.05552	0.1130	0.01748	5166	5166	−148.38	1319.16	1070.78	−0.3019	2.5310	2.2291	22
23	0.05823	0.1186	0.01748	4936	4936	−147.88	1319.10	1071.22	−0.3008	2.5256	2.2248	23
24	0.06105	0.1243	0.01748	4717	4717	−147.39	1319.05	1071.66	−0.2998	2.5203	2.2205	24
25	0.06400	0.1303	0.01748	4509	4509	−146.89	1318.98	1072.09	−0.2988	2.5150	2.2163	25
26	0.06708	0.1366	0.01748	4311	4311	−146.40	1318.93	1072.53	−0.2978	2.5097	2.2119	26
27	0.07030	0.1431	0.01745	4122	4122	−145.90	1318.87	1072.97	−0.2968	2.5045	2.2077	27
28	0.07365	0.1500	0.01748	3943	3943	−145.40	1318.81	1073.41	−0.2957	2.4991	2.2034	28
29	0.07715	0.1571	0.01747	3771	3771	−144.90	1318.75	1073.85	−0.2947	2.4939	2.1992	29
30	0.08080	0.1645	0.01747	3608	3608	−144.40	1318.69	1074.29	−0.2937	2.4887	2.1950	30
31	0.08461	0.1723	0.01747	3453	3453	−143.90	1318.63	1074.73	−0.2927	2.4835	2.1908	31
32	0.08858	0.1803	0.01747	3305	3305	−143.40	1318.56	1075.16	−0.2918	2.4783	2.1867	32
*32	0.089586	0.18238	0.01602	3304.6	3304.6	0.00	1075.16	1075.16	0.00000	2.1867	2.1867	32*

Fahr. Temp. t(F)	Absolute Pressure p_s Lb/Sq In.	Absolute Pressure p_s In. Hg	Specific Volume, cu ft per lb Sat. Liquid v_f	Specific Volume, cu ft per lb Evap. v_{fg}	Specific Volume, cu ft per lb Sat. Vapor v_g	Enthalpy, Btu per lb Sat. Liquid h_f	Enthalpy, Btu per lb Evap. h_{fg}	Enthalpy, Btu per lb Sat. Vapor h_g	Entropy, Btu per (Lb) (°F) Sat. Liquid s_f	Entropy, Btu per (Lb) (°F) Evap. s_{fg}	Entropy, Btu per (Lb) (°F) Sat. Vapor s_g	Fahr. Temp. t(F)
33	0.092227	0.18778	0.01602	3180.5	3180.5	1.01	1074.59	1075.60	0.00205	2.1811	2.1831	33
34	0.095999	0.19546	0.01602	3061.7	3061.7	2.01	1074.03	1076.04	0.00408	2.1755	2.1796	34
35	0.099908	0.20342	0.01603	2947.8	2947.8	3.02	1073.46	1076.48	0.00613	2.1700	2.1761	35
36	0.10396	0.21166	0.01602	2838.7	2838.7	4.02	1072.90	1076.92	0.00815	2.1644	2.1726	36
37	0.10815	0.22020	0.01602	2734.1	2734.1	5.03	1072.33	1077.36	0.01018	2.1589	2.1691	37
38	0.11249	0.22904	0.01603	2633.8	2633.8	6.03	1071.77	1077.80	0.01220	2.1535	2.1657	38
39	0.11699	0.23819	0.01603	2537.6	2537.6	7.04	1071.20	1078.24	0.01422	2.1480	2.1622	39
40	0.12164	0.24767	0.01609	2445.4	2445.4	8.04	1070.64	1078.68	0.01622	2.1426	2.1588	40
41	0.12646	0.25745	0.01602	2356.9	2356.9	9.05	1070.06	1079.11	0.01824	2.1372	2.1554	41
42	0.13145	0.26763	0.01602	2273.0	2273.0	10.05	1069.50	1079.55	0.02024	2.1318	2.1520	42
43	0.13660	0.27813	0.01602	2190.5	2190.5	11.05	1068.94	1079.99	0.02224	2.1265	2.1487	43
44	0.14194	0.28899	0.01602	2113.3	2113.3	12.06	1068.37	1080.43	0.02423	2.1211	2.1453	44
45	0.14746	0.30023	0.01602	2037.3	2037.3	13.06	1067.81	1080.87	0.02622	2.1158	2.1420	45
46	0.15317	0.31185	0.01602	1965.2	1965.2	14.06	1067.24	1081.30	0.02820	2.1105	2.1387	46
47	0.15907	0.32387	0.01602	1896.0	1896.0	15.06	1066.68	1081.74	0.03018	2.1052	2.1354	47
48	0.16517	0.33629	0.01602	1829.5	1829.5	16.07	1066.11	1082.18	0.03216	2.0999	2.1321	48
49	0.17148	0.34913	0.01602	1765.7	1765.7	17.07	1065.55	1082.62	0.03413	2.0947	2.1288	49
50	0.17799	0.36240	0.01602	1704.3	1704.3	18.07	1064.99	1083.06	0.03610	2.0895	2.1256	50
51	0.18473	0.37611	0.01602	1645.4	1645.4	19.07	1064.42	1083.49	0.03806	2.0842	2.1223	51
52	0.19169	0.39028	0.01602	1588.7	1588.7	20.07	1063.86	1083.93	0.04002	2.0791	2.1191	52
53	0.19888	0.40492	0.01603	1534.3	1534.3	21.07	1063.30	1084.37	0.04197	2.0739	2.1159	53
54	0.20630	0.42002	0.01603	1481.9	1481.9	22.08	1062.72	1084.80	0.04392	2.0688	2.1127	54
55	0.21397	0.43564	0.01603	1431.5	1431.5	23.08	1062.16	1085.24	0.04587	2.0637	2.1096	55
56	0.22188	0.45176	0.01603	1383.1	1383.1	24.08	1061.60	1085.68	0.04781	2.0586	2.1064	56
57	0.23006	0.46840	0.01603	1336.5	1336.5	25.08	1061.04	1086.12	0.04975	2.0535	2.1033	57
58	0.23849	0.48558	0.01603	1291.7	1291.7	26.08	1060.47	1086.55	0.05168	2.0485	2.1002	58
59	0.24720	0.50330	0.01603	1248.6	1248.6	27.08	1059.91	1086.99	0.05361	2.0434	2.0970	59
60	0.25618	0.52160	0.01603	1207.1	1207.1	28.08	1059.34	1087.43	0.05553	2.0385	2.0940	60
61	0.26545	0.54047	0.01604	1167.2	1167.2	29.08	1058.78	1087.86	0.05745	2.0334	2.0909	61
62	0.27502	0.55994	0.01604	1128.7	1128.7	30.08	1058.22	1088.30	0.05937	2.0284	2.0878	62
63	0.28488	0.58002	0.01604	1091.7	1091.7	31.08	1057.65	1088.73	0.06129	2.0235	2.0848	63
64	0.29505	0.60078	0.01604	1056.1	1056.1	32.08	1057.09	1089.17	0.06320	2.0186	2.0818	64
65	0.30554	0.62209	0.01604	1021.7	1021.7	33.08	1056.52	1089.60	0.06510	2.0136	2.0787	65
66	0.31636	0.64411	0.01604	988.63	988.65	34.07	1055.97	1090.04	0.06700	2.0087	2.0757	66
67	0.32750	0.66681	0.01605	956.76	956.78	35.07	1055.40	1090.47	0.06890	2.0039	2.0728	67
68	0.33900	0.69021	0.01605	926.06	926.08	36.07	1054.84	1090.91	0.07080	1.9990	2.0698	68
69	0.35084	0.71433	0.01605	896.47	896.49	37.07	1054.27	1091.34	0.07269	1.9941	2.0668	69
70	0.36304	0.73918	0.01606	867.95	867.97	38.07	1053.71	1091.78	0.07458	1.9893	2.0639	70
71	0.37561	0.76476	0.01605	840.45	840.47	39.07	1053.14	1092.21	0.07648	1.9843	2.0610	71

[a] Compiled by John A. Goff and S. Gratch.

* Extrapolated to represent metastable equilibrium with undercooled liquid.

Table 3 . . . . Thermodynamic Properties of *WATER* at Saturation[a] (*Continued*)

Fahr. Temp. t(F)	Absolute Pressure p_s Lb/Sq In.	p_s In. Hg	Specific Volume, cu ft per lb Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Enthalpy, Btu per lb Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Entropy, Btu per (lb) (°F) Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	Fahr. Temp. t(F)
72	0.35856	0.73113	0.01606	813.95	813.97	40.07	1052.58	1092.65	0.07834	1.9797	2.0580	72
73	0.40190	0.81829	0.01606	765.35	765.40	41.07	1053.01	1093.08	0.08023	1.9749	2.0551	73
74	0.41564	0.84626	0.01606	763.73	763.75	42.06	1051.45	1093.52	0.08209	1.9701	2.0522	74
75	0.42979	0.87506	0.01606	739.93	739.97	43.06	1050.89	1093.95	0.08396	1.9654	2.0494	75
76	0.44435	0.90472	0.01606	717.01	717.03	44.06	1050.32	1094.38	0.08582	1.9607	2.0465	76
77	0.45935	0.93524	0.01607	694.88	694.90	45.06	1049.76	1094.82	0.08769	1.9560	2.0437	77
78	0.47478	0.96666	0.01607	673.52	673.54	46.06	1049.19	1095.25	0.08954	1.9513	2.0408	78
79	0.49066	0.99900	0.01607	652.91	652.93	47.06	1048.63	1095.68	0.09140	1.9466	2.0380	79
80	0.50701	1.0323	0.01607	633.01	633.03	48.05	1048.07	1096.12	0.09325	1.9419	2.0352	80
81	0.52382	1.0665	0.01608	613.80	613.82	49.05	1047.50	1096.55	0.09510	1.9373	2.0324	81
82	0.54113	1.1017	0.01608	595.25	595.27	50.05	1046.93	1096.98	0.09694	1.9328	2.0297	82
83	0.55892	1.1380	0.01608	577.34	577.36	51.05	1046.37	1097.42	0.09878	1.9281	2.0269	83
84	0.57723	1.1752	0.01608	560.06	560.08	52.05	1045.80	1097.85	0.10062	1.9236	2.0242	84
85	0.59604	1.2136	0.01609	543.33	543.35	53.05	1045.23	1098.28	0.10246	1.9189	2.0214	85
86	0.61540	1.2530	0.01609	527.19	527.21	54.04	1044.67	1098.71	0.10429	1.9144	2.0187	86
87	0.63530	1.2935	0.01609	511.60	511.62	55.04	1044.10	1099.14	0.10611	1.9099	2.0160	87
88	0.65575	1.3351	0.01610	496.54	496.54	56.04	1043.54	1099.58	0.10794	1.9054	2.0133	88
89	0.67678	1.3779	0.01610	481.98	481.98	57.04	1042.97	1100.01	0.10976	1.9008	2.0106	89
90	0.69838	1.4219	0.01610	467.88	467.90	58.04	1042.40	1100.44	0.11158	1.8963	2.0079	90
91	0.72059	1.4671	0.01610	454.23	454.23	59.03	1041.84	1100.87	0.11339	1.8919	2.0053	91
92	0.74340	1.5136	0.01611	441.12	441.12	60.03	1041.27	1101.30	0.11520	1.8874	2.0026	92
93	0.76684	1.5613	0.01611	428.38	428.40	61.03	1040.70	1101.73	0.11701	1.8830	2.0000	93
94	0.79091	1.6103	0.01611	416.07	416.09	62.03	1040.13	1102.16	0.11881	1.8786	1.9974	94
95	0.81564	1.6607	0.01613	404.17	404.19	63.03	1039.56	1102.59	0.12061	1.8741	1.9947	95
96	0.84103	1.7124	0.01612	392.65	392.67	64.02	1039.00	1103.02	0.12241	1.8698	1.9922	96
97	0.86711	1.7655	0.01612	381.51	381.53	65.02	1038.43	1103.45	0.12420	1.8654	1.9896	97
98	0.89388	1.8200	0.01613	370.73	370.75	66.02	1037.86	1103.88	0.12600	1.8610	1.9870	98
99	0.92137	1.8759	0.01613	360.30	360.32	67.02	1037.29	1104.31	0.12778	1.8566	1.9844	99
100	0.94959	1.9334	0.01613	350.20	350.22	68.02	1036.73	1104.74	0.12957	1.8523	1.9819	100
101	0.97854	1.9923	0.01614	340.42	340.44	69.01	1036.16	1105.17	0.13135	1.8480	1.9793	101
102	1.0083	2.0529	0.01614	330.98	330.98	70.01	1035.58	1105.59	0.13313	1.8437	1.9768	102
103	1.0388	2.1149	0.01614	321.80	321.82	71.01	1035.01	1106.02	0.13490	1.8394	1.9743	103
104	1.0700	2.1786	0.01614	313.93	313.95	72.01	1034.44	1106.45	0.13667	1.8351	1.9718	104
105	1.1021	2.2440	0.01615	304.34	304.36	73.01	1033.87	1106.88	0.13844	1.8309	1.9693	105
106	1.1351	2.3110	0.01615	296.02	296.04	74.01	1033.29	1107.30	0.14021	1.8266	1.9668	106
107	1.1688	2.3798	0.01616	287.96	287.98	75.00	1032.73	1107.73	0.14197	1.8224	1.9644	107

Fahr. Temp. t(F)	Absolute Pressure p_s Lb/Sq In.	p_s In. Hg	Specific Volume, cu ft per lb Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Enthalpy, Btu per lb Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Entropy, Btu per (lb) (°F) Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	Fahr. Temp. t(F)
108	1.2035	2.4503	0.01616	280.14	280.16	76.00	1032.16	1108.16	0.14373	1.8182	1.9619	108
109	1.2390	2.5228	0.01616	272.58	272.60	77.00	1031.58	1108.58	0.14549	1.8140	1.9595	109
110	1.2754	2.5968	0.01617	265.26	265.28	78.00	1031.01	1109.01	0.14724	1.8098	1.9570	110
111	1.3128	2.6728	0.01617	258.16	258.18	79.00	1030.44	1109.44	0.14899	1.8056	1.9546	111
112	1.3510	2.7507	0.01617	251.27	251.27	80.00	1029.86	1109.86	0.15074	1.8015	1.9522	112
113	1.3902	2.8306	0.01618	244.57	244.59	80.99	1029.30	1110.29	0.15248	1.7973	1.9498	113
114	1.4305	2.9125	0.01618	238.10	238.12	81.99	1028.73	1110.71	0.15423	1.7932	1.9474	114
115	1.4717	2.9963	0.01618	231.82	231.84	82.99	1028.15	1111.14	0.15596	1.7890	1.9450	115
116	1.5139	3.0823	0.01619	225.73	225.75	83.99	1027.57	1111.56	0.15770	1.7849	1.9426	116
117	1.5571	3.1703	0.01619	219.83	219.85	84.99	1026.99	1111.98	0.15943	1.7809	1.9403	117
118	1.6014	3.2606	0.01620	214.10	214.12	85.99	1026.42	1112.41	0.16116	1.7767	1.9379	118
119	1.6468	3.3530	0.01620	208.54	208.56	86.98	1025.85	1112.83	0.16289	1.7727	1.9356	119
120	1.6933	3.4477	0.01620	203.18	203.18	87.98	1025.28	1113.26	0.16461	1.7687	1.9333	120
121	1.7409	3.5446	0.01621	197.93	197.95	88.98	1024.70	1113.68	0.16634	1.7647	1.9310	121
122	1.7897	3.6438	0.01621	192.87	192.89	89.98	1024.13	1114.10	0.16805	1.7606	1.9286	122
123	1.8396	3.7455	0.01622	187.93	187.95	90.98	1023.54	1114.52	0.16977	1.7566	1.9264	123
124	1.8907	3.8496	0.01622	183.15	183.17	91.98	1022.96	1114.94	0.17148	1.7526	1.9241	124
125	1.9430	3.9561	0.01622	178.51	178.53	92.98	1022.39	1115.37	0.17319	1.7486	1.9218	125
126	1.9966	4.0651	0.01623	174.00	174.02	93.98	1021.81	1115.79	0.17490	1.7446	1.9195	126
127	2.0514	4.1768	0.01623	169.63	169.65	94.97	1021.24	1116.21	0.17660	1.7407	1.9173	127
128	2.1075	4.2910	0.01624	165.38	165.40	95.97	1020.66	1116.63	0.17830	1.7367	1.9150	128
129	2.1649	4.4078	0.01624	161.26	161.28	96.97	1020.08	1117.05	0.18000	1.7328	1.9128	129
130	2.2237	4.5274	0.01625	157.25	157.27	97.97	1019.50	1117.47	0.18170	1.7289	1.9106	130
131	2.2838	4.6498	0.01625	153.35	153.38	98.97	1018.92	1117.89	0.18339	1.7250	1.9084	131
132	2.3452	4.7750	0.01626	149.58	149.60	99.97	1018.34	1118.31	0.18508	1.7211	1.9062	132
133	2.4081	4.9030	0.01626	145.91	145.93	100.97	1017.76	1118.73	0.18676	1.7172	1.9040	133
134	2.4725	5.0340	0.01626	142.34	142.36	101.97	1017.18	1119.15	0.18845	1.7134	1.9018	134
135	2.5382	5.1679	0.01627	138.87	138.89	102.97	1016.59	1119.56	0.19013	1.7095	1.8996	135
136	2.6055	5.3049	0.01627	135.50	135.52	103.97	1016.01	1119.98	0.19181	1.7056	1.8974	136
137	2.6743	5.4450	0.01628	132.22	132.24	104.97	1015.43	1120.40	0.19348	1.7018	1.8953	137
138	2.7446	5.5881	0.01628	129.04	129.06	105.97	1014.85	1120.82	0.19516	1.6979	1.8931	138
139	2.8165	5.7345	0.01629	125.94	125.96	106.97	1014.26	1121.23	0.19683	1.6940	1.8910	139
140	2.8900	5.8842	0.01629	122.94	122.96	107.96	1013.69	1121.65	0.19850	1.6903	1.8888	140
141	2.9651	6.0371	0.01630	120.01	120.03	108.96	1013.11	1122.07	0.20016	1.6865	1.8867	141
142	3.0419	6.1934	0.01630	117.16	117.18	109.96	1012.52	1122.48	0.20183	1.6828	1.8846	142
143	3.1204	6.3533	0.01631	114.40	114.42	110.96	1011.94	1122.90	0.20345	1.6790	1.8825	143

[a] Compiled by John A. Goff and S. Gratch.

Table 3 . . . . Thermodynamic Properties of *WATER* at Saturation[a] (*Concluded*)

Fahr. Temp. t(F)	Absolute Pressure p_s Lb/Sq In.	p_s In. Hg	Specific Volume, cu ft per lb Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Enthalpy, Btu per lb Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Entropy, Btu per (lb) (°F) Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	Fahr. Temp. t(F)
144	3.2006	6.5184	0.01631	111.70	111.72	111.96	1011.35	1123.31	0.20514	1.6753	1.8804	144
145	3.2825	6.6832	0.01632	109.09	109.11	112.96	1010.77	1123.73	0.20679	1.6715	1.8783	145
146	3.3662	6.8516	0.01632	106.56	106.58	113.96	1010.18	1124.14	0.20845	1.6678	1.8763	146
147	3.4517	7.0277	0.01633	104.08	104.08	114.96	1009.59	1124.55	0.21010	1.6641	1.8742	147
148	3.5390	7.2056	0.01633	101.65	101.67	115.96	1009.01	1124.97	0.21174	1.6604	1.8721	148
149	3.6282	7.3879	0.01634	99.306	99.322	116.96	1008.43	1125.38	0.21339	1.6567	1.8701	149
150	3.7194	7.5727	0.01634	97.022	97.038	117.96	1007.83	1125.79	0.21503	1.6530	1.8680	150
151	3.8124	7.7622	0.01635	94.790	94.816	118.96	1007.24	1126.20	0.21667	1.6493	1.8660	151
152	3.9074	7.9566	0.01635	92.635	92.651	119.96	1006.66	1126.62	0.21830	1.6457	1.8640	152
153	4.0044	8.1532	0.01636	90.528	90.544	120.97	1006.06	1127.03	0.21994	1.6421	1.8620	153
154	4.1035	8.3548	0.01636	88.477	88.493	121.97	1005.47	1127.44	0.22157	1.6384	1.8600	154
155	4.2045	8.5607	0.01637	86.480	86.496	122.97	1004.88	1127.85	0.22320	1.6348	1.8580	155
156	4.3078	8.7708	0.01637	84.538	84.552	123.97	1004.29	1128.26	0.22482	1.6312	1.8560	156
157	4.4132	8.9853	0.01638	82.642	82.658	124.97	1003.70	1128.67	0.22645	1.6276	1.8540	157
158	4.5207	9.2043	0.01638	80.794	80.814	125.97	1003.11	1129.08	0.22807	1.6239	1.8520	158
159	4.6304	9.4278	0.01639	79.001	79.017	126.97	1002.51	1129.48	0.22969	1.6204	1.8501	159
160	4.7424	9.6556	0.01639	77.251	77.267	127.97	1001.92	1129.89	0.23130	1.6168	1.8481	160
161	4.8566	9.8882	0.01640	75.548	75.564	128.97	1001.33	1130.30	0.23291	1.6133	1.8462	161
162	4.9733	10.126	0.01640	73.885	73.901	129.97	1000.74	1130.71	0.23453	1.6097	1.8442	162
163	5.0921	10.368	0.01641	72.267	72.283	130.98	1000.13	1131.11	0.23614	1.6062	1.8423	163
164	5.2134	10.615	0.01642	70.690	70.706	131.98	999.54	1131.52	0.23774	1.6027	1.8404	164
165	5.3372	10.867	0.01642	69.153	69.169	132.98	998.94	1131.92	0.23935	1.5990	1.8384	165
166	5.4634	11.124	0.01643	67.654	67.670	133.98	998.35	1132.33	0.24095	1.5956	1.8365	166
167	5.5921	11.386	0.01643	66.194	66.210	134.98	997.75	1132.73	0.24255	1.5920	1.8346	167
168	5.7233	11.653	0.01644	64.770	64.786	135.98	997.16	1133.14	0.24414	1.5887	1.8328	168
169	5.8572	11.925	0.01644	63.383	63.398	136.99	996.55	1133.54	0.24574	1.5852	1.8309	169
170	5.9936	12.203	0.01645	62.029	62.045	137.99	995.95	1133.94	0.24733	1.5817	1.8290	170
171	6.1328	12.487	0.01645	60.710	60.726	138.99	995.36	1134.35	0.24892	1.5782	1.8271	171
172	6.2748	12.775	0.01646	59.423	59.439	139.99	994.76	1134.75	0.25051	1.5748	1.8253	172
173	6.4192	13.070	0.01647	58.168	58.184	141.00	994.15	1135.15	0.25209	1.5713	1.8234	173
174	6.5666	13.370	0.01647	56.944	56.960	142.00	993.55	1135.55	0.25367	1.5679	1.8216	174
175	6.7168	13.676	0.01648	55.750	55.766	143.00	992.95	1135.95	0.25525	1.5644	1.8197	175
176	6.8699	13.987	0.01648	54.586	54.603	144.00	992.35	1136.35	0.25683	1.5611	1.8179	176
177	7.0259	14.305	0.01649	53.450	53.466	145.00	991.75	1136.75	0.25841	1.5577	1.8161	177

Fahr. Temp. t(F)	Absolute Pressure p_s Lb/Sq In.	p_s In. Hg	Specific Volume, cu ft per lb Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Enthalpy, Btu per lb Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Entropy, Btu per (lb) (°F) Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	Fahr. Temp. t(F)
178	7.1849	14.629	0.01650	52.357	52.357	146.01	991.14	1137.15	0.25998	1.5543	1.8143	178
179	7.3489	14.959	0.01650	51.280	51.276	147.01	990.54	1137.55	0.26155	1.5508	1.8124	179
180	7.5119	15.295	0.01651	50.203	50.220	148.01	989.93	1137.94	0.26313	1.5475	1.8106	180
181	7.6801	15.637	0.01651	49.173	49.190	149.03	989.32	1138.34	0.26468	1.5442	1.8089	181
182	7.8514	15.986	0.01652	48.168	48.185	150.03	988.72	1138.74	0.26625	1.5408	1.8071	182
183	8.0258	16.341	0.01652	47.187	47.204	151.03	988.12	1139.14	0.26781	1.5375	1.8053	183
184	8.2035	16.703	0.01653	46.229	46.246	152.03	987.50	1139.53	0.26937	1.5341	1.8035	184
185	8.3845	17.071	0.01654	45.294	45.311	153.03	986.89	1139.93	0.27093	1.5308	1.8017	185
186	8.5688	17.446	0.01654	44.381	44.398	154.04	986.28	1140.32	0.27248	1.5275	1.8000	186
187	8.7565	17.829	0.01655	43.506	43.523	155.04	985.67	1140.71	0.27404	1.5242	1.7983	187
188	8.9476	18.218	0.01656	42.619	42.636	156.04	985.07	1141.11	0.27559	1.5209	1.7965	188
189	9.1422	18.614	0.01656	41.769	41.786	157.05	984.45	1141.50	0.27713	1.5176	1.7947	189
190	9.3403	19.017	0.01657	40.939	40.956	158.05	983.84	1141.89	0.27868	1.5143	1.7930	190
191	9.5420	19.428	0.01658	40.135	40.145	159.06	983.23	1142.28	0.28022	1.5111	1.7913	191
192	9.7473	19.846	0.01658	39.337	39.354	160.06	982.61	1142.67	0.28176	1.5078	1.7896	192
193	9.9562	20.271	0.01659	38.563	38.580	161.06	982.00	1143.06	0.28330	1.5045	1.7878	193
194	10.169	20.704	0.01659	37.807	37.824	162.07	981.38	1143.45	0.28484	1.5013	1.7861	194
195	10.386	21.145	0.01660	37.066	37.086	163.08	980.76	1143.84	0.28638	1.4980	1.7844	195
196	10.606	21.594	0.01661	36.348	36.365	164.08	980.15	1144.23	0.28791	1.4949	1.7828	196
197	10.830	22.050	0.01661	35.643	35.660	165.08	979.54	1144.63	0.28944	1.4917	1.7811	197
198	11.058	22.515	0.01662	34.954	34.971	166.09	978.91	1145.00	0.29097	1.4884	1.7794	198
199	11.290	22.987	0.01663	34.281	34.298	167.10	978.29	1145.39	0.29250	1.4852	1.7777	199
200	11.526	23.468	0.01663	33.622	33.640	168.10	977.68	1145.78	0.29402	1.4820	1.7760	200
201	11.767	23.957	0.01664	32.980	32.997	169.11	977.05	1146.16	0.29554	1.4788	1.7744	201
202	12.011	24.455	0.01665	32.351	32.368	170.11	976.43	1146.54	0.29706	1.4756	1.7727	202
203	12.260	24.961	0.01665	31.737	31.754	171.12	975.81	1146.93	0.29858	1.4725	1.7711	203
204	12.513	25.476	0.01666	31.136	31.153	172.12	975.19	1147.31	0.30010	1.4693	1.7694	204
205	12.770	26.000	0.01667	30.549	30.566	173.13	974.56	1147.69	0.30161	1.4662	1.7678	205
206	13.031	26.533	0.01667	29.974	29.991	174.14	973.94	1148.08	0.30312	1.4631	1.7662	206
207	13.297	27.074	0.01668	29.412	29.430	175.14	973.32	1148.46	0.30463	1.4600	1.7646	207
208	13.568	27.625	0.01669	28.863	28.880	176.15	972.69	1148.84	0.30614	1.4568	1.7629	208
209	13.843	28.185	0.01669	28.325	28.342	177.16	972.06	1149.22	0.30765	1.4538	1.7613	209
210	14.123	28.754	0.01670	27.801	27.818	178.17	971.43	1149.60	0.30915	1.4506	1.7597	210
211	14.407	29.333	0.01671	27.287	27.304	179.17	970.81	1149.98	0.31065	1.4474	1.7581	211
212	14.696	29.921	0.01671	26.784	26.801	180.18	970.17	1150.35	0.31215	1.4444	1.7565	212

[a] Compiled by John A. Goff and S. Gratch.

Table 4 .... Properties of *SATURATED STEAM*: Pressure Table[a]

Abs. Press. In. Hg p	Temp. F t	Specific Volume		Enthalpy			Entropy			Abs. Press. In. Hg p
		Sat. Liquid v_f	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
0.25	40.23	0.01602	2423.7	8.28	1071.1	1079.4	0.0166	2.1423	2.1589	0.25
0.50	58.80	0.01604	1256.4	26.86	1060.6	1087.5	0.0532	2.0453	2.0985	0.50
0.75	70.43	0.01606	856.1	38.47	1054.0	1092.5	0.0754	1.9881	2.0635	0.75
1.00	79.03	0.01608	652.3	47.05	1049.2	1096.3	0.0914	1.9473	2.0387	1.00
1.5	91.72	0.01611	444.9	59.71	1042.0	1101.7	0.1147	1.8894	2.0041	1.5
2	101.14	0.01614	339.2	69.10	1036.6	1105.7	0.1316	1.8481	1.9797	2
4	125.43	0.01622	176.7	93.34	1022.7	1116.0	0.1738	1.7476	1.9214	4
6	140.78	0.01630	120.72	108.67	1013.6	1122.3	0.1996	1.6881	1.8877	6
8	152.24	0.01635	92.16	120.13	1006.9	1127.0	0.2186	1.6454	1.8640	8
10	161.49	0.01640	74.76	129.38	1001.4	1130.8	0.2335	1.6121	1.8456	10
12	169.28	0.01644	63.03	137.18	996.7	1133.9	0.2460	1.5847	1.8307	12
14	176.05	0.01648	54.55	143.96	992.6	1136.6	0.2568	1.5613	1.8181	14
16	182.05	0.01652	48.14	149.98	988.9	1138.9	0.2662	1.5410	1.8072	16
18	187.45	0.01655	43.11	155.39	985.7	1141.1	0.2746	1.5231	1.7977	18
20	192.37	0.01658	39.07	160.33	982.7	1143.0	0.2822	1.5069	1.7891	20
22	196.90	0.01661	35.73	164.87	979.8	1144.7	0.2891	1.4923	1.7814	22
24	201.09	0.01664	32.94	169.09	977.2	1146.3	0.2955	1.4789	1.7744	24
26	205.00	0.01667	30.56	173.02	974.8	1147.8	0.3014	1.4665	1.7679	26
28	208.67	0.01669	28.52	176.72	972.5	1149.2	0.3069	1.4550	1.7619	28
30	212.13	0.01672	26.74	180.19	970.3	1150.5	0.3122	1.4442	1.7564	30
Lb/Sq In.										*Lb/Sq In.*
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	14.696
16	216.32	0.01674	24.75	184.42	967.6	1152.0	0.3184	1.4313	1.7497	16
18	222.41	0.01679	22.17	190.56	963.6	1154.2	0.3275	1.4128	1.7403	18
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319	20
22	233.07	0.01687	18.375	201.33	956.8	1158.1	0.3431	1.3811	1.7242	22
24	237.82	0.01691	16.938	206.14	953.7	1159.8	0.3500	1.3672	1.7172	24
26	242.25	0.01694	15.715	210.62	950.7	1161.3	0.3564	1.3544	1.7108	26
28	246.41	0.01698	14.663	214.83	947.9	1162.7	0.3623	1.3425	1.7048	28
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	30
32	254.05	0.01704	12.940	222.59	942.8	1165.4	0.3733	1.3209	1.6941	32
34	257.58	0.01707	12.226	226.18	940.3	1166.5	0.3783	1.3110	1.6893	34
36	260.95	0.01709	11.588	229.60	938.0	1167.6	0.3831	1.3017	1.6848	36
38	264.16	0.01712	11.015	232.89	935.8	1168.7	0.3876	1.2929	1.6805	38
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763	40
42	270.21	0.01717	10.029	239.04	931.6	1170.7	0.3960	1.2764	1.6724	42
44	273.05	0.01720	9.601	241.95	929.6	1171.6	0.4000	1.2687	1.6687	44
46	275.80	0.01722	9.209	244.75	927.7	1172.4	0.4038	1.2613	1.6652	46
48	278.45	0.01725	8.848	247.47	925.8	1173.3	0.4075	1.2542	1.6617	48
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585	50
52	283.49	0.01729	8.208	252.63	922.2	1174.8	0.4144	1.2409	1.6553	52
54	285.90	0.01731	7.922	255.09	920.5	1175.6	0.4177	1.2346	1.6523	54
56	288.23	0.01733	7.656	257.50	918.8	1176.3	0.4209	1.2285	1.6494	56
58	290.50	0.01736	7.407	259.82	917.1	1176.9	0.4240	1.2226	1.6466	58
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438	60
62	294.85	0.01740	6.957	264.30	913.9	1178.2	0.4300	1.2112	1.6412	62
64	296.94	0.01742	6.752	266.45	912.3	1178.8	0.4328	1.2059	1.6387	64
66	298.99	0.01744	6.560	268.55	910.8	1179.4	0.4356	1.2006	1.6362	66
68	300.98	0.01746	6.378	270.60	909.4	1180.0	0.4383	1.1955	1.6338	68
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	70
72	304.83	0.01750	6.044	274.57	906.5	1181.1	0.4435	1.1857	1.6292	72
74	306.68	0.01752	5.890	276.49	905.1	1181.6	0.4460	1.1810	1.6270	74
76	308.50	0.01754	5.743	278.37	903.7	1182.1	0.4484	1.1764	1.6248	76
78	310.29	0.01755	5.604	280.21	902.4	1182.6	0.4508	1.1720	1.6228	78
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207	80
82	313.74	0.01759	5.346	283.79	899.7	1183.5	0.4554	1.1633	1.6187	82
84	315.42	0.01761	5.226	285.53	898.5	1184.0	0.4576	1.1592	1.6168	84
86	317.07	0.01762	5.111	287.24	897.2	1184.4	0.4598	1.1551	1.6149	86
88	318.68	0.01764	5.001	288.91	895.9	1184.8	0.4620	1.1510	1.6130	88
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112	90
92	321.83	0.01768	4.796	292.18	893.5	1185.7	0.4661	1.1433	1.6094	92
94	323.36	0.01769	4.699	293.78	892.3	1186.1	0.4682	1.1394	1.6076	94
96	324.87	0.01771	4.606	295.34	891.1	1186.4	0.4702	1.1358	1.6060	96
98	326.35	0.01772	4.517	296.89	889.9	1186.8	0.4721	1.1322	1.6043	98
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	100
150	358.42	0.01809	3.015	330.51	863.6	1194.1	0.5138	1.0556	1.5694	150
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	200
300	417.33	0.01890	1.5433	393.84	809.0	1202.8	0.5879	0.9225	1.5104	300
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	400
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	500

[a] Reprinted by permission from *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes published by John Wiley and Sons, Inc., 1936 edition.

Table 5 .... Coefficients A, B, C Appearing in Equations 30, 31, 32, Maximum Values of Corrections Defined by Equations 30, 31, 32. Degree of Saturation at Which These Maxima Occur, $\bar{\mu}_m$. Maximum Value of Correction Defined by Equation 33. Degree of Saturation at Which This Maximum Occurs, $\bar{\mu}_m$

(Standard Atmospheric Pressure)

t (F)	A (ft³/lb$_a$)	B (Btu/lb$_a$)	C (Btu/F/lb$_a$)	$\bar{v}_{max}$ (ft³/lb$_a$)	$\bar{h}_{max}$ (Btu/lb$_a$)	$\bar{s}_{max}$ (Btu/F/lb$_a$)	$\bar{\mu}_m$	$\bar{s}_{max}$ (Btu/F/lb$_a$)	$\bar{\mu}_m$
96	0.0018	0.0268	0.00004	0.0004	0.0069	0.00001	0.4925	0.0015	0.3650
112	0.0042	0.0650	0.00009	0.0010	0.0155	0.00002	0.4878	0.0025	0.3632
128	0.0096	0.1439	0.00020	0.0022	0.0332	0.00005	0.4805	0.0040	0.3602
144	0.0215	0.3149	0.00042	0.0047	0.0693	0.00009	0.4691	0.0065	0.3557
160	0.0487	0.6969	0.00091	0.0099	0.1418	0.00019	0.4511	0.0106	0.3485
176	0.1169	1.636	0.00207	0.0207	0.2903	0.00037	0.4213	0.0179	0.3363
192	0.3363	4.608	0.00567	0.0451	0.6180	0.00076	0.3662	0.0333	0.3129

air in relation to saturation, is necessary for locating and describing states on the chart. The arrangement of the families of curves of constant dry-bulb temperature, wet-bulb temperature, volume, relative humidity, and degree of saturation, are shown in Fig. 4.

Mass and energy balances deal only with net changes between definite states; the detailed history of a change is not involved. A chart used to facilitate such calculations must primarily aid in clearly establishing states. Lines drawn on the chart to connect different states need have no other significance than being loci lines, that is, lines which contain the two terminal points of a change according to the particular overall conditions imposed. Loci lines are commonly called *condition lines* for the processes concerned. On the ASHAE Chart a condition line is characterized by the ratio $(h_2 - h_1)/(W_2 - W_1)$.

An abridgment of the ASHAE Psychrometric Chart appears in Fig. 5. A large size (24 x 32) chart will be found inside of the back cover.

The chart is drawn for standard atmospheric pressure. Steady flow changes commonly involve pressure drops with the flow, but so long as these pressure drops remain a small fraction of the barometric pressure, no appreciable errors need arise from using a constant pressure chart. For many design problems, the use of the standard pressure chart will not incur any undue error up to about 2000 ft above sea level.

The region above the saturation curve is a two-phase region giving equilibrium states for water in both the liquid and vapor phases. The ordinate of a point in this region is the weight of water in both phases per pound of dry air in the vapor phase, neglecting any dissolved air in the condensed phase. The ordinate at the saturation curve, or the

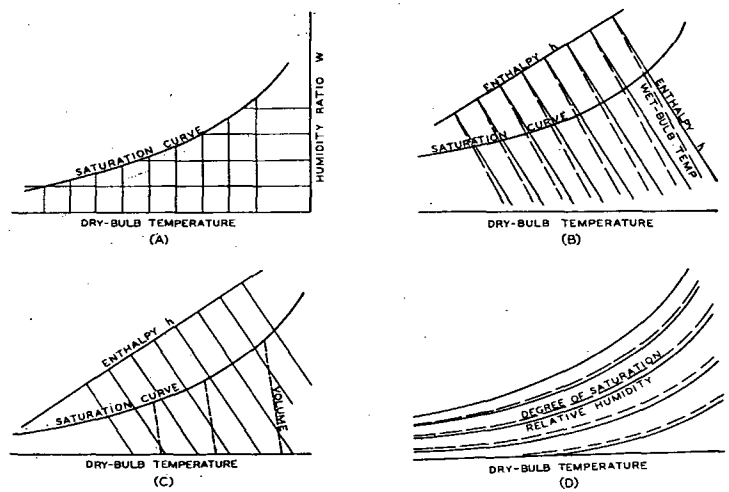


Fig. 4 .... Arrangement of Families of Curves on ASHAE Psychrometric Chart

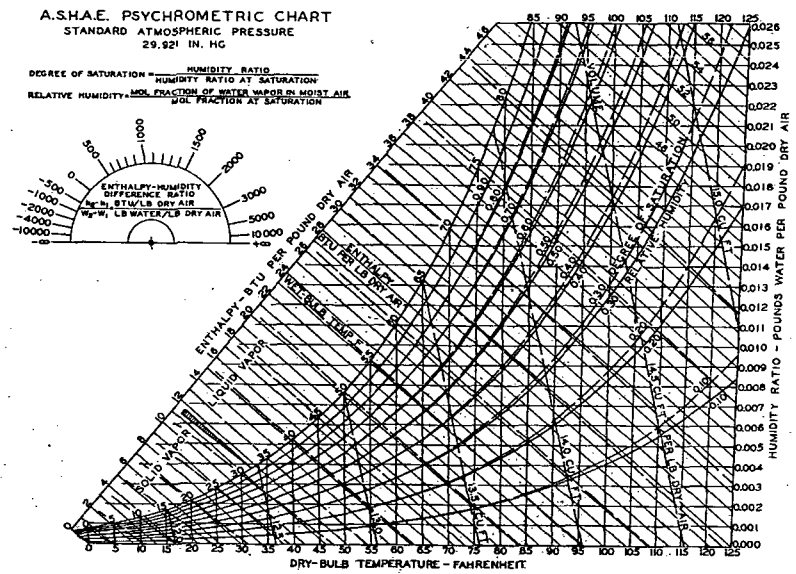


Fig. 5 . . . . Abridgment of ASHAE Psychrometric Chart

point at which the break in the isotherm through the point in question occurs, is the weight of water per pound of dry air in the vapor phase. Consequently, the difference between the two ordinates is the weight of condensed moisture per pound of dry air in the vapor phase.

The shaded solid-liquid-vapor region at 32 F is an isothermal three-phase zone, and separates the liquid-vapor zone from the solid-vapor zone. The temperature is 32 F throughout the shaded area.

It is possible to obtain two values for the wet-bulb temperature when this temperature is below 32 F. If the bulb of a thermometer is dipped into water at a temperature slightly above 32 F and held in a stream of air whose wet-bulb temperature is below 32 F, the temperature indicated by the thermometer will drop rapidly until a minimum is reached below 32 F. This will be accomplished without the formation of ice on the bulb of the thermometer. After reaching this minimum temperature, the reading will jump back to 32 F and remain there until the water on the bulb is frozen, after which it will slowly drop again until equilibrium is reached. The final temperature may be higher or lower than the first minimum reading, or it may be the same depending on the amount of moisture present in the mixture. In the absence of reliable data on the wet-bulb temperature over subcooled water, the chart, below 32 F, has been drawn for the equilibrium condition, that is, the values plotted on the ASHAE chart are for the condition where the minimum temperature is reached with ice on the bulb of the thermometer.

USE OF TABLE 2 AND THE ASHAE PSYCHROMETRIC CHART

The use of Table 2 and the ASHAE PSYCHROMETRIC CHART in analyzing typical air-conditioning problems is best explained by means of illustrative examples. In each of the following it is to be understood that the processes in question take place at a constant pressure of 29.921 in. Hg, i.e., standard atmospheric pressure.

Example 1: Determine the enthalpy of moist air at 80 F dry-bulb temperature and 0.40 degree of saturation.

Solution a: From the data of Table 2, at 80 F, h_a = 19.221 Btu per lb of dry air and h_{as} = 24.47 Btu per lb of dry air. Then h at the specified conditions is 19.221 + 0.40(24.47) = 29.01 Btu per lb of dry air.

Solution b: From the ASHAE Chart. Follow the 80 F dry-bulb line upward until it intersects the 0.40 degree of saturation line. From this intersection, follow the line of constant enthalpy to the enthalpy scale and read 29.00 Btu per lb of dry air.

Example 2: Determine the thermodynamic wet-bulb temperature of moist air at the conditions of *Example 1*.

Solution a: From the data of Table 2. Applying Equation 8, h_1 = 29.01 Btu per lb of dry air (*Example 1*). As a first approximation this is $h_1{}^*$ the enthalpy at saturation at the thermodynamic wet-bulb temperature which is, therefore, approximately 63.5 F. W^* at 63.5 F is 12.57 × 10⁻³ lb of water vapor per lb of dry air, and W_1 is 0.02233 × 0.40 = 0.00893 lb of water vapor per lb of dry air. The specific enthalpy of liquid water at 63.5 F is 31.58 Btu per lb of water. As a second approximation, h^* = 29.01 + (0.01257 − 0.00893) (31.58) = 29.12 Btu per lb of dry air. Interpolation in Table 2 gives as the final answer t^* = 63.64 F.

Solution b: From the ASHAE Chart. At the intersection of the 80 F dry-bulb temperature line and the 0.40 degree of saturation line, read the thermodynamic wet-bulb temperature.

Heating of Moist Air at Constant Pressure Without Addition of Moisture

Example 3: Air initially at 20 F, 0.80 degree of saturation, is heated to 120 F. Find the quantity of heat required to process 20,000 cfm of heated air.

The process is diagrammatically illustrated in Fig. 6. The energy equation for the process is

$$Gh_1 + {}_1q_2 = Gh_2$$

$${}_1q_2 = G(h_2 - h_1)$$

Solution a: From the data of Table 2. The initial humidity ratio, which is the same as the final humidity ratio, is 0.80 (0.002152) = 0.001722 lb of water vapor per lb of dry air; the initial enthalpy is 4.804 + 0.80(2.302) = 6.646 Btu per lb of dry air; the final degree of saturation is 0.001722/0.08149 = 0.02113; the final enthalpy is 28.841 + 0.02113(90.70) = 30.757 Btu per lb of dry air; the final volume is 14.611 + 0.02113 (1.905) = 14.651 cu ft per lb of dry air. Since 20,000 cfm of heated air are to be processed, the total quantity of heat required is

$${}_1q_2 = (20{,}000/14.651) \times 24.111 = 32{,}914 \text{ Btu per min.}$$

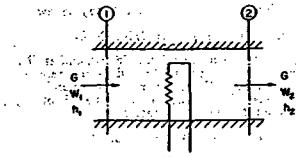


Fig. 6 . . . . Illustration of Process of *Example 3*

Solution b: From the ASHAE Chart. The process is represented by the horizontal line 1–2, Fig. 7. The initial enthalpy, at 20 F dry-bulb temperature and 0.80 degree of saturation, is 6.65 Btu per lb of dry air. Since the final humidity ratio is the same as the initial humidity ratio, the ratio $(h_2 - h_1)/(W_2 - W_1) = \infty$. The horizontal line 1–2, Fig. 7, then represents the condition line for the process, and the final state of the moist air must lie on this line. The final state is located at the point at which the 120 F dry-bulb temperature line crosses the condition line, and is labeled Point 2 on the figure. At this condition the final enthalpy is 30.8 Btu per lb of dry air and the final specific volume is 14.65 cu ft per lb of dry air. Substituting these values in the energy equation,

$${}_1q_2 = (20{,}000/14.65) \times (30.8 - 6.65) = 32{,}950 \text{ Btu per min.}$$

Cooling of Moist Air at Constant Pressure with Condensation of Water

Referring to Fig. 8, moist air cooled from State 1 passes through successive states along the line, $W = W_1$ = constant, until the saturation line is intersected. The temperature at this point of intersection is by definition the dew-point temperature for State 1.

Further cooling through successive equilibrium states is accompanied by condensation. The succession of states for the total system, moist air, and liquid water, is represented by a continuation of the $W = W_1$ line into the liquid-vapor region. (Temperatures below 32 F would involve the solid-vapor region.) Consider that the final temperature is t_3. The final enthalpy is then h_3; the liquid water formed is $(W_1 - W_3)$,

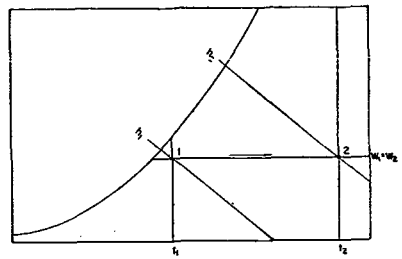


Fig. 7 . . . . Solution of *Example 3* on ASHAE Psychrometric Chart

where Point 3 is at the intersection of the isotherm through 2 and the saturation curve; the final humidity ratio of the moist air is W_3; and this final moist air has dew-point, wet-bulb, and dry-bulb temperatures all equal to t_3.

Example 4: How much heat must be removed from 20,000 cfm of air at 95 F dry-bulb temperature and 0.50 degree of saturation to cool the air to 70 F, saturated?

Solution a: From the data of Table 2. The initial humidity ratio is 0.50(0.03673) = 0.01837 lb of water vapor per lb of dry air; the initial enthalpy is 22.827 + 0.50(40.49) = 43.072 Btu per lb of dry air; the humidity ratio at saturation at the final temperature is 0.01582 lb of water vapor per lb of dry air; the quantity of liquid formed is 0.01837 − 0.01582 = 0.00255 lb of water vapor per lb of dry air; h_{ws} at 70 F is 38.11 Btu per lb of water; the initial specific volume is 13.980 + 0.50(0.822) = 14.391 cu ft per lb of dry air.

Fig. 9 illustrates the process diagrammatically. The energy equation for the process is

$$Gh_1 = Gh_3 + G(W_1 - W_3)h_{ws} + {}_1q_3$$

$${}_1q_3 = G[h_1 - h_3 - (W_1 - W_3)h_{ws}]$$

$$= \frac{20{,}000}{14.391} \times (43.072 - 34.09 - 0.00255 \times 38.07)$$

$$= 12{,}350 \text{ Btu per min.}$$

Solution b: From the ASHAE Chart. Two methods may be used to solve the problem by use of the psychrometric chart. The simpler is to use the region to the left of the saturation line (Fig. 8). From Point 1 draw a horizontal line on the chart

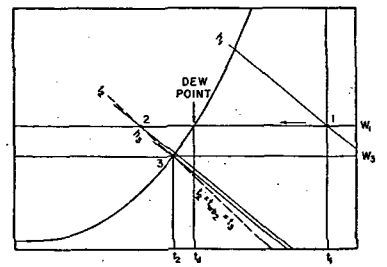


Fig. 8 . . . . Cooling of Air at Constant Pressure Shown on ASHAE Psychrometric Chart

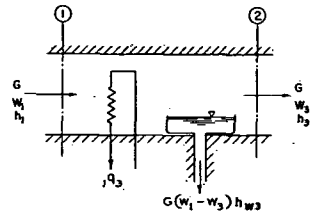


Fig. 9 .... Illustration of Process of *Example 4*

until it intersects the constant temperature line in the liquid-vapor region corresponding to the final temperature 70 F. This is shown as Point 2 on the diagram. Then,

$$_1q_3 = G(h_1 - h_3)$$

The initial enthalpy is 43 Btu per lb of dry air; the initial specific volume is 14.4 cu ft per lb of dry air; and the final enthalpy is 34.2 Btu per lb of dry air. The solution of the problem is

$$_1q_3 = \frac{20{,}000}{14.4} \times (43 - 34.2) = 12{,}200 \text{ Btu per min.}$$

The other method is to use an energy balance,

$$_1q_3 = G[h_1 - h_3 - h_{w3}(W_1 - W_3)]$$

The initial humidity ratio is 0.0183 lb of water vapor per lb of dry air, and the final humidity ratio is 0.0158 lb of water vapor per lb of dry air. Therefore, the heat to be removed is

$$_1q_3 = \frac{20{,}000}{14.4} \times (43 - 34.1 - 0.0025 \times 38.07)$$

$$= 12{,}130 \text{ Btu per min.}$$

Adiabatic Mixing of Two Steady Flow Air Streams at Constant Pressure

The process is diagrammed in Fig. 10. By applying the principles of the conservation of mass and energy, three equations may be written:

Mass balance for the dry air,

$$G_1 + G_2 = G_3$$

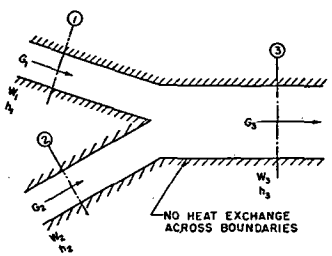


Fig. 10 .... Illustration of Mixing of Two Steady Flow Streams at Constant Pressure

Energy balance for the process,

$$G_1h_1 + G_2h_2 = G_3h_3$$

Mass balance for the water vapor,

$$G_1W_1 + G_2W_2 = G_3W_3$$

Eliminating G_3 and combining the three equations yields the equation,

$$\frac{h_2 - h_3}{h_3 - h_1} = \frac{W_2 - W_3}{W_3 - W_1} = \frac{G_1}{G_2} \tag{34}$$

Example 5: Outdoor air at 0 F dry-bulb temperature and 0.80 degree of saturation is to be mixed adiabatically with recirculated indoor air at 70 F dry-bulb temperature and 0.20 degree of saturation, in the ratio of one pound of dry air in the former to four in the latter. Find the temperature and degree of saturation in the resulting mixture.

Solution a: From the data of Table 2. The only unknown properties are the humidity ratio W_3 and the enthalpy h_3 of the resulting mixture. These may be determined from Equation 34. Thus,

$$\frac{20.270 - h_3}{h_3 - 0.668} = \frac{0.003164 - W_3}{W_3 - 0.000630} = \frac{1}{4}$$

from which $h_3 = 16.350$ and $W_3 = 0.002657$. The enthalpy of the final mixture may also be expressed by Equation 28:

$$h_3 = h_a + \mu h_{as}$$

Since μ by definition is W_3/W_s, Equation 28 may be rewritten as

$$16.350 = h_a + (0.002657/W_s) \times h_{as}$$

At 56 F the right side of the equation is 16.332, and at 57 F it is 16.582. Interpolation gives as the final dry-bulb temperature of the mixture 56.07 F. At this temperature the humidity ratio at saturation is 0.00960 lb of water vapor per lb of dry air. Therefore, the final degree of saturation is

$$\mu = 0.002657/0.00960 = 0.277$$

Solution b: From the ASHAE Chart. Equation 34 indicates that the state point of the resulting mixture lies on a straight line connecting the state points of the two streams being mixed, and divides this line into two segments whose respective lengths are inversely proportional to the rates of dry air flow in the corresponding streams. This is illustrated in Fig. 11. Points 1 and 2 are located and connected by a straight line. The state of the final mixture is set so that

$$\frac{G_1}{G_2} = \frac{D_{2-3}}{D_{3-1}} = \frac{1}{4}$$

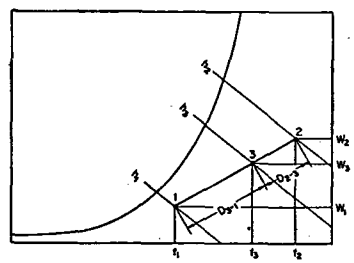


Fig. 11 .... Solution of *Example 5* on ASHAE Psychrometric Chart

Scaling the distances on the chart, the required solution to *Example 5* is 56 F dry-bulb temperature and 0.28 degree of saturation.

Addition of Moisture to an Adiabatic Stream

Consider a stream of moist air flowing adiabatically between two sections, 1 and 2, as in Fig. 12, with moisture addition at the rate $G_1(W_2 - W_1)$ and the moisture having the enthalpy h_w Btu per pound of moisture.

An energy balance yields

$$G_1h_1 + G_1(W_2 - W_1)h_w = G_1h_2 \tag{35}$$

Example 6: Liquid water chilled to 40 F is injected into an air stream initially at 95 F dry-bulb temperature and 80 F thermodynamic wet-bulb temperature. At what temperature will saturation be reached? How much water must be evaporated to reach saturation?

Solution a: From the data of Table 2. The solution of Equation 35 for h_2 yields

$$h_2 = h_1 + (W_2 - W_1)h_w$$

The initial enthalpy of the moist air h_1 must be found from Equation 8,

$$h_1 = h^* - (W^* - W_1)h_w^*$$

$$22.827 + \mu 40.49 = 43.69 - (0.02233 - 0.03673\mu) \quad (48.05)$$

from which $\mu = 0.511$.

Hence,

$$h_1 = 22.827 + 0.511(40.49)$$

$$= 43.52 \text{ Btu per lb of dry air.}$$

and

$$W_1 = 0.03673(0.511)$$

$$= 0.01877 \text{ lb per lb of dry air.}$$

The solution of Equation 35 is

$$h_2 = 43.52 + (W_2 - 0.01877)(8.09)$$

By trial and error, this equation will be satisfied at the temperature 79.87 F. At this temperature the humidity ratio W_2 is 0.02223. The weight of water evaporated is therefore $0.02223 - 0.01877 = 0.00346$ lb per lb of dry air.

Solution b: From the ASHAE Chart. Solution of Equation 35 for the ratio $(h_2 - h_1)/(W_2 - W_1)$ yields

$$\frac{h_2 - h_1}{W_2 - W_1} = h_w \tag{36}$$

The slope of the condition line is therefore determined by

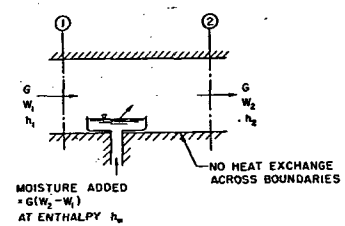


Fig. 12 .... Illustration of Addition of Moisture to an Adiabatic Stream

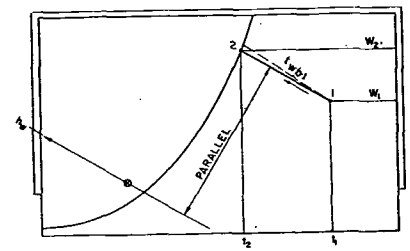


Fig. 13 .... Solution of *Example 6* on ASHAE Psychrometric Chart

the enthalpy of the water which is supplied. This slope is established on the chart by connecting the center of the protractor on the psychrometric chart with the value of h_w on the protractor. Draw a line parallel to this reference line through the initial Point 1 (Fig. 13). The second line is the condition line for the process. Since the conditions of the problem require the final point to lie on the saturation line, the intersection of the condition line with the saturation line gives the desired solution.

Adiabatic Saturation

Adiabatic saturation is the designation given any process in which the state of moist air is changed from some initial unsaturated condition to a saturated one without the addition or removal of heat. According to this definition, the addition of moisture to an adiabatic stream may become an adiabatic saturation process. *Example 6* is an illustration.

A type of adiabatic saturation of further practical interest is the use of continually recirculated spray water in a saturating air washer. Here the spray water will ultimately come to the same temperature as the saturated leaving air; this temperature is, by definition, the thermodynamic wet-bulb temperature. Hence, adiabatic saturation in this manner will have the final state point on the saturation curve, with the same thermodynamic wet-bulb temperature as the original state point.

In a process such as this, the moist air enthalpy changes very slightly. The moisture added and temperature change may be obtained from the psychrometric chart as sketched in Fig. 14.

Example 7: Moist air at 75 F dry-bulb temperature and 0.60 degree of saturation, is saturated adiabatically with recircu-

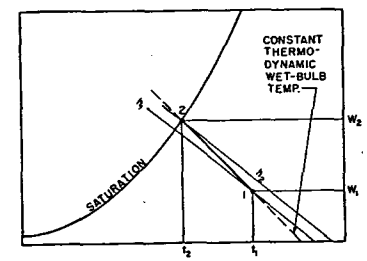


Fig. 14 .... Moisture Added and Temperature Change

lating spray water. Find the amount of water added and the change in enthalpy.

Solution a: From the data of Table 2. As previously stated, the final temperature of the mixture will be the thermodynamic wet-bulb temperature at the initial state. This must first be determined by the method of *Example 2*, and is 65.51 F. At this temperature the humidity ratio at saturation and enthalpy at saturation are 0.01350 and 30.45, respectively. The humidity ratio at the initial state is 0.60(0.01882) = 0.01129; the initial enthalpy is 0.60(20.59) + 18.018 = 30.372. The weight of water added is therefore 0.01350 − 0.01129 = 0.00221 lb per lb of dry air; the enthalpy change is 30.45 − 30.372 = 0.078 Btu per lb of dry air.

Solution b: From the ASHAE Chart. Since the initial and final states have the same thermodynamic wet-bulb temperature, the results may be read directly from the chart.

Addition of Heat and Water Vapor to an Air Stream in Steady Flow

Fig. 15 is a schematic representation of a system operating at constant pressure, *where*

G_1 is the rate of flow of dry air, pounds per minute.

G_w is the rate of evaporation of the water, pounds per minute.

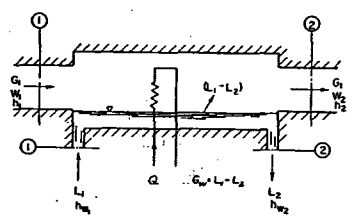


Fig. 15 .... Illustration of Addition of Heat and Water Vapor to an Air Stream in Steady Flow

h_w is the enthalpy of the liquid water entering, Btu per pound.

Q is the rate of heat addition, Btu per minute.

An energy balance for the system gives

$$G_1h_1 + Q + G_wh_w = G_1h_2 \tag{37}$$

A mass balance gives

$$G_1 + G_1W_1 + G_w = G_1 + G_1W_2$$

or

$$G_1W_1 + G_w = G_1W_2 \tag{38}$$

Combining Equations 37 and 38 and solving for the ratio $(h_2 - h_1)/(W_2 - W_1)$,

$$\frac{h_2 - h_1}{W_2 - W_1} = \frac{Q}{G_w} + h_w \tag{39}$$

Example 8: Moist air at 20 F dry-bulb temperature and 0.80 degree of saturation is heated and humidified until it is at 120 F dry-bulb temperature and 71.5 F thermodynamic wet-bulb temperature. Water at 55 F is supplied. If the air-flow rate is 20,000 cfm at the initial conditions, how much heat is required?

Solution a: From the data of Table 2. The initial humidity ratio is 0.80(0.002152) = 0.00172; the initial enthalpy is 4.804 + 0.80(2.302) = 6.6456; the initial specific volume is 12.084 + 0.80(0.042) = 12.118. The degree of saturation at the final

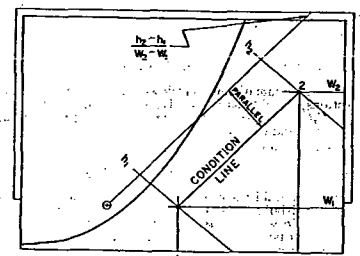


Fig. 16 .... Solution of *Example 8* on ASHAE Psychrometric Chart

state may be determined from Equation 8 which may be rewritten as

$$h_{as} + \mu h_{as_2} + h_w^*(W^* - \mu W_{s_2}) = h^*$$

The values of these properties are: $h^* = 35.39$; $W^* = 0.01668$; $h_w^* = 39.61$; $h_{as_2} = 90.70$; $W_{s_2} = 0.08149$; $h_{s_2} = 28.84$.

Making the proper substitutions and solving for degree of saturation,

$$\mu = 0.0681$$

The final humidity ratio is therefore 0.0681(0.08149) = 0.005549; the final enthalpy is 28.84 + 0.0681(90.70) = 35.02 Btu per lb dry air.

The rate of water addition is obtained from Equation 38.

$$G_w = \frac{20{,}000}{12.12}(0.005549 - 0.00172)$$

$$= 6.32 \text{ lb per min.}$$

The heat supplied is obtained from Equation 37.

$$Q = G_1(h_2 - h_1) - G_wh_w$$

$$= \frac{20{,}000}{12.12}(35.02 - 6.65) - 6.32(28.08)$$

$$= 46{,}667 \text{ Btu per min.}$$

Solution b: From the ASHAE Chart. Locate the initial and final states on the chart and connect them with a straight line. Through the reference point on the chart, draw a line parallel to the line connecting the initial and final state points, the condition line, and read the value of the ratio $(h_2 - h_1)/(W_2 - W_1)$ as 7500 from the protractor on the chart (Fig. 16). From Equation 39

$$\frac{h_2 - h_1}{W_2 - W_1} = \frac{Q}{G_w} + h_w = 7500$$

The rate of water supply was determined in *Solution a*, but will be found from the chart. It is

$$G_w = \frac{20{,}000}{12.1}(0.0055 - 0.0017)$$

$$= 6.28 \text{ lb per min}$$

$$Q = G_w(7500 - h_w)$$

$$= 6.28(7500 - 23)$$

$$= 46{,}900 \text{ Btu per min.}$$

U. S. STANDARD ATMOSPHERE

The definition of the U. S. Standard Atmosphere is important to the air-conditioning engineer as an essential standard of reference. The basic assumptions in defining the Standard Atmosphere are:

1. There is a linear decrease in temperature T with altitude up to the limit of the isothermal atmosphere at 35,332 ft. Thus,

$$T = T_s - 0.003566\,Z \tag{40}$$

2. The air is dry.
3. Air is a perfect gas obeying the laws of Charles and Boyle:

$$PV = RT$$

4. Gravity is constant at all altitudes with the standard value.
5. The temperature of the isothermal atmosphere is −66 F.

Table 6 .... Pressure and Temperature for Altitudes in U. S. Standard Atmosphere

Altitude Feet Z	Pressure In. of Hg P	Temp F t
−1,000	31.02	+62.6
−500	30.47	+60.8
0	29.921	+59.0
+500	29.38	+57.2
+1,000	28.86	+55.4
+5,000	24.89	+41.2
10,000	20.58	+23.4
15,000	16.88	+5.5
20,000	13.75	−12.3
25,000	11.10	−30.1
30,000	8.88	−47.9
35,000	7.04	−65.8
40,000	5.54	−67.0
45,000	4.36	−67.0
50,000	3.436	−67.0

Standard values at sea level, which are part of the definition of the Standard Atmosphere, are:

Pressure	29.921 in. Hg
Temperature	59 F
Absolute Temperature	518.67 F abs
Gravity	32.1740 ft per (sec) (sec)
Density	0.076505 lb per cu ft

Values of pressure and temperature are listed in Table 6 for altitudes in the standard atmosphere from −1000 to 50,000 ft above sea level. For further explanation, References 9 and 10 should be consulted.

LETTER SYMBOLS USED IN CHAPTER 3

μ = degree of saturation (decimal).
ρ = density of fluid, pounds per cubic foot.
ϕ = relative humidity (decimal).
a = ratio of apparent molecular weight of dry air (28.966) to the molecular weight of water (18.016) = 1.6078.
A = coefficient from Table 5 for use in Equation 30 (obtained from Table 5).
B = coefficient to be used in Equation 31 (obtained from Table 5).
C = coefficient for use in Equation 32 (obtained from Table 5).
f_s = factor accounting for effect of mixing air and water, dimensionless.
G = flow rate of dry air, pounds per hour.
G_1 = flow rate of dry air, pounds per minute.
G_w = rate of evaporation of water, pounds per minute.
H = enthalpy of the system.
H = enthalpy of the flowing medium, Btu per pound of dry air.
h = enthalpy of moist air, Btu per pound of dry air.
$\bar{h}$ = enthalpy correction term to be added above 150 F, to enthalpy.
h_a = specific enthalpy of dry air, Btu per pound.
$h_{as} = h_s - h_a$ = the difference between the enthalpy of moist air *at saturation* per pound of dry air, and the specific enthalpy of the dry air itself, Btu per pound of dry air.
h_g = enthalpy of saturated water vapor, Btu per pound.
h_s = enthalpy of moist air at saturation per pound of dry air, Btu per pound of dry air.
$h_s{}^*$ = enthalpy of moist air at saturation at *thermodynamic wet-bulb temperature* $t_s{}^*$ Btu per pound of dry air.
h_w = specific enthalpy of condensed water (liquid or solid) at standard pressure, Btu per pound water.
$h_w{}^*$ = specific enthalpy of water as added at the *thermodynamic wet-bulb temperature* $t_s{}^*$ Btu per pound of dry air.
h_{wl} = enthalpy of liquid water, Btu per pound.
h_{ws} = enthalpy of solid water, Btu per pound.
KE = kinetic energy, Btu per pound.
$\overline{KE}$ = average kinetic energy, Btu per pound.
L = flow rate of liquid water, pounds per hour.
m = weight of dry air crossing any duct section, pounds per minute.
n_a = mols dry air.
n_s = mols of water vapor at saturation.
n_w = mols of water vapor.
P = absolute pressure.
P_a = atmospheric pressure, inches Hg.
P_s = standard atmospheric pressure by definition 29.921 in. Hg.
P_s = saturation pressure of pure water at prevailing temperature.
p = total pressure of a mixture of air and water vapor, pounds per square inch or inches Hg.
p_a = partial pressure of dry air, pounds per square inch or inches Hg.
p_s = saturation pressure of pure water vapor, pounds per square inch or inches Hg.
p_w = partial pressure of water vapor in mixture of air and water vapor, pounds per square inch or inches Hg.
PE = potential energy, Btu per pound dry air.
$\overline{PE}$ = average potential energy, Btu per pound dry air.
Q = total heat added or subtracted, between sections, Btu per minute.
q = ratio of energy added (or removed) to water added (or removed), Btu per pound. Also called specific enthalpy of *water added*.
$_1q_2$ = energy added between points 1 and 2.
$_1q_2$ = heat added between sections 1 and 2, Btu per pound dry air.
R = universal gas constant, 1545 foot-pounds per (Fahrenheit degree) (mol).
R_a = gas constant for dry air.
R_w = gas constant for water vapor.
S = flow rate of solid water, pounds per hour.
s = entropy of moist air per pound of dry air, Btu per (pound) (Fahrenheit degree).

LETTER SYMBOLS (Continued)

$\bar{s}$ = correction to be added to entropy of moist air obtained from Table 2.
$\bar{\bar{s}}$ = additional correction to be added to entropy because of "mixing entropy" (obtained from Table 5). Correction to be added to value of s obtained from Table 2.
s_a = specific entropy of dry air, Btu per (pound) (Fahrenheit degree, absolute).
s_{as} = the difference between the entropy of moist air *at saturation* per pound of dry air, and the specific entropy of the dry air itself, Btu per (pound of dry air) (Fahrenheit degree, absolute).
s_s = entropy of moist air at saturation per pound of dry air, Btu per (pound of dry air) (Fahrenheit degree, absolute).
s_w = specific entropy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per (pound of water) (Fahrenheit degree, absolute).
T = absolute temperature, Fahrenheit degrees.
T_a = standard atmospheric temperature, by definition 518.4 F absolute.
t = temperature, Fahrenheit degrees.
t^* = thermodynamic wet-bulb temperature, Fahrenheit degrees.
U = internal energy of system.
U = internal energy.
V = volume.
$\overline{V}$ = average velocity, feet per minute.
v = volume of moist air per pound of dry air, cubic feet per pound.
$\bar{v}$ = correction to be added to volume of moist air per pound of dry air, above 150 F.
v_a = specific volume of dry air, cubic feet per pound.
$v_{as} = v_s - v_a$, the difference between volume of moist air *at saturation*, per pound of dry air, and the volume of the dry air itself, cubic feet per pound of dry air.
v_s = volume of moist air at saturation per pound of dry air, cubic feet per pound of dry air.
v_T = total volume, cubic feet.

LETTER SYMBOLS (Concluded)

W = humidity ratio of moist air, pounds of water per pound of dry air.
W_s = humidity ratio, *at saturation*, weight of water vapor per pound of dry air, pound per pound.
W_s^* = *humidity ratio* corresponding to *thermodynamic wet-bulb temperature* t,* pounds of water per pound of dry air.
w = work done by system.
w = shaft work withdrawn between sections 1 and 2, Btu per pound of air.
Z = elevation above any datum, feet.
$\bar{Z}$ = average elevation, feet.

Subscripts with symbols have following meanings: 1, 2, 3 indicate section of flow; a = air, w = water, wl = liquid water, ws = solid water, s = saturation, m = mixture; * indicates that the value is at thermodynamic wet-bulb temperature.

REFERENCES

[1] J. A. Goff and S. Gratch: The humidity of ratio of moist air at saturation (University of Pennsylvania, *Thermodynamic Research Laboratory Special Report*, March 1948).
[2] J. A. Goff: Standardization of thermodynamic properties of moist air (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 459).
[3] J. H. Arnold: The theory of the psychrometer (*Physics*, Vol. 4, 1933).
[4] J. A. Goff and S. Gratch: Thermodyuamic properties of moist air (ASHVE TRANSACTIONS, Vol. 51, 1945, p. 125).
[5] J. A. Goff and S. Gratch: Low pressure properties of water in the range—160 to 212 F (ASHVE TRANSACTIONS, Vol. 52, 1946, p. 95).
[6] J. H. Keenan and F. G. Keyes: *Thermodynamic Properties of Steam* (John Wiley and Sons, New York, 1936).
[7] R. Mollier: Ein neues diagramm für dampfluftgemische (*ZVDI*, Vol. 67, September 8, 1923, p. 869).
[8] R. Mollier: Das i-x diagramm für dampfluftgemische (*ZVDI*, Vol. 73, July 20, 1929, p. 1009).
[9] *National Advisory Committee for Aeronautics Technical Report* No. 218 (1925).
[10] *National Advisory Committee for Aeronautics Technical Report* No. 538 (1935).

CHAPTER 4

FLUID FLOW

Theory of Fluid Flow, Pressure Loss in Circular Pipes, Pressure Loss in Non-Circular Pipes; Flow of Compressible Fluids, Ideal Flow Through Nozzle or Orifice; Flow Measurement, Head Meters, Location of Head Meters and Pressure Taps Pitot Tube, Variable Area Flow Meters

THE flow of fluids which is part of the branch of engineering science known as fluid mechanics will be discussed here insofar as it applies to the work of engineers in the fields of heating, ventilating, and air conditioning. Probably air is the most frequently handled fluid, but other gases and liquids are often involved. Compressible fluids (gases) and incompressible fluids (liquids) vary somewhat in behavior, though in cases where pressure and density changes are small, the gases may be treated as incompressible fluids.

THEORY OF FLUID FLOW

The following energy equation for one dimensional steady flow processes will serve as a basis for the theory of the flow of fluids. This equation is presented in several ways in various texts, but a suitable form is

$$\frac{V_1^2}{2g_c} + Ju_1 + p_1v_1 + Jq + \frac{g}{g_c}z_1 = \frac{V_2^2}{2g_c} + Ju_2 + p_2v_2 + W + \frac{g}{g_c}z_2 \quad (1)$$

where

V = velocity, feet per second.
g = gravitational acceleration, feet per (second) (second).
g_c = gravitational conversion factor = 32.174 (pounds mass per pound force) × ft per (second) (second).
J = mechanical equivalent of heat = 778 foot pounds per Btu.
u = internal energy, Btu per pound of fluid.
p = pressure per square foot, pounds.
v = specific volume, cubic feet per pound.
W = mechanical work done by the fluid, foot pounds per pound of fluid.
q = heat transferred to the fluid, Btu per pound of fluid flowing.
z = elevation above some arbitrary datum, feet.

Subscript 1 refers to the entrance, subscript 2 to the exit.

Introducing the enthalpy h, which by definition is $u + \frac{pv}{J}$, expressed in Btu per pound of fluid, Equation 1 becomes

$$\frac{V_1^2}{2g_c} + Jh_1 + Jq + \frac{g}{g_c}z_1 = \frac{V_2^2}{2g_c} + Jh_2 + W + \frac{g}{g_c}z_2 \quad (2)$$

The equivalent differential form for energy Equation 1 is

$$\frac{1}{2g_c}dV^2 + J\,du + d(pv)\frac{g}{g_c}dz - J\,dq + dW = 0 \quad (3)$$

Replacing v by its equal $g/g_c\rho$ (where ρ is density in pounds weight per cubic foot) and rearranging, Equation 3 becomes

$$\frac{1}{2g}dV^2 + \frac{1}{\rho}dp + dz + \frac{g_c}{g}\{J\,du + p\,dv - J\,dq + dW\} = 0 \quad (4)$$

In the case of flow through a pipe, no outside work is performed so that $dW = 0$. Furthermore,

$$J\,du + p\,dv = JT\,ds = J\,dq + JT\,ds' \quad (5)$$

where

ds = total change in entropy.
ds' = change in entropy due to internal irreversibility from turbulence and friction.

Accordingly, Equation 4 may be written

$$\frac{1}{2g}dV^2 + \frac{dp}{\rho} + dz + \frac{g_c}{g}JT\,ds' = 0^* \quad (6)$$

In cases where there is no internal irreversibility, $ds' = 0$, and Equation 6 may be integrated to give

$$\frac{V_1^2}{2g} + \frac{p_1}{\rho_m} + z_1 = \frac{V_2^2}{2g} + \frac{p_2}{\rho_m} + z_2 \quad (7)$$

where

ρ_m = proper mean density.

This is commonly called the Bernoulli equation, named after the Swiss mathematician and physician who first propounded

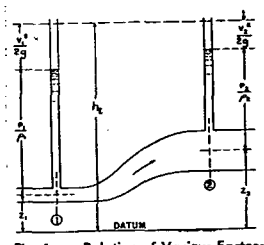


Fig. 1 .... Relation of Various Factors in Bernoulli Equation

* In the analysis of subsequent portions of this chapter the distinction between g and g_c will be omitted. Aside from the dimensional consistency the factor, g/g_c, is not in general significant in fluid flow analysis.

the theory. $\frac{V^2}{2g}$ is known as the velocity head, $\frac{p}{\rho}$ is the pressure head, and z is the elevation head, all in feet of the fluid; the total head, h_t is the sum of the other three heads. Fig. 1 shows diagrammatically the relation of the various factors. The pressure at point 2 is lower than at point 1 because of the elevation of point 2 over point 1, and the velocity at point 2 is lower than at point 1 because of the larger pipe diameter at point 2. If the pipe diameter were the same throughout, the velocity, and consequently the velocity head, would be the same at both points, but the higher elevation at point 2 would still be responsible for a loss in pressure. The utility of the equation is evident, though it should be remembered that in it the *effects of friction and turbulence are neglected,* and that Fig. 1 represents ideal conditions. It should also be noted that care must be taken in determining the proper mean density. Accordingly, the Bernoulli equation is applied most conveniently to incompressible fluids for which density is constant.

Pressure Loss in Circular Pipes

The pressure loss in circular pipes is customarily expressed by the formula:

$$h_f = \frac{flV^2}{2g\,d} \qquad (8)$$

where

h_f = the loss in head of the fluid under conditions of flow, feet.
l = the length of the pipe, feet.
V = the velocity, feet per second.
g = the acceleration due to gravity = 32.174 ft per (second) (second).
d = the internal diameter of the pipe, feet.
f = a dimensionless friction coefficient.

The formula is generally known by the name of Darcy or Fanning, though it seems to have been originated by d'Aubisson de Voisins in 1834.

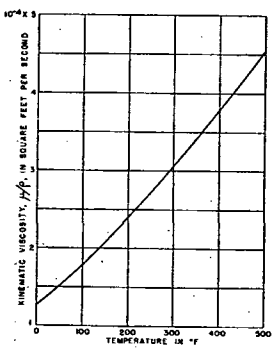


Fig. 2 .... Relation of Kinematic Viscosity to Temperature of Air

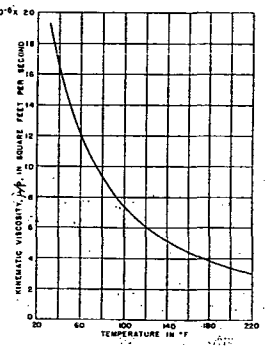


Fig. 3 .... Relation of Kinematic Viscosity to Temperature of Water

The factor f is a function of the Reynolds number,

$$N_{Re} = \frac{d\,V\rho}{\mu} \qquad (9)$$

where

N_{Re} = Reynolds number.
ρ = the density, pounds per cubic foot.
μ = the absolute viscosity, pounds per foot-second.

Both f and the Reynolds number are dimensionless. To aid in computing the Reynolds number, values of $\frac{\mu}{\rho}$, the kinematic viscosity, are shown as a function of temperature for air in Fig. 2, and for water in Fig. 3.

Fig. 4 shows the relation between f and the Reynolds number, adapted from a review by Moody.[1] The straight line sloping downward at the left of the chart supplies the values of f for laminar flow determined by the formula:

$$f = \frac{64}{N_{Re}} \qquad (10)$$

With laminar flow, the velocity profile is a parabola, having the formula:

$$V = \frac{\rho h_f}{4\mu l}(r^2 - L^2) \qquad (11)$$

where

r = the radius of the pipe, feet.
L = distance perpendicularly from the axis of the pipe, feet.

Accordingly, the maximum velocity occurs at the center of the pipe and is twice the average velocity; the average velocity is found when $L = 0.707\,r$. It is worth noting that roughness of the pipe wall has no effect on the loss in head for laminar flow.

Between values of the Reynolds number of 2000 and 4000,

[1] Superior numbers refer to the references at the end of chapter.

there is an unstable region where the flow changes from laminar to turbulent, or vice versa. The actual value is impossible of prediction for any conditions of flow, though in general it may be said that the prevailing type of flow persists into the unstable region; however, once a change starts, it proceeds very rapidly.

When the flow is turbulent, the velocity profile is essentially parabolic over four-fifths of the pipe diameter, but near the pipe walls, the effect of friction becomes evident, and in the boundary layer at the pipe wall the flow is laminar. Fig. 5 compares the velocity profiles for three different Reynolds numbers, but for the same average velocity.

The lower curve in the turbulent region in Fig. 4 represents the relation of f to the Reynolds number for smooth pipe, such as drawn brass tubing or glass tubing. The effect of roughness on f, an effect which is considerable in turbulent flow, is open to some conjecture; artificially roughened pipes, for instance, give results at variance with actual tests. The curves above the smooth pipe curve of Fig. 4 represent a summary of tests on rough pipe, each of them identified by a value of e/d, with e signifying the absolute roughness in feet. Values of e for different pipes are given in Table 1.

To find the friction loss for any pipe, follow the curve with the proper value of e/d, to the pertinent value of N_{Re}, and from this point proceed horizontally to left margin to find the value of f for use in Equation 8.

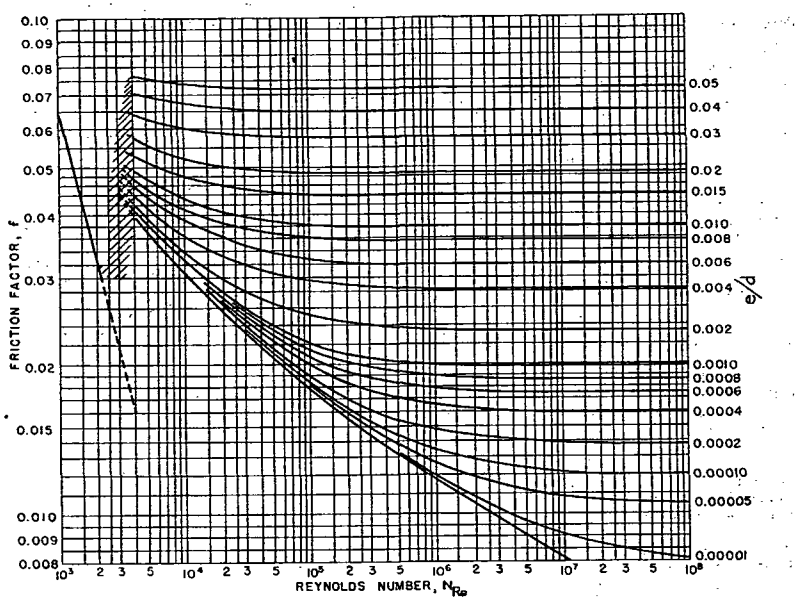
Table 1 .... Values of e for Different Kinds of Pipe

Type of Pipe	e
Smooth drawn tubing	0.000005
Commercial steel or wrought iron	0.00015
Asphalted cast-iron	0.0004
Galvanized iron	0.0005
Cast-iron	0.00085
Wood stave	0.0006 to 0.003
Concrete	0.001 to 0.01
Riveted steel	0.003 to 0.03

The curves in Fig. 4 may be approximated very closely by the empirical formula:[1]

$$f = 0.0055\left[1 + \left(20{,}000\,\frac{e}{d} + \frac{10^6}{N_{Re}}\right)^{1/3}\right] \qquad (12)$$

Equation 8 is applicable to all liquids, and to gases when



Note: The straight line at left shows values of Friction Factor for laminar flow.

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Fig. 4 .... Relation Between Friction Factor and Reynolds Number

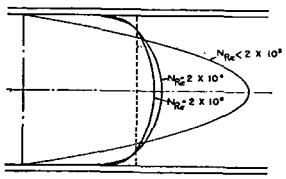


Fig. 5 .... Comparison of Velocity Profiles for 3 Different Reynolds Numbers but for Same Average Velocity

the pressure loss is less than 10 percent of the initial pressure. When the loss in head is high, the formula to be used for gases is

$$\frac{p_1^2 - p_2^2}{p_1^2} = \frac{flV_1^2}{gd\,p_1v_1} \tag{13}$$

which may be arranged to give the loss in pressure,

$$p_1 - p_2 = p_1\left[1 - \sqrt{1 - \frac{flV_1^2}{gd\,p_1v_1}}\right] \tag{14}$$

Pressure Loss in Non-Circular Pipes

The formulas for friction loss in pipes are based on the use of pipes of circular cross-section. The same formulas may be extended to noncircular sections, by suitable modification. In the basic formula, Equation 8, the internal diameter d is to be replaced by the hydraulic diameter d_H defined by the equation:

$$d_H = \frac{4 \times \text{area of cross-section}}{\text{wetted perimeter of cross-section}} \tag{15}$$

For example, in a rectangular duct, 1 ft by 2 ft, the cross-section area is 2 sq ft, and the perimeter 6 ft. Then the hydraulic diameter will be $d_H = (4 \times 2)/6 = 1\frac{1}{3}$ ft.

In the case of a round pipe,

$$d_H = \frac{4 \times \pi d^2/4}{\pi d} = d \tag{16}$$

In computing the Reynolds number, and from that the friction factor, the hydraulic diameter is not to be used. A better approximate procedure is to replace the length in the Reynolds number by the shortest dimension plus one-fourth of the hydraulic diameter. Thus, in a duct of dimension a x b where a < b, N_{Re}, for the purposes of calculating friction factors, is

$$N_{Re} = (a + 0.25d_H)V\rho/\mu \tag{17}$$

This value of N_{Re} may be used in Equation 10 for laminar flow, and in Equation 12 or Fig. 4 for turbulent flow. The error in the approximation is somewhat greater for laminar than for turbulent flow. In the former case, the relative error may be as much as 10 percent, while in the latter it almost always is less than 3 percent.

FLOW OF COMPRESSIBLE FLUIDS

In the flow of compressible fluids, the large density variations make it impracticable to use the Bernoulli equation, (Equation 7). In certain special cases, however, the exact equations for compressible flow may be stated. If flow occurs with no friction or other internal irreversibility Equation 6 becomes

$$\frac{1}{2g}dV^2 + \frac{dp}{\rho} = 0 \tag{18}$$

If, in addition, the flow is adiabatic,

$$pp^{-k} = p_1v_1^{-k} \tag{19}$$

so that Equation 18 becomes

$$\frac{1}{2g}dV^2 + \frac{p_1^{1/k}}{\rho_1}\frac{dp}{p^{1/k}} = 0 \tag{20}$$

or by integration,

$$\frac{1}{2g}(V_2^2 - V_1^2) + \frac{k}{k-1}\frac{p_1}{\rho_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right] = 0 \tag{21}$$

This extension to compressible flow of Bernoulli's equation reduces to the more familiar form if the pressure change is small.

The ratio of specific heats, k, is used extensively in fluid dynamics; values of k for various gases are given in Table 2.

It is convenient in the analysis of compressible flow to introduce the velocity of propagation of pressure impulses or, more familiarly, the sonic velocity, a. For perfect gases this is given by the equation:

$$a^2 = kgp/\rho = kgRT \tag{22}$$

Accordingly, Equation 21 may be written

$$\tfrac{1}{2}(V_2^2 - V_1^2) + \frac{a_1^2}{k-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right] = 0 \tag{23}$$

or, by rearrangement,

$$\frac{p_2}{p_1} = \left[1 - \frac{k-1}{a_1}\left(\frac{V_2^2 - V_1^2}{2}\right)\right]^{\frac{k}{k-1}} \tag{24}$$

which permits the calculation of the ratio of pressures at entrance and exit of the steady flow device—pipe, orifice, or nozzle. From Equations 19 and 22 it follows that

$$\frac{a_2^2}{a_1^2} = \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} \tag{25}$$

so that

$$\frac{k-1}{2}(V_2^2 - V_1^2) + a_2^2 - a_1^2 = 0 \tag{26}$$

and

$$\frac{a_2^2}{a_1^2} = \frac{1 + \frac{k-1}{2}\frac{V_1^2}{a_1^2}}{1 + \frac{k-1}{2}\frac{V_2^2}{a_2^2}} \tag{27}$$

The ratio of flow velocity to sonic velocity is known as the *Mach number*,

$$M = V/a$$

This parameter is particularly useful in compressible flow analysis. In general, if $M \leq 0.1$ the flow may be considered

to be incompressible. This is generally true in heating and ventilating air ducts.

In terms of the Mach number

$$\frac{a_2^2}{a_1^2} = \frac{T_2}{T_1} = \frac{1 + \frac{k-1}{2}M_1^2}{1 + \frac{k-1}{2}M_2^2} \tag{28}$$

and

$$\frac{p_2}{p_1} = \left(\frac{1 + \frac{k-1}{2}M_1^2}{1 + \frac{k-1}{2}M_2^2}\right)^{\frac{k}{k-1}} \tag{29}$$

The quantity,

$$p^0 = p\left(1 + \frac{k-1}{2}M_1^2\right)^{\frac{k}{k-1}} \tag{30}$$

is called the stagnation pressure, and gives a measure of pressure energy. For incompressible flow

$$p^0 = p + \frac{1}{2g}\rho V^2 = p + q \tag{31}$$

where

$$q = \frac{1}{2g}\rho V^2 \tag{32}$$

and is the dynamic pressure. A total head tube measures stagnation pressure directly.

From Equation 29 it follows that for frictionless, adiabatic flow

$$p_2^0 = p_1^0 \tag{33}$$

This represents another extension of the Bernoulli equation to compressible flow. Friction will cause a loss in pressure energy.

Ideal Flow through Nozzle or Orifice

The majority of low-head measuring systems depend upon a correlation between pressure drop, area, and quantity of flow. The basic formulas may be stated on the assumption that the flow is frictionless and adiabatic. Designating the main stream by station 1, and flow at some measuring restriction by station 2, the flow in pounds per second is

$$w = \rho_1A_1V_1 = \rho_2A_2V_2 \tag{34}$$

or, in terms of *Mach number*,

$$w = A_2M_2\sqrt{kgp_2v_2} \tag{35}$$

According to Equation 29

$$\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} = \frac{1 + \frac{k-1}{2}M_2^2}{1 + \frac{k-1}{2}M_1^2} \tag{36}$$

from which

$$M_2^2 = \frac{2}{k-1}\left(\frac{1 + \frac{k-1}{2}M_1^2}{1 - M_1^2/M_2^2}\right)\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right] \tag{37}$$

Table 2 .... Ratio of Specific Heat at Constant Pressure to Specific Heat at Constant Volume for Compressible Fluids

Compressible Fluid	Ratio $k = c_p/c_v$
Helium and other monatomic gases	1.66
Air and other diatomic gases	1.40
Ammonia and hydrogen sulfide	1.34
Carbon dioxide, methane, natural gas, superheated steam, moist steam down to a quality of 97 percent	1.28 to 1.32
Sulfur dioxide, ethylene, acetylene	1.24 to 1.26

so that

$$w = A_2\sqrt{\frac{1 + \frac{k-1}{2}M_1^2}{1 - M_1^2/M_2^2}}\sqrt{\frac{2kgp_2v_2}{k-1}\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right]} \tag{38}$$

If the initial velocity is sufficiently small, M_1^2 will be negligible so that

$$w = A_2\sqrt{\frac{2kgp_2v_2}{k-1}\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right]}$$

$$= A_2\sqrt{\frac{2kgp_1v_1}{k-1}\left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right]} \tag{39}$$

If this is computed and the figures are plotted, the curved line (partly solid and partly broken) of Fig. 6 is found. The maximum value of $\frac{p_2}{p_1}$ may be computed by differentiating w with respect to p_2 and equating the result to zero. This operation produces the equation:

$$\frac{p_2}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{40}$$

For air, with $k = 1.40$, $\frac{p_2}{p_1} = 0.53$.

Actually, the broken part of the curve is not attained for

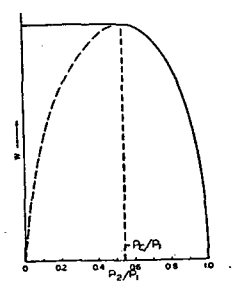
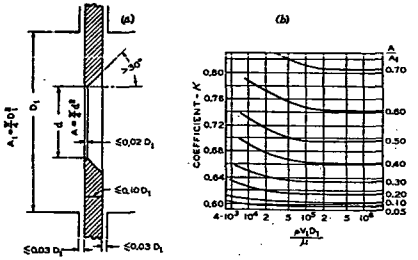


Fig. 6 .... Relation of Flow of Gas to Pressure Drop in a Converging Tube



(Coefficient shown as a function of Reynolds Number and Ratio, A/A_1.)
From Reference 4. Used by permission.

Fig. 7 .... Dimensions and Flow Coefficient for Standard Square-edged Orifice with Corner Taps or Annular Slits

the flow in the nozzle. If the ratio of p_2 to p_1 is decreased from unity, the mass rate of discharge, as well as the volume, increases from zero to a maximum, as shown by the solid section of the curve in Fig. 6; thereafter, as p_2/p_1 is decreased further, the discharge is constant, as indicated by the horizontal line. The value of p_2 at the maximum point is called the critical pressure, or p_c, and it is seen that p_c is approximately 53 percent of p_1 when air is flowing.

To find the velocity at the critical pressure, it is assumed that the upstream velocity V_1 is so small as to be negligible. Using the subscript c to indicate conditions at the critical point, from Equation 29

$$\frac{p_c}{p_1} = \frac{1}{\left(1 + \frac{k-1}{2} M_c^2\right)^{\frac{k}{k-1}}} \tag{41}$$

or

$$M_c = \sqrt{\frac{2}{k-1}\left[\left(\frac{p_1}{p_c}\right)^{\frac{k-1}{k}} - 1\right]} \tag{42}$$

Substituting the critical pressure ratio from Equation 40 it follows that

$$M_c = 1 \tag{43}$$

or that the velocity at the throat is equal to the local sonic velocity.

In developing the working equations for orifices and nozzles, it is customary to start with the incompressible form of the flow Equation 38. In this case both M_1 and M_2 are small quantities, $\rho_1 = \rho_2$, and $(p_1 - p_2)/p_2 = \Delta p/p_2$ is small. Retaining only first order terms, it follows from Equation 35 that

$$\frac{M_1}{M_2} = \frac{A_2}{A_1} \tag{44}$$

so that

$$\sqrt{\frac{1 + \frac{k-1}{2} M_1^2}{1 - (M_1^2/M_2^2)}} \cong \frac{1}{\sqrt{1 - (A_2/A_1)^2}} = \frac{1}{\sqrt{1 - \beta^4}} \tag{45}$$

where $\beta = D_2/D_1$. The quantity $1/\sqrt{1-\beta^4}$ is the velocity of approach factor as generally used, with β being the ratio of the throat or orifice diameter to the pipe diameter.

Since $\Delta p/p_2$ is small

$$\frac{2k}{k-1}\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right] \cong \frac{2\Delta p}{p_2} \tag{46}$$

and the mass flow is

$$w = \frac{A_2}{\sqrt{1-\beta^4}}\sqrt{2g\rho\Delta p} \tag{47}$$

The volume flow is then

$$Q = A_2 \frac{1}{\sqrt{1-\beta^4}}\sqrt{2g\Delta p/\rho} = A_2 \frac{1}{\sqrt{1-\beta^4}}\sqrt{2gh_f} \tag{48}$$

FLOW MEASUREMENT

The measurement of quantity of flow of fluids is generally accomplished either by making observations on the change of state due to flow system configuration, such as the pressure drop across a metering orifice, or by the displacement of some device, such as a rotating vane system. The selection of the metering system will be determined by the type of fluid, the precision of measurement desired, the cost of equipment and installation, the range of flow quantity, the ease of maintenance, and the method of observation and recording.

Fluid meters may be classified as follows: (1) *Head Meters* (Pressure Sensitive), which may be of *Venturi*, *flow nozzle*, *orifice plate*, or *Pitot tube* types; (2) *Area Meters*, which may be of *gate*, *tapered tube*, or *tapered plug* types; (3) *Force Meters*, which may be of *vane*, *propellor*, or *turbine* types; (4) *Quantity Meters*, which may be of *weighing tank*, *reciprocating piston*, or *geared impeller* types.

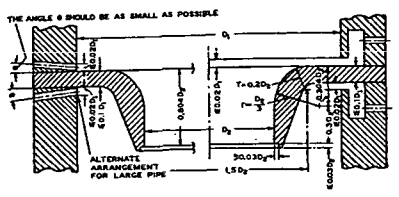
Rate meters are generally of the first three groups, although rates are obtainable from timed observations of quantity meters. Similar quantity measurements can be obtained by suitable integration of rate meter indications.

Head Meters

The head meter is of sufficient flexibility so that almost any type of flow measurement can be handled. For this reason, standard and reference measurements are usually made in this way. With proper care, extreme precision can be obtained. Also, an inexpensive installation can be made to give moderate precision. In general, a head meter requires fairly competent installation and maintenance to give satisfactory service.

Among the types of head meters, the Venturi has the advantage of having a low pressure loss, but requires considerable length of space. The orifice plate reduces the required length to a minimum at the cost of considerable pressure drop. The flow nozzle represents a compromise, but must be calibrated individually, or made with extreme care from standard specifications, to obtain a good precision of measurement. Standard configurations, for representative orifice plates and flow nozzles, are shown in Figs. 7 and 8, respectively. Additional specifications are given in References 2, 3, and 4 at end of chapter.

The measurement of flow in head meters is dependent



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Fig. 8 .... Dimensions for International Standards Association Flow Nozzle

upon observations of a static pressure difference between two parts of the system. Standard locations for these static pressure measurements are illustrated in Figs 8 and 16. In general, there exists a reduction in area, either by a smooth contour as in the case of a Venturi or flow nozzle, or by a vena contracta following an orifice. In either case, the flow will be given by a suitable modification of Equations 47 and 48. Adding a correction term, these become

$$w = \frac{A_2 C}{\sqrt{1-\beta^4}}\sqrt{2g\rho\Delta p} \tag{49}$$

and

$$Q = \frac{A_2 C}{\sqrt{1-\beta^4}}\sqrt{2gh_f} \tag{50}$$

In all cases, A_2 refers to the minimum area of the Venturi, nozzle, or orifice. The correction factor C is introduced to account for any loss due to departure from isentropic flow, and for any deviation between the downstream measured pressure and the actual pressure at the minimum section. Since these corrections are usually dependent on configuration, as described by the ratio of upstream to minimum area A_1/A_2, it is often convenient to introduce a combined flow coefficient, K which is

$$K = \frac{C}{\sqrt{1-\beta^4}} \tag{51}$$

so that

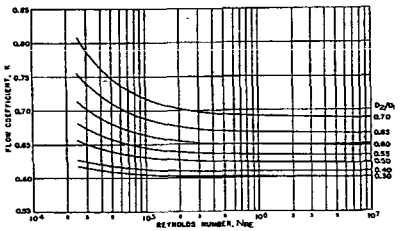
$$w = KA_2\sqrt{2g\rho\Delta p} \tag{52}$$

and

$$Q = KA_2\sqrt{2gh_f} \tag{53}$$

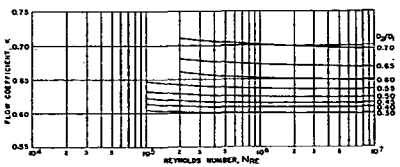
The coefficients K and C will be determined by the area ratio A_1/A_2, or by the diameter ratio D_1/D_2 and the Reynolds number. Extensive data are available from various sources.[3, 4] Figs. 9, 10, 11, 12 show representative values of K based on N_{Re}, the Reynolds number, and the ratio of diameter of the orifice or nozzle throat to pipe diameter.

In any specific application, numerical values for areas, densities, and pressure units can be substituted in Equations 52 and 53 to obtain compact working formulas. In planning flow measurements, standard flow meter dimensions and standard pressure tap locations should preferably be used so



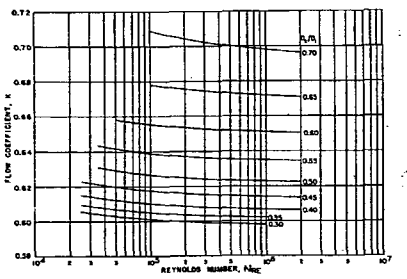
From Table 6 of Reference 2.

Fig. 9 .... Flow Coefficients, K, for Square-edged Orifice Plates and Flange Taps in Smooth Pipe



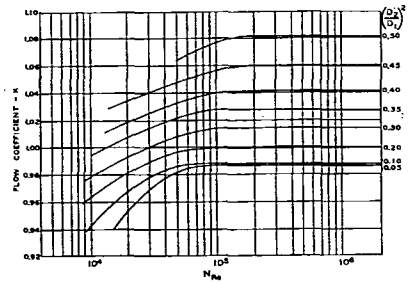
From Fig. 36d of Reference 3.

Fig. 10 .... Flow Coefficients, K, for Square-edged Orifice Plates and Radius Taps in Smooth Pipe



From Table 7 of Reference 2.

Fig. 11.... Flow Coefficients, K, for Square-edged Orifice Plates and Vena Contracta Taps, in Smooth Pipe



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Fig. 12 .... Flow Coefficient for International Standards Association Flow Nozzle Shown in Fig. 8 as a Function of Area Ratio $(D_2/D_1)^2$ and the Reynolds Number N_{Re}

that published flow coefficients can be used for accurate determination of flow rates.

In measuring the flow of compressible fluids, the approximate Equations 52 and 53 must be corrected for density variations. The need for such correction is evident by comparing Equations 35 and 38 with Equation 47. Introducing a multiplicative correction factor ϕ the equation for flow with no loss becomes:

$$w = \frac{A_2\phi}{\sqrt{1-\beta^4}}\sqrt{2g\rho_1(p_1-p_2)} \tag{54}$$

By comparison with Equation 35,

$$\phi = \sqrt{\frac{k}{2}\frac{p_2}{p_1}\frac{M_2^2(1-\beta^4)}{(p_1/p_2)-1}} \tag{55}$$

From Equations 34 and 35,

$$M_1^2/M_2^2 = \beta^4(p_2/p_1)^{2/k} \tag{56}$$

Hence, by Equation 38 for a small value of M_1,

$$\phi = \sqrt{(p_2/p_1)^{1/k}\left(\frac{k}{k-1}\right)\frac{(p_1/p_2)^{(k-1)/k}-1}{(p_1/p_2)-1}\cdot\frac{1-\beta^4}{1-\beta^4(p_2/p_1)^{2/k}}} \tag{57}$$

From Equations 37 and 56, if M_2 is not too large,

$$\frac{2}{k}\frac{1}{M_2^2}\left(\frac{p_1}{p_2}-1\right) \cong 1-\beta^4+\tfrac{1}{4}[1-2(k-2)\beta^4]M_2^2 \tag{58}$$

so that, approximately,

$$\phi \cong -\frac{3}{8}\left[1+\left(1-\frac{2k}{3}\right)\beta^4\right]M_2^2 \tag{59}$$

where to the same degree of approximation from Equation 36

$$M_2^2 = 2\frac{p_1-p_2}{kp_1} \tag{60}$$

Values of ϕ are also given by Figs. 13 and 14.

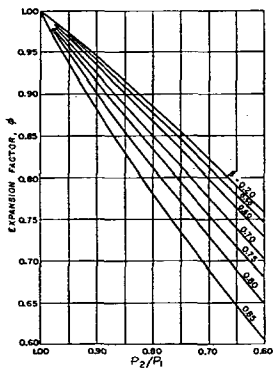


Fig. 13 .... Relation of Expansion Factor, ϕ, for Nozzles to Diameter Ratio and Pressure Loss for Air and Other Diatomic Gases

While ϕ may be used for smooth nozzles and Venturi tubes, it is necessary in the case of orifices, where the departure from ideal flow is significant, to replace ϕ by an empirical factor Y obtained by the equation

$$Y = 1-(0.41+0.35\beta^4)\left(\frac{p_1-p_2}{p_1}/k\right) \tag{61}$$

When this correction is used, the flow will be given by the equation

$$w = KA_2Y\sqrt{2g\rho_1(p_1-p_2)} \tag{62}$$

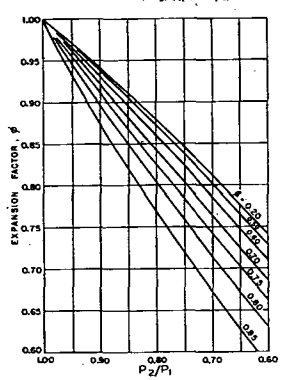


Fig. 14 .... Relation of Expansion Factor, ϕ, for Nozzles to Diameter Ratio and Pressure Loss for Steam, Carbon Dioxide and Natural Gas

Location of Head Meters and Pressure Taps

In the installation of Venturis, orifices, and nozzles, care must be taken in regard to upstream and downstream flow conditions. Recommended practice, with reference to fittings and valves, is shown by Fig. 15. If these conditions cannot be met, some flexibility is possible by introducing straightening vanes, as described more fully in References 2 and 5.

The static pressure difference across the head meter is measured by suitably located static pressure taps. For orifices, the different sets of pressure taps are called flange taps, radius taps, vena contracta taps, and pipe or full-flow taps. The relative locations of the first three of these are shown in *Fig. 16*.

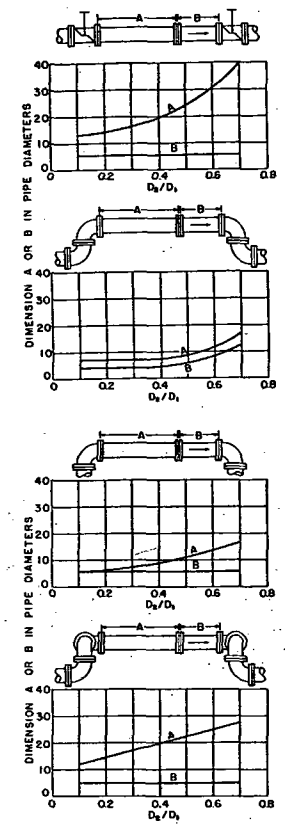


Fig. 15 .... Minimum Conditions to be Observed When Installing Orifices and Nozzles Between Fittings and Valves

The flow coefficients in Figs. 9, 10, and 11 apply only to measurements made with these pressure tap locations. The need for applying suitable coefficients to measurements obtained by means of the different taps, is indicated by the curve of change, in pressure of the flowing fluid, shown in the lower part of Fig. 16. Pipe taps are located 2½ pipe diameters upstream and 8 pipe diameters downstream (both distances being measured from the upstream face of the orifice plate) so that pressures are measured before the orifice plate has had any effect on the flow, and after the recovery in pressure has been completed. The use of pipe or full-flow taps has been limited to the metering of natural fuel gas in certain areas. As they are not suited to use in heating and ventilating work, no data for them are given in this chapter.

Still another type of pressure tap, the corner tap, is used in European practice. Pressures are taken from recesses in the flange connected to annular slits in the corners formed by the pipe wall and the orifice plate. Coefficients for these taps have been adopted by the *International Standards Association*, but are not used commercially in America.

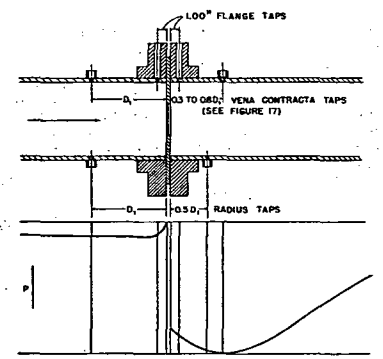


Fig. 16 .... Relative Location of Flange, Radius and Vena Contracta Taps

It will be noted that the location of the downstream pressure tap of the vena contracta arrangement is variable. Vena contracta is the term applied to the minimum cross-section of the jet from the orifice, where the static pressure is at a minimum. Its location, and the location of the downstream vena contracta tap, vary with the ratio of orifice to pipe diameter, and with rate of flow, as shown in Fig. 17; the tap is generally located in accordance with the mean curve in the figure.

Similar precautions are required when using flow nozzles and Venturi tubes. In many cases, these will be supplied

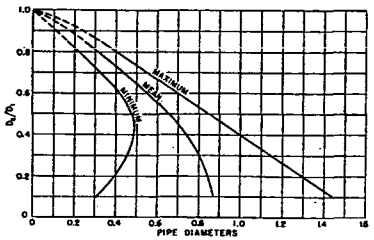


Fig. 17 . . . . Location of Vena Contracta in Relation to Ratio of Orifice to Pipe Diameter and to Rate of Flow

ready for installation. In such cases, the manufacturer's instructions should be followed with care in order to avoid serious errors. Further information on such systems is given in References 2, 3, and 5.

Pitot Tube

In certain cases, such as in rectangular ducts, it is impracticable to use standard orifices, and consequently, either a specially designed orifice must be calibrated, or an independent flow device must be used. In either case, the Pitot tube is useful. It consists essentially of an inner bent tube with its open end pointing upstream so as to measure total pressure, and an outer tube having small holes on the side for communicating static pressure to a manometer. (See Fig. 3, Chapter 44). The difference in liquid level in the manometer will be proportional to the square of the velocity, for incompressible flow, so that in general

$$V = \sqrt{2gh_f} \qquad (63)$$

For compressible flow, the differential head is to be divided by the correction F_c from Equation 64,

$$F_c = 1 + \tfrac{1}{4}M^2 + \tfrac{1}{40}M^4 \qquad (64)$$

where

$$M^2 = 2\frac{p_0 - p}{kp} \qquad (65)$$

In the use of a Pitot tube system, care is required in obtaining correct total and static pressures. The total pressure tube must be smoothly constructed, and should point directly upstream. The static pressure tap must be located so that local flow interferences will not reduce the value. In any case, it is better to obtain an independent calibration, or use a specially designed and manufactured probe. A number of such Pitot tube probes are available, and can be used without calibration.

In using Pitot tubes to obtain flow rates, it is necessary to make a traverse of the pipe and thereby to obtain one of the profiles of Fig. 5. In rectangular ducts or near valves or fittings, a disturbed flow pattern would be obtained, and therefore, in such cases, a fairly complete survey should be made. The flow in such cases will be computed from the average of the local velocities, as obtained by Equation 63.

Variable Area Flow Meters

For permanent installations where high precision, ruggedness, and ease of operation are important, the variable area flow meter has proved very satisfactory. Its most frequent use is in measurement of liquids or gases in small diameter pipes. For ducts or pipes over 6 in. in diameter, the expense of this meter may not be warranted. In large systems, however, the meter might be placed in a by-pass line and used in conjunction with an orifice.

In its most common form, the variable area meter, Fig. 18, consists essentially of a float which is free to move vertically in a transparent tapered tube. The fluid to be metered enters at the narrow bottom end of the tube and moves upward, passing at some point through the annulus formed between the float and inside wall of the tube. At any particular rate of flow, the float assumes a definite position in the tube, its

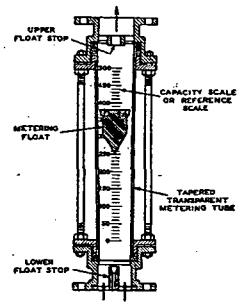


Fig. 18 . . . . Schematic Diagram of Variable Area Flow Meter

location being indicated by means of a calibrated scale on the tube.

The position of the float is established by a balance between the fluid pressure forces across the annulus and the weight of the float itself. The buoyant force which must support the float, $v_f(\rho_f - \rho)$, is balanced by the pressure difference acting on the cross-section area of the float, $A_f\Delta p$, where ρ_f, A_f, v_f, are, respectively, the float density, float cross-section area, and float volume. Accordingly, the difference in head across the annulus is given by

$$h_f = \frac{\Delta p}{\rho} = \frac{v_f(\rho_f - \rho)}{A_f\rho} \qquad (66)$$

The volume flow follows from Equation 53 as

$$Q = KA_2\sqrt{2gv_f(\rho_f - \rho)/\rho A_f} \qquad (67)$$

and the mass flow as

$$w = \rho Q = KA_2\sqrt{2gv_f(\rho_f - \rho)\cdot\rho/A_f} \qquad (68)$$

The flow for any selected fluid is, accordingly, very nearly proportional to the area, so that a convenient calibration of

the tube may be obtained. The behavior of the flow coefficient, K, has been investigated[6] and the action of the flow meter as just outlined, experimentally confirmed. The flow coefficient variation for any float must be known in order to use the meter for different fluids. Some developments have been carried on in the design of the float to reduce the variation of the flow coefficient with Reynolds number, and also with regard to float materials, to reduce the dependence of mass flow calibration on fluid density.

This type of flow meter is usually furnished in standard sizes calibrated for specific fluids by the manufacturer. The compactness, reliability, and ease of installation are particularly advantageous when many measurements of essentially the same type are to be made.

LETTER SYMBOLS USED IN CHAPTER 4

β = ratio, throat or orifice diameter to pipe diameter.
μ = absolute viscosity, pounds per foot second.
μ/ρ = kinematic viscosity, square feet per second.
ρ = density of flowing fluid, pounds per cubic foot.
ρ_m = proper mean density, pounds per cubic foot.
ρ_w = density of water at 60 F (62.37 lb per cubic foot).
ρ_f = density of float in variable area meters.
ϕ = expansion factor for nozzles.
a = velocity of sound, feet per second.
A = cross-sectional area of flow, square feet.
C = correction factor (coefficient of discharge) for flow through orifice, nozzle or Venturi.
c_p = specific heat of gas at constant pressure.
c_v = specific heat of gas at constant volume.
D = diameter of fluid stream, feet.
d = internal diameter of pipe, feet.
d_H = hydraulic diameter, feet.
e = absolute roughness of pipe surface, feet.
F_c = correction factor for differential head in compressible flow.
f = dimensionless friction coefficient.
g = gravitational acceleration, feet per (second) (second).
g_c = gravitational conversion factor = 32.174 (pounds mass per pound force) × feet per (second) (second).
h = enthalpy, Btu per pound of fluid.
h_f = loss of head, feet of fluid.
h_t = total head, feet of fluid.
J = mechanical equivalent of heat = 778 foot pounds per Btu.
K = flow coefficient (correction factor), including velocity of approach correction factor, for flow through orifice, nozzle or Venturi.
k = ratio of specific heat at constant pressure to specific heat at constant volume.
L = perpendicular distance from axis of pipe, feet.
l = length of pipe, feet.
M = Mach number.
N_{Re} = Reynolds number.
p = pressure, pounds per square foot.
p^0 = stagnation pressure, pounds per square foot.
p_c = critical pressure.
Q = discharge rate, cubic feet per second.
q = heat transferred to the fluid per pound of fluid flowing, Btu.
R = gas constant.
r = radius of pipe, feet.
s = entropy of fluid in Btu per (pound) (Fahrenheit degree).
T = temperature, absolute, Fahrenheit degrees.
u = internal energy, Btu per pound of fluid.
V = velocity, feet per second.
V_c = critical velocity, feet per second.
v = specific volume, cubic feet per pound.
W = mechanical work per pound of fluid flowing, foot pounds.
w = mass flow of gas, pounds per second.
Y = expansion factor (correcting for expansion of gas under reduced down-stream pressure).
z = elevation above some arbitrary datum, feet.

REFERENCES

[1] L. F. Moody: Friction factors for pipe flow (*ASME Transactions*, Vol. 66, 1944, p. 671, with discussion p. 678); also An approximate formula for pipe friction factors (*Mechanical Engineering*, Vol. 69, 1947, p. 1005).

[2] American Society of Mechanical Engineers: *Fluid Meters, Their Theory and Application* (1937, 4th ed.).

[3] American Society of Mechanical Engineers: *Flow Measurement, Power Test Codes*, 1949, Part 5, Chapter 4.

[4] National Advisory Committee for Aeronautics, NACA Tech. Mem. 952, 1940: *Standards for Discharge Measurement* (Translation of German Industrial Standard, 1932).

[5] American Gas Association, Natural Gas Department: *Gas Measurement Committee Report* No. 2, 1948.

[6] E. M. Schoenborn Jr. and A. P. Colburn: The flow mechanism and performance of the rotameter (*American Institute of Chemical Engineers Transactions*, Vol. 35, 1939, p. 359).

BIBLIOGRAPHY

E. F. Obert: *Thermodynamics* (McGraw-Hill Book Co., 1948).

N. A. Hall: *Thermodynamics of Fluid Flow* (Prentice Hall, 1951).

R. A. Dodge and M. J. Thompson: *Fluid Mechanics* (McGraw-Hill Book Co., 1937).

R. C. Binder: *Fluid Mechanics* (Prentice Hall, 2nd Ed., 1949).

P. P. Ewald, H. Poschl and L. Prandtl: *The Physics of Solids and Fluids* (Blackie, 1936).

Emory Kemler: A study of the data on the flow of fluids in

pipes (Hydraulic Paper HYD-55-2 *ASME Transactions*, Vol. 55, No. 10, pp. 23–32, 1933).

R. J. S. Pigott: The flow of fluids in closed conduits (*Mechanical Engineering*, Vol. 55, 1933, pp. 497–501 and p. 515).

American Society of Mechanical Engineers: *Fluid Meters, Their Selection and Installation* (1933).

A. D. MacLean: *The Orifice Meter for Measurement of Flow of Gases and Liquids* (Pittsburgh Equitable Meter Co., 1938).

Edward S. Cole: Pitot tube practice (*ASME Transactions*, Vol. 57, 1935, pp. 281–294; with discussion Vol. 58, 1936, pp. 146–156).

Edward S. Cole and E. Shaw Cole: Pitot tubes in large pipes (*ASME Transactions*, Vol. 61, 1939, pp. 465–473, with discussion pp. 473–475).

C. W. Hubbard: Investigation of errors of pitot tubes (*ASME Transactions*, Vol. 61, 1939, pp. 477–497, with discussion pp. 497–506).

R. E. Sprenkle: Piping arrangements for acceptable flowmeter accuracy (*ASME Transactions*, Vol. 67, 1945, pp. 345–357, with discussion pp. 357–360).

CHAPTER 5

HEAT TRANSFER

Conduction, Convection, Radiation; Equations for Conduction, Convection, Radiation, and Combined Convection and Radiation; Heat-Flow Resistance, in Series and Parallel; Practical Heat Transfer Problems; Periodic and Transient Heat Flow

HEAT is the form of energy that is transferred by virtue of an existing temperature difference. The temperature difference is the potential which causes the transfer, the latter in turn being resisted by the thermal properties of the material combined in a single term known as the resistance. Energy exchange associated with evaporation, condensation, etc., is treated elsewhere such as in the section Cooling Tower Theory in Chapter 40. The objectives of this chapter are to:

1. Describe the mechanisms and present the rate equations for the different modes of heat transfer.
2. Illustrate the application of the basic concepts to steady-state problems (temperature independent of time or a cyclic variable thereof) by means of several typical solutions of heat transfer systems.
3. Present concise summaries of available methods of analysis for transient and periodic heat transfer problems.

Further applications to specific systems will be found throughout THE GUIDE.

CONDUCTION, CONVECTION, AND RADIATION

Thermal conduction is the term applied to the mechanism of heat transfer whereby the molecules of higher kinetic energy transmit part of their energy to adjacent molecules of lower kinetic energy by direct molecular action. Since the temperature is proportional to the average kinetic energy of the molecules, thermal transfer will occur in the direction of decreasing temperature. The motion of the molecules is random; there is no net material flow associated with the conduction mechanism. In the case of flowing fluids, thermal conduction is significant in the region very close to a solid boundary or wall, where the flow is laminar and parallel with the wall surface, and where practically no cross currents exist in the direction of the heat transfer across the solid fluid boundary. In solid bodies the significant mechanism of heat transfer is always thermal conduction.

In contrast to the thermal conduction mechanism, *thermal convection* involves energy transfer by eddy mixing and diffusion[1] in addition to conduction. This is shown schematically in Fig. 1 which exhibits transfer from a pipe wall at surface temperature t_s to a colder fluid at a bulk temperature t_f. (Bulk temperature is that which would be attained if the fluid stream were drawn off at a certain section and mixed. It is therefore somewhat higher than the lowest temperature in the stream.) In the *laminar* sublayer, immediately adjacent to the wall, the heat transfer occurs by thermal conduction; in the *transition* region, which is called the buffer layer, eddy mixing as well as conduction effects are significant; in the *eddy* or *turbulent* region the major fraction of the transfer occurs by eddy mixing.

In most commercial equipment the main body of the fluid is in turbulent flow, and the laminar film exists at the solid walls only, as shown in Fig. 1. In cases of low-velocity flow in small tubes, or with viscous liquids such as heavy oil (low Reynolds numbers), the entire flow may be laminar. In these latter cases there is no transition or eddy region.

When the fluid currents are produced by sources external to the heat transfer region, as for example by a pump, the described solid to fluid heat transfer is termed *forced convection*. In contrast, if the fluid currents are generated internally, as a result of non-homogeneous densities arising from the temperature variations, the heat transfer is termed *free convection*.

In the conduction and convection mechanisms, the transfer of heat is associated with matter. For *radiant heat transfer*, however, a change in energy form takes place, from internal energy at the source to electromagnetic energy for transmission, then back to internal energy at the receiver.

The rate of heat transfer, corresponding to the three transfer mechanisms previously described, may be expressed by three rate equations. These are similar to Ohm's Law for electrical flow, the current flow through a resistance being proportional to the potential. The convection and radiation flow rate expressions may be approximated by a potential (temperature difference) and a resistance in order that heat transfer calculations may be effected more conveniently and rapidly.

Thermal Conduction Equation

Equation 1 states symbolically that the thermal conduction per unit transfer area normal to the flow, q/A, Btu per (hour) (square foot), is proportional to the temperature gradient $(dt)/(dL)$, Fahrenheit degrees per foot. The proportionality factor is termed the *thermal conductivity*, k, Btu per (hour) (square foot) (Fahrenheit degree per foot of thickness).

$$\frac{q}{A} = -k\frac{dt}{dL} \tag{1}$$

The minus sign on the right side of the equation is introduced to indicate positive transfer in the direction of decreasing temperature. Fig. 2 shows the physical significance of the indicated quantities.

It should be emphasized that the thermal conductivity used should be expressed in consistent units; either using the inch or foot throughout.

Expressions of conductivity used in the heating field are usually inconsistent in this sense, in that it is customary to refer to the conductivity *per square foot* but for one inch of thickness. This custom has been adopted for the reason that wall thicknesses are usually expressed in inches, whereas if expressed in feet, decimal or fractional thicknesses would result. When dealing with flat walls, no complication is involved in using the inconsistent expression of conductivity. However, where curved or spherical walls are concerned,

considerable complication is involved. Therefore, in this discussion the consistent units of conductivity expressed in Btu per (hour) (square foot) (Fahrenheit degrees per one foot thickness) are used throughout. *Conductivity values obtained from Chapter 9 or Table 1 in this chapter, must therefore be converted for use in the calculations of this chapter by dividing by 12.* As an example, the conductivity of brick listed as 5.0 in Table 4 of Chapter 9, becomes 0.42 when used in the calculations of this chapter. Also, it should be emphasized that in order to make the calculations and applications consistent in this chapter, *all dimensions of thickness must be expressed in feet.*

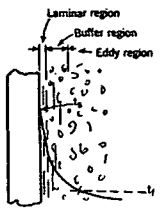


Fig. 1 .... Thermal Convection Conditions

Thermal Convection Equation

$$\frac{q}{A} = h_c(t_s - t_f) \qquad (2)$$

This rate equation states that the thermal convection per unit transfer area (q/A), Btu per (hour) (square foot) is proportional to the temperature difference ($t_s - t_f$) which is the temperature of the surface less that of the fluid. The particular fluid temperature to use for a given system will be noted under the discussion of that system. The proportionality factor is termed the *unit thermal convective conductance* (sometimes called the film coefficient for convection), h_c, Btu per (hour) (square foot) (Fahrenheit degree). Fig. 1 shows the conditions associated with convection.

The heat transmission by free or natural convection for objects surrounded by air can be conveniently expressed as in Equation 3:

$$\frac{q}{A} = C\left(\frac{1}{D}\right)^{0.2}\left(\frac{1}{T_{av}}\right)^{0.181}(t_s - t_f)^{1.27} \qquad (3)$$

where

$\frac{q}{A}$ = heat transmission by convection, Btu per (square foot) (hour).

Table 1 .... Approximate Unit Thermal Conductivities[a]

Conductivity, k = Btu per (hr) (sq ft) (F deg per in.)

Material	k	Material	k
Air	0.168	Lead	240.0
Aluminum	1416.0	Nickel	408.0
Brass (70 – 30)	720.0	Soil	2.4–12.0
Cast Iron	336.0	Steel, mild	312.0
Copper	2640.0	Water, liquid	4.08
Glass	3.6–7.32		

[a] Thermal conductivities depend to some extent on temperature. The above magnitudes are approximate only. Refer to Chapter 9, and Reference 4 for additional data.

C = a constant depending upon the shape of the surface.

D = diameter of pipe or circular duct or height of vertical wall, inches. (Effect of diameter or height becomes constant at 24 in.)

T_{av} = average of wall surface and surrounding air temperature, Fahrenheit degrees absolute.

$t_s - t_f$ = temperature excess between wall surface and surrounding air, Fahrenheit degrees.

For horizontal cylinders, the value of $C = 1.02$ has been well established by various investigations. For vertical plates, the value of $C = 1.39$ has been fairly well established. Suggested values[3] of C for horizontal plates warmer than the surrounding air are 1.79 when facing upward, and 0.89 when facing downward.

Problems in either forced convection or natural convection may be solved by the simple first-power equation if the convection coefficient is expressed as a unit conductance:

$$q = h_c A(t_1 - t_2) \qquad (4)$$

where

q = heat transmission by convection, Btu per hour.

A = surface area, square feet.

$t_1 - t_2$ = temperature difference between the surface and the fluid, Fahrenheit degrees.

h_c = unit convective conductance, from Table 2, Btu per (square foot) (hour) (Fahrenheit degree temperature difference).

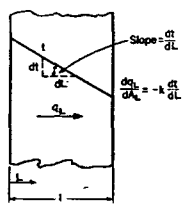


Fig. 2 .... Thermal Conduction in a Flat Slab

Thermal Radiation Equation

The relation given by Equation 5 is applicable to systems in which radiant exchange takes place between the surfaces of solids, as schematically shown in Fig. 3.

$$q_r = \sigma A_1 F_A F_E (T_1^4 - T_2^4) \qquad (5)$$

Gaseous and luminous radiation are not considered in this discussion. Equation 5 states that the net radiation per unit transfer area of surface 1, q_r/A Btu per (hour) (square foot), which *sees* surface 2 through a non-absorbing medium, is proportional to the difference of the fourth powers of the absolute surface temperatures ($T_1^4 - T_2^4$). The proportionality factor ($\sigma F_A F_E$) may be conveniently separated into three parts (excepting in some problems involving interreflections, where it is not possible to divide the product ($F_A F_E$) into separate terms):

σ = the Stefan-Boltzmann radiation constant = 1730×10^{-12} Btu per (hour) (square foot) (Fahrenheit degree absolute temperature to the fourth power).

Table 2 .... Approximate Unit Conductances for Thermal Convection for Several Flow Systems

Expressed in Convenient Empirical Form

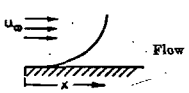
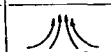
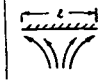
Case	System	Heat Transfer Equation[a] and Its Limits of Application
	Forced Convection	
1 Ref. 3	 Longitudinal flow in a circular cylinder.[b]	*General Equation* $\frac{h_c D}{k} = 0.0225\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{C_p\mu}{k}\right)^{0.33}$ where $x > 4.4D$ and $\left(\frac{DG}{\mu}\right) > 2200$
2 Ref. 3	Same as Case 1.[b]	*Equation for Air* $h_c = 5.4 \times 10^{-4}(T_f)^{0.8}\frac{G^{0.8}}{D^{0.2}}$ where $x > 4.4D$ and $\left(\frac{DG}{\mu}\right) > 2200$
3 Ref. 3	Same as Case 1.[b]	*Equation for Liquid Water* (32–400 F) $h_c = 13.5(t_f)^{0.64}\frac{u_m^{0.8}}{D^{0.2}}$ where $x > 4.4D$ and $\left(\frac{DG}{\mu}\right) > 2200$
4 Ref. 4	 Flow normal to a single cylinder.	*General Equation* $\frac{h_c D}{k} = 0.26\left(\frac{DG}{\mu}\right)^{0.6}\left(\frac{c_p\mu}{k}\right)^{0.3}$ where $1000 < \frac{DG}{\mu} < 50{,}000$
5 Ref. 4	Same as Case 4.	*Equation for Air* $h_{c(average)} = 0.211(T_f)^{0.42}\frac{(u_\infty\rho)^{0.6}}{D^{0.4}}$ where $1000 < \frac{DG}{\mu} < 50{,}000$
6 Ref. 4	Same as Case 4.	*Equation for Liquid Water* (32–400 F) $h_{c(average)} = 34.0(t_f)^{0.25}\left(\frac{(u_\infty)^{0.6}}{D^{0.4}}\right)$ where $1000 > \frac{DG}{\mu} > 50{,}000$
7 Ref. 3	 Flow along a flat plate.	*General Equation* (Turbulent Flow) $\frac{h_{cx}x}{k} = 0.0296\left(\frac{xG}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{k}\right)^{0.33}$ where $\left(\frac{xG}{\mu}\right) > 500{,}000$ and $h_a = 1.25h_{cx}$
8 Ref. 3	Same as Case 7.	*Equation for Air* (Turbulent Flow) $h_{cx} = 0.51(T_f)^{0.8}\frac{(u_\infty\rho)^{0.8}}{(x)^{0.2}}$ where $\left(\frac{xG}{\mu}\right) > 500{,}000$ and $h_{c(average)} = 1.25h_{cx}$
9 Ref. 3	Same as Case 7.	*General Equation* (Laminar Flow) $\frac{h_{cx}x}{k} = 0.332\left(\frac{xG}{\mu}\right)^{0.5}\left(\frac{c_p\mu}{k}\right)^{0.33}$ For $\left(\frac{xG}{\mu}\right) < 500{,}000$ and $h_{c(average)} = 2h_{cx}$
10 Ref. 3	Same as Case 7.	*Equation for Air* (Laminar Flow) $h_{cx} = 0.0562(T_f)^{0.8}\frac{(u_\infty\rho)^{0.8}}{x^{0.2}}$ For $\left(\frac{xG}{\mu}\right) < 500{,}000$

Table 2 .... Approximate Unit Conductances for Thermal Convection for Several Flow Systems (Concluded)

Case	System	Heat Transfer Equation[a] and Its Limits of Application
		Free Convection[a]
11 Ref. 5	 Free convection past a heated horizontal cylinder.	*Equation for Air* $h_c = 0.271 \left(\frac{P}{P_a}\right)^{0.5} \left(\frac{\Delta t}{D}\right)^{0.25}$ *where* $10^3 < N_{Gr} < 10^7$
12 Ref. 6	 Free convection past a single vertical surface.	*Equation for Air* $h_c = 0.354 \left(\frac{P}{P_a}\right)^{0.50} \left(\frac{\Delta t}{l}\right)^{0.25}$ *where* $10^3 < N_{Gr} < 10^7$ $h_c = 0.420 \left(\frac{P}{P_a}\right)^{0.50} \left(\frac{\Delta t}{l}\right)^{0.25}$ *where* $10^4 < N_{Gr} < 10^{10}$
13 Ref. 5	 Free convection past a heated horizontal surface (face up).	*Equation for Air* $h_c = 0.478 \left(\frac{P}{P_a}\right)^{0.50} \left(\frac{\Delta t}{l}\right)^{0.25}$ *where* $10^3 < N_{Gr} < 10^7$
14 Ref. 5	 Free convection past a heated horizontal surface (face down).	*Equation for Air* $h_c = 0.239 \left(\frac{P}{P_a}\right)^{0.50} \left(\frac{\Delta t}{l}\right)^{0.25}$ *where* $10^3 < N_{Gr} < 10^7$

Nomenclature and Dimensions for Table 2

- c_p = heat capacity at constant pressure, Btu per (pound) (Fahrenheit degree).
- D = cylinder diameter, feet.
- f = subscript denoting film.
- g = body force per unit mass, feet per hour per hour. (For static system on earth, $g = 32.2 \times 3600^2$ feet per hour per hour.)
- $G = 3600\, u_m \rho$ = mass flow per unit cross-sectional area normal to flow, pounds per (hour) (square foot of flow cross-section).
- N_{Gr} = Grashof modulus, dimensionless ($N_{Gr} = D^3 p^2 g \beta \Delta t / \mu^2$).
- h_x = average unit thermal convective conductance from the leading edge of surface to the position x, Btu per (hour) (square foot) (Fahrenheit degree).
- h_{cx} = local unit thermal convective conductance, at the position x from the leading edge of surface, Btu per (hour) (square foot) (Fahrenheit degree).
- k = thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot thickness).
- l = a dimension of the system, feet.
- m = a subscript denoting *mean*.
- P = pressure, atmospheres.
- P_a = pressure (atmospheric), atmospheres.
- t = temperature, Fahrenheit.
- T = temperature, Fahrenheit, absolute.
- u = fluid velocity, feet per second.
- V = volume, cubic feet.
- x = a dimension of the system, feet.
- β = coefficient of cubical expansion ($\beta = \frac{1}{V}\left(\frac{dV}{dT}\right)_p$); for perfect gases $\beta = 1/T$.
- Δt = difference between wall and fluid temperatures, Fahrenheit degrees.
- μ = fluid viscosity, pounds per (hour) (foot).
- ρ = density, pounds per cubic foot.
- ∞ = infinity, referring the quantity to a point not directly affected by the phenomenon in question.

[a] Fluid properties should be evaluated at the arithmetic mean fluid temperature, $t_f = (t_{surface} + t_{fluid})$ divided by 2.
[b] These expressions are suitable approximations to longitudinal flow in other than right circular cylinders, provided the hydraulic diameter is employed as the conduit dimension parameter. For non-circular cross-sections, the hydraulic diameter is equal to four times the cross-sectional area divided by the wetted perimeter.
[c] For low rates of heat transfer by free convection the exponent decreases towards zero, and for higher rates, increases towards 0.33. The above equations employing an exponent equal to 0.25 are applicable in the intermediate range indicated.

F_A = the geometrical factor which is dimensionless and ≤ 1. This factor accounts for the shape and relative position of the two surfaces. The value of $F_A = 1$ may be used in the cases of large parallel planes, long concentric cylinders or smaller bodies in large enclosures.

F_E = the emissivity factor which is also dimensionless and ≤ 1. This factor accounts for the absorption and emission characteristics of the surfaces for the radiation which exists. Emissivities or absorptivities (ϵ) for many common surfaces, are given in Table 3. The value of F_E for large parallel planes, long concentric cylinders, or large enclosed bodies is $1 \div (1/\epsilon_1 + 1/\epsilon_2 - 1)$.

The radiation under black-body conditions, or for an emissivity of 1.0, is given in Table 4[7] for cold surfaces as low as -39 F to warmer surfaces as high as 139 F. Some net radiation exchange solutions for several common radiation systems are given in Table 5.

There are several methods by which the geometrical factor F_A can be determined. One method involves the use of a mechanical geometrical integrator (Reference 8). Photographic and other methods are given in References 9 and 10.

Equivalent Conductance for Radiation

Although Equation 5 is a suitable equation for describing radiant exchange, it is not convenient for computations where other modes of energy transfer are operative. For such cases, it is convenient to define an *equivalent conductance for radiation* by the equation:

$$q_r = h_r A (t_1 - t_2) \tag{6}$$

The conductance h_r thus defined is a function of the shape-emissivity factor, as well as the temperatures of the radiator and receiver. Fig. 7 shows a plot of the equivalent conductance for two *black bodies* (i.e., with emissivities equal to unity) which exchange energy only with one another.

Combined Convection and Radiation

It should be noted that the previous equations and tables give the heat transfer by convection and by radiation computed separately. In many practical cases it is desirable to treat convection and radiation as a single combined process, using a first-power equation:

$$q_{rc} = h_{rc} A (t_1 - t_2) \tag{7}$$

where q_{rc} is the total heat flow due to radiation and convection, in Btu per hour. Values of h_{rc}, the surface or film conductance for combined radiation and convection, are given in Table 3 and Fig. 4 of Chapter 9. Complete tables for the combined heat transfer of steam and hot water radiators, pipes, coverings, etc., will be found in the appropriate chapters.

HEAT-FLOW RESISTANCE

In most of the steady-state heat transfer problems encountered in air conditioning applications, more than one of the heat transfer mechanisms are effective, and the thermal current flows through several resistances in series or in parallel. In using the resistance concept, the calculations involved are analogous to the application of Ohm's Law in electricity, viz., the heat flow or thermal current is directly proportional to the thermal potential or temperature difference, and inversely proportional to the thermal resistance:

$$q_{rc} = \frac{t_1 - t_2}{R} \tag{8}$$

Following the electrical analogy, when there is a thermal current flowing through several *resistances in series*, the resistances are additive:

$$R_T = R_1 + R_2 + R_3 + \cdots + R_n \tag{9}$$

Similarly, conductance is the reciprocal of resistance, and for heat flow through several *resistances in parallel*, the conductances are additive:

$$C_T = \frac{1}{R_T} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \cdots + \frac{1}{R_n} \tag{10}$$

Practical Heat Transfer Problems

The use of these relations for resistance and conductance makes possible the solution of many practical heat transfer problems. As discussed in Chapters 9, 23, and 32, the practical analyses of heat transfer in building walls, in fin-tube coils, and in pipe coverings, are usually computed by this method. The same resistance analysis may be applied to complicated steady-state *conduction* problems. Table 6 gives the resistances in six common cases of steady-state conduction.

A complete analysis by the resistance method is well illustrated by considering the heat transfer from the air outside to the cold water inside of an insulated pipe. The temperature gradients and the nature of the resistance analysis are indicated by the two sketches of Fig. 8.

Since air is sensibly transparent to radiation, there will be some heat transfer by both radiation and convection to the outer insulation surface. The mechanisms act in parallel on the air side. The total transfer by radiation and convection then passes through the insulating layer and the pipe wall by *thermal conduction, and thence by convection and radiation* into main cold water streams. (Radiation is not significant on the water side as liquids are sensibly opaque to radiation, although water transmits energy in the visible region.) The contact resistance between the insulation and the pipe wall is presumed to be equal to zero.

Referring to Fig. 8, the heat transferred for a given length N of pipe, q_{rc}, Btu per hour, may be thought of as flowing through the parallel resistances R_r and R_c, associated with the insulation surface radiation and convection transfer. Then the flow is through the resistance offered to thermal conduction by the insulation, R_3, through the pipe wall resistance, R_2, and into the water stream through the convection resistance, R_1. Note the analogy to the direct current electrical circuit problem. A temperature (potential) drop is required to overcome these resistances to the flow of thermal current. The total resistance to heat transfer, R_t, (hour) (Fahrenheit degrees) per Btu, is the summation of the individual resistances:

$$R_T = R_1 + R_2 + R_3 + R_4 \tag{11}$$

where the resultant parallel resistance R_4 is obtained from

$$\frac{1}{R_4} = \frac{1}{R_r} + \frac{1}{R_c} \tag{12}$$

Provided the individual resistances may be evaluated, the total resistance can be obtained from this relation. Then the heat transfer for the length of pipe (N, ft) can be established by the relation

$$q_{rc}\ (\text{Btu per hour}) = \frac{(t_s - t_f)}{R_T} \tag{13}$$

For a unit length of the pipe the heat transfer rate is

$$\frac{q_{rc}}{N}\ \text{Btu per (hour) (foot)} = \frac{(t_s - t_f)}{R_T N} \tag{14}$$

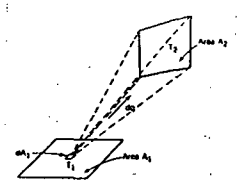


Fig. 3 .... Radiation Between Surfaces

Table 3 . . . . Radiation Factors or Emissivities, ϵ*

For the determination of factor F_E in Equation 3

Class	Surfaces	Fraction of Black-Body Radiation		Absorptivity for Solar Radiation
		At 50–100 F	At 1000 F	
1	A small hole in a large box, sphere, furnace, or enclosure	0.97 to 0.99	0.97 to 0.99	0.97 to 0.99
2	Black non-metallic surfaces such as asphalt, carbon, slate, paint, paper	0.90 to 0.98	0.90 to 0.98	0.85 to 0.98
3	Red brick and tile, concrete and stone, rusty steel and iron, dark paints (red, brown, green, etc.)	0.85 to 0.95	0.75 to 0.90	0.65 to 0.80
4	Yellow and buff brick and stone, firebrick, fire clay	0.85 to 0.95	0.70 to 0.85	0.50 to 0.70
5	White or light-cream brick, tile, paint or paper, plaster, whitewash	0.85 to 0.95	0.60 to 0.75	0.30 to 0.50
6	Window glass	0.90 to 0.95	. . . . .	Transparent*
7	Bright aluminum paint; gilt or bronze paint	0.40 to 0.60	. . . . .	0.30 to 0.50
8	Dull brass, copper, or aluminum; galvanized steel; polished iron	0.20 to 0.30	0.30 to 0.50	0.40 to 0.65
9	Polished brass, copper, monel metal	0.02 to 0.05	0.05 to 0.15	0.30 to 0.50
10	Highly polished aluminum, tin plate, nickel, chromium	0.02 to 0.04	0.05 to 0.10	0.10 to 0.40

* Emissivities of other materials may be found in Reference 4. * Reflects about 8 percent.

The temperature drop, Δt, through an individual resistance may then be calculated from the relation:

$$\Delta t = Rq_r \qquad (15)$$

where R is the resistance in question.

The problem is now reduced to one of evaluating the individual resistances of the system. This entails suitable manipulation of the rate Equations 1, 2, and 5 to produce expressions of the form:

$$q = \frac{\Delta t}{R} \qquad (16)$$

where q is the heat transfer rate, and Δt is the potential drop or temperature difference through the resistance R. Table 6 lists such solutions for six different conduction systems. Table 4 in Chapter 9 and Table 1 of this chapter indicate the magnitudes of the thermal conductivities, k, to be employed in the expressions of Table 6, after dividing k by 12.

The solution applicable to the problem depicted in Fig. 8, for the calculation of R_2 and R_3, is Case 2 in Table 6. Thus for a *1-ft length* of 2 in. nominal size pipe (I. D. = 2.067 in., O. D. = 2.375 in.) insulated with 1 in. of material having a conductivity of 0.025:

$$R_2 = \frac{\log_e \frac{1.188}{1.033}}{2\pi \times 26 \times 1} = 8.5 \times 10^{-4} \text{ (hr) (F deg) per Btu.}$$

$$R_3 = \frac{\log_e \frac{2.188}{1.188}}{2\pi \times 0.025 \times 1} = 3.9 \text{ (hr) (F deg) per Btu.}$$

The convection resistances to heat transfer from the pipe wall to the cold water, R_1, and from the air to the surface of the insulating material, R_4, are dependent on the flow conditions prevailing at these surfaces, and on the thermal properties of the fluids. These resistances are also directly dependent upon the temperature distribution, and for this reason it is necessary first to guess, on the basis of the problem statement, a temperature distribution upon which to base the initial calculations. Since the values of h_c for heat transfer between water and pipe walls are relatively high in this temperature range, it is logical to assume only a small temperature difference between the temperature of the fluid body and the temperature of the pipe wall. For the purpose of an initial guess, this temperature difference will be assumed to be 2 deg. On the other hand, h_c for heat transfer from air to a body is relatively small, and a higher temperature difference would be expected between these masses. The value initially assumed here will be 20 deg. In summary, the temperature distribution in the system is assumed as follows:

Fluid temperature = 34 F.
Inner pipe wall temperature = 36 F.
Outer insulation surface temperature = 100 F.
Ambient air temperature = 120 F.

With these assumptions and the problem statement, it is now possible to calculate values for the convective resistances. If it is found in the ultimate solution of the problem that the temperature distribution is different from that assumed, it will then be necessary to repeat the solution procedure.

If reference now be made to Table 2, it is found that Case 3 of this table is a system similar to that encountered in the

Table 4 . . . . Heat Transmission by Radiation for Black-Body Conditions*

Expressed in Btu per (square foot) (hour)

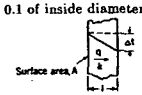
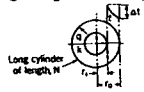
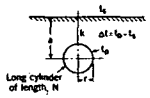
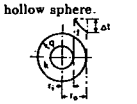
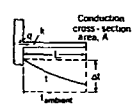
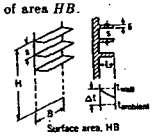
Temp F Deg	0	−1	−2	−3	−4	−5	−6	−7	−8	−9
−30	59.3	58.7	58.2	57.7	57.2	56.7	56.2	55.7	55.2	54.7
−20	65.2	64.7	64.1	63.5	62.9	62.3	61.7	61.1	60.5	59.9
−10	71.4	70.8	70.1	69.5	68.9	68.3	67.7	67.1	66.4	65.8
0	78.0	77.4	76.7	76.0	75.4	74.7	74.0	73.4	72.7	72.1
	0	+1	+2	+3	+4	+5	+6	+7	+8	+9
0	78.0	78.7	79.4	80.1	80.8	81.5	82.2	82.9	83.6	84.3
10	85.0	85.7	86.5	87.2	88.0	88.7	89.4	90.2	90.9	91.7
20	92.4	93.3	94.0	94.8	95.6	96.4	97.2	98.0	98.8	99.6
30	100	101	102	103	104	105	105	106	107	108
40	109	110	111	112	112	113	114	115	116	117
50	118	119	120	121	122	123	123	124	125	126
60	127	128	129	130	131	132	133	134	135	136
70	137	138	139	140	142	143	144	145	146	147
80	148	149	150	151	152	153	154	155	156	157
90	159	160	161	162	163	164	166	167	168	169
100	170	171	173	174	175	176	178	179	180	182
110	183	184	185	187	188	189	191	192	193	195
120	196	197	199	200	201	203	204	206	207	209
130	211	212	214	215	217	218	220	221	222	224

* *Example:* Radiation from walls of room at 33 F to surface at −25 F for effective emissivity of 0.95 = (102 − 63.3) 0.95 = 37.7 Btu per (square foot) (hour).

Table 5 . . . Net Radiation Solutions

System	Solution	Remarks
$\epsilon_1 \quad \epsilon_2$ Two infinite parallel planes.	$\frac{q_r}{A} = \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \sigma(T_1^4 - T_2^4)$	Considering interreflections. (Reference 5)
$\epsilon_1 \quad \epsilon_3 \quad \epsilon_2$ $\epsilon_1 = \epsilon_2$ One radiation shield between two infinite parallel planes.	$\frac{q_r}{A} = \left(\frac{1}{2}\right) \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \sigma(T_1^4 - T_2^4)$	Considering interreflections. (Reference 5)
$\epsilon_1 = \epsilon_2 = \epsilon_3 = \cdots = \epsilon_n$ n radiation shields between two infinite parallel planes.	$\frac{q_r}{A} = \frac{1}{n+1} \left(\frac{q_r}{A}\right)_s$ where $\left(\frac{q_r}{A}\right)_s$ is the net radiation exchange without the shields.	Considering interreflections. (Reference 5)
Two concentric spheres or two infinitely long cylinders.	$\frac{q_r}{A_1} = \frac{1}{\frac{1}{\epsilon_1} + \frac{A_1}{A_2}\left(\frac{1}{\epsilon_2} - 1\right)} \sigma(T_1^4 - T_2^4)$	Considering interreflections and diffuse surfaces. (Reference 5)
Two areas dA_1 and dA_2	$\frac{dq_r}{dA_1} = \epsilon_1 \epsilon_2 F_A (T_1^4 - T_2^4)$	Surface diffuse, neglecting interreflection. (Reference 5)
Tube of infinite length parallel to an infinite wall.	$\frac{q_r}{2\pi RN} = \left(\frac{1}{2}\right) \epsilon_1 \epsilon_2 \sigma (T_1^4 - T_2^4)$ where N is the length of cylinder from which q_r is exchanged.	Neglecting interreflections. (Reference 5)
Surfaces are perfect radiators. Surface element dA and rectangle above and parallel to it, with one corner of rectangle contained in normal to dA.	See Fig. 4	(Reference 4)
Surfaces are perfect radiators. Adjacent rectangles in perpendicular planes.	See Fig. 5	(Reference 4)
Surfaces are perfect radiators. Opposed parallel rectangles and discs of equal size.	See Fig. 6	(Reference 4)

Table 6 . . . . Solutions for Some Steady-State Thermal Conduction Problems[a, b]

No.	System	Expressions for the Resistance R Entering into the Equation: $q = \Delta t/R$ (Btu per hour)
1.	Flat wall or curved wall if curvature is small (wall thickness less than 0.1 of inside diameter). 	$R = \frac{L}{kA}$
2.	Radial flow through a right circular cylinder. 	$R = \frac{\log_e \frac{r_o}{r_i}}{2\pi kN}$ (See footnote c).
3.	The buried cylinder. 	$R = \frac{\log_e \left(\frac{a + \sqrt{a^2 + r^2}}{r}\right)}{2\pi kN} = \frac{\cosh^{-1}\left(\frac{a}{r}\sqrt{1 + \frac{r^2}{a^2}}\right)}{2\pi kN}$ For $\frac{a}{r} \geq 3$, satisfactory approximation is: $R = \frac{\log_e \frac{2a}{r}}{2\pi kN} = \frac{\cosh^{-1}\frac{a}{r}}{2\pi kN}$
4.	Radial flow in a hollow sphere. 	$R = \frac{\frac{1}{r_i} - \frac{1}{r_o}}{4\pi k}$
5.	The straight fin or rod heated at one end. 	$R = \frac{m}{h_s p \tanh mL}$ (see footnotes d and e). For $ml > 2.3$, $\tanh mL \approx 1$ $m = \sqrt{h_s p/kA}$ A = conduction cross-section area. p = perimeter of cross-section A. h_s = unit conductance to the surroundings from the fin surface. k = thermal conductivity fin material. Δt = wall temperature—ambient temperature.
6.	Finned surface of area HB. 	$R = \frac{(s + \delta)}{h_s \left(\frac{2}{m}\tanh ml + s\right) HB}$ $m = \sqrt{\frac{h_s p}{kA}} = \sqrt{\frac{2h_s}{k\delta}}$ Δt defined as in Case 5 above.

[a] The dimensions to be employed in these solutions are: length of dimension p, L, r = feet; units of k = Btu per (hour) (square foot) (Fahrenheit degree for *one foot thickness*); units of k, Btu per (hour) (square foot) (Fahrenheit degree); units of area, A = square feet.

[b] The thermal conductivity, k, in these solutions should be taken at the average material temperature.

[c] $\log_e x = 2.303 \log_{10} x$.

[d] This expression can also be employed as an approximation for tapered fins or of annular fins by employing average magnitudes of A and p.

[e] tanh is the hyperbolic tangent.

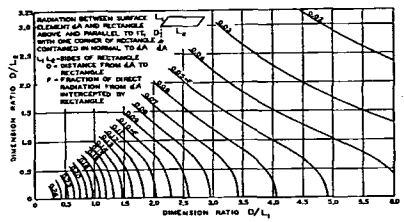


Fig. 4 . . . . Geometrical Factor F for Direct Radiation Between an Element dA and a Parallel Rectangle*

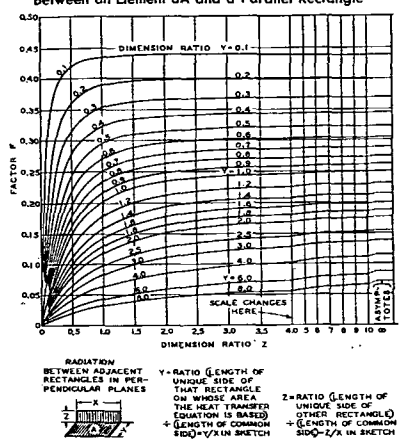


Fig. 5 . . . . Geometrical Factor F for Direct Radiation Between Adjacent Rectangles in Perpendicular Planes*

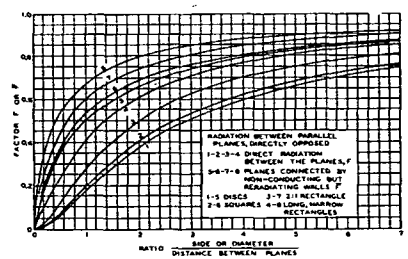


Fig. 6 . . . . Geometrical Factor F for Direct Radiation Between Opposed Parallel Rectangle and Discs of Equal Size*

* H. C. Hottel, Radiant heat transmission, (*Mechanical Engineering*, July 1930, pp. 700 to 702).

convection between the water and the pipe wall. The equation for this case is the following:

$$h_c = 13.9(t_f)^{0.545}\frac{u_m^{0.8}}{D^{0.2}} \qquad (17)$$

where

$$u_m = 5 \text{ fps}$$

$$D = \frac{2.067}{12} = 0.1725 \text{ ft}$$

$$t_f = \frac{34 + 36}{2} = 35 \text{ F}$$

$$h_c = \frac{13.9 \times 6.9 \times 3.62}{0.703} = 494 \text{ Btu per (hr) (sq ft) (F deg).}$$

This heat transfer rate is through the inner surface of the pipe and it is, therefore, this area that determines the resistance R_i.

$$A = \pi D = 0.542 \text{ sq ft per unit length of pipe,}$$

and therefore

$$R_i = \frac{1}{h_c A} = \frac{1}{494 \times 0.542}$$

$$R_i = 3.73 \times 10^{-3} \text{ (hr) (F deg) per Btu.}$$

Case 11 of Table 2 fits the conditions of the problem if only free convection heating of the pipe is assumed. The equation in this case is as follows:

$$h_c = 0.271\left(\frac{P}{P_a}\right)^{0.5}\left(\frac{\Delta t}{D}\right)^{0.25} \qquad (18)$$

where

$$\Delta t = 20 \text{ F}$$
$$D = 0.364 \text{ ft}$$
$$P = P_a = \text{one atmosphere}$$

therefore,

$$h_c = 0.271\left(\frac{20}{0.364}\right)^{0.25}$$

$$h_c = 0.737 \text{ Btu per (hr) (sq ft) (F deg)}$$

Using the surface area of the insulation, the value of the resistance per unit length is determined.

$$A = \pi\left(\frac{4.375}{12}\right) \times 1 = 1.14 \text{ sq ft}$$

$$R_c = \frac{1}{h_c A} = \frac{1}{0.737 \times 1.14}$$

$$R_c = 1.19 \text{ (hr) (F deg) per Btu.}$$

This result may not be deemed conservative inasmuch as the expression is for still air. If, however, the air is not still, but flows at approximately 5 mph or 7 fps, the heat transfer equation for forced convection would apply. This equation is Case 5 of Table 2.

$$h_{c(\text{forced})} = 0.211(T_f)^{0.43}\frac{(u\rho)^{0.8}}{D^{0.4}} \qquad (19)$$

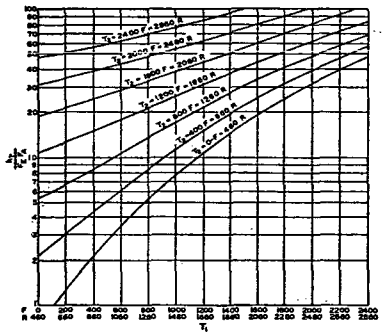


Fig. 7 .... Equivalent Conductance for Radiation Between Two Black Bodies Exchanging Energy Only with One Another

$$T_f = \frac{100 + 120}{2} + 460 = 570 \text{ Rankine (F absolute)}$$

$$u_m = 7 \text{ fps}$$

$$\rho = 0.076\left(\frac{520}{570}\right) = 0.0694 \text{ lb per cu ft}$$

$$D = 0.364 \text{ ft}$$

$$h_{c(\text{average})} = \frac{0.211(570)^{0.43}(7 \times 0.0694)^{0.8}}{(0.364)^{0.4}}$$

$$h_{c(\text{average})} = 2.73 \text{ Btu per (hr) (sq ft) (F deg)}$$

and

$$R_s \text{ (Forced Convection)} = \frac{1}{h_c A} = \frac{1}{2.73 \times 1.14}$$

$$R_s = 0.321 \text{ (hr) (F deg) per Btu.}$$

The radiation resistance, R_r, which acts in parallel with the resistance just calculated, can be computed with the aid of Fig. 7. The pipe wall, assumed at 100 F *sees* the surroundings at 120 F. If these two temperatures are used with Fig. 7, a value for $\frac{h_r}{F_A F_E}$ is determined directly.

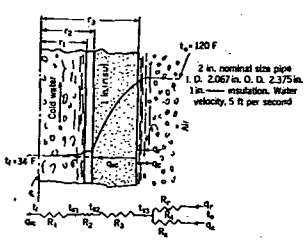


Fig. 8 .... Heat Transfer Conditions in an Insulated Cold Water Line

$$\frac{h_r}{F_A F_E} = 1.4 \text{ Btu per (hr) (F deg) (sq ft).}$$

The angle factor, F_A, is unity, and for an estimated surface emissivity of 0.95 (see Table 3), $F_E = 0.95$. Therefore,

$$h_r = 1.4\, F_A F_E = 1.4 \times 1 \times 0.95$$
$$h_r = 1.33 \text{ Btu per (hr) (F deg) (sq ft)}$$

and the radiation resistance, R_r, is then the following:

$$R_r = \frac{1}{h_r A} = \frac{1}{1.33 \times 1.14}$$

$$R_r = 0.659 \text{ (hr) (F deg) per Btu.}$$

The resultant resistance of R_s and R_r acting in parallel (see Fig. 8) can now be evaluated as:

$$\frac{1}{R_4} = \frac{1}{R_s} + \frac{1}{R_r} = \frac{1}{0.321} + \frac{1}{0.659} = 4.54 \text{ Btu per (hr) (F deg)}$$

$$R_4 = 0.216 \text{ (hr) (F deg) per Btu.}$$

The overall resistance, R_T, surroundings to cold water, is the sum of $R_1 + R_2 + R_3 + R_4 = 4.12$ (hr) (F deg) per Btu for 1-ft length of pipe. Note that the controlling resistances are R_3 and R_4, and that neglect of both R_1 and R_2 would not significantly influence the total resistance, R_T.

On the basis of this resistance calculation, the heat transfer from the surroundings to the cold water may be evaluated as:

$$\frac{q_{rc}}{N} = \frac{t_s - t_f}{R_T} = \frac{120 - 34}{4.12} = 20.8 \text{ Btu per (hr) (ft)}$$

or about 0.175 tons of refrigeration per 100 ft of pipe.

Since the calculation is based on a 1-ft pipe length,

$$q_{rc} = 20.8 \text{ Btu per hr.}$$

The temperature drops through the various resistances are now readily evaluated by Equation 16 as:

$$\Delta t = qR$$

$t_s - t_{s1}$ (air to insulation surface)
$= qR_4 = 20.8 \times 0.216 = 4.49$ F deg

$t_{s1} - t_{s2}$ (through the insulation)
$= qR_3 = 20.8 \times 3.9 = 81.2$ F deg

$t_{s2} - t_{s1}$ (through the pipe wall)
$= qR_2 = 20.8 \times 8.5 \times 10^{-4} = 0.018$ F deg

$t_{s1} - t_f$ (pipe wall to cold water)
$= qR_1 = 20.8 \times 3.73 \times 10^{-3} = 0.078$ F deg

This solution was obtained on the temperature distribution assumptions initially made. It is apparent that a better solution could be obtained if the whole problem were reiterated using the temperature distribution just calculated.

PERIODIC AND TRANSIENT HEAT FLOW

The foregoing data and examples dealt with steady-state heat transfer (not varying with time). In most practical

heat transfer problems the heat flow depends upon time. Such cases can usually be divided into two classes: periodic and transient. Periodic heat transfer repeats periodically in time. Transient heat transfer exhibits no periodicity. Graphical, analytical, and numerical methods are available for solving transient or periodic heat flow problems.[4,5,10,11,12] Graphical and numerical methods are the most versatile, and can be applied with minimum mathematical training.

A large number of analytical solutions for the case of heat conduction in variously shaped solids are available in the literature. Table 7 gives a summary of the cases reported and tabulated. Many more analytical solutions are available in the form of infinite series,[11,12,18,19] but are not tabulated. Certain complex cases may be treated by combining the simple analytical solutions as discussed in Reference 17. (See also Reference 20.)

Frequently, transient heat flow problems in one dimension have boundary conditions which make the problem difficult

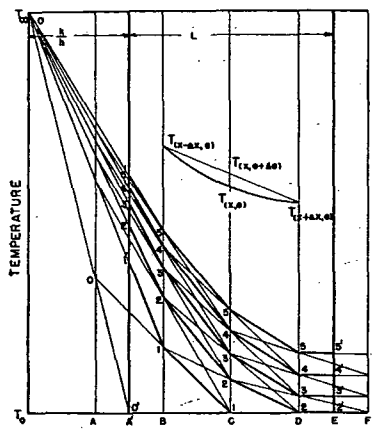


Fig. 9 .... Example of a Graphical Solution to a Problem in Transient Heat Conduction

to treat analytically. In such cases, recourse may be made to a graphical method of solution sometimes called the Schmidt method.[4,10,19,21,22] This method will be briefly outlined for the case of transient heat flow in a slab insulated on one face, and suddenly exposed on the other face through a fixed thermal resistance to a higher temperature. The technique is general, however, and methods may be devised for any boundary conditions,[5,22] and also, for one dimensional (radial) heat flow in spheres and cylinders.[22,23]

Consider the slab to be divided, as shown in Fig. 9, by n equidistant planes parallel to the slab surface and a distance Δx apart. Let the temperature of the slab at any plane and any time (θ) be denoted by $T_{x,\theta}$. Then the temperature of the slab at the two adjacent planes at the same time will be denoted as $T_{(x+\Delta x,\theta)}$ and $T_{(x-\Delta x,\theta)}$. In a similar manner the temperature of the x plane at a time $\Delta\theta$ later will be $T_{(x,\theta+\Delta\theta)}$.

In accordance with this nomenclature, the temperature at any plane x and time $\theta + \Delta\theta$ is given as

$$T_{(x,\theta+\Delta\theta)} = \frac{T_{(x+\Delta x,\theta)} + T_{(x-\Delta x,\theta)}}{2} \tag{20}$$

which may be interpreted as follows. The temperature of the slab at any plane, x, and any time, θ, is equal to the average temperature of the two adjacent planes obtained at the time $(\theta - \Delta\theta)$.

The time interval $\Delta\theta$ is determined by the equation

$$\Delta\theta = \frac{\Delta x^2}{2a} \tag{21}$$

Omitting the graphical construction at the slab boundaries, reference to Fig. 9 demonstrates the graphical method by means of which the temperature at each plane is determined at successive intervals of time in accordance with Equation 20.

For the problem stated, the boundary condition at the insulated surface is specified by the equation

$$\frac{\partial T}{\partial x} = 0$$

and at the uninsulated face by the equation

$$h(T_\infty - T) = -k\frac{\partial T}{\partial x}$$

In terms of finite differences these two equations (employing nomenclature established by Fig. 9) become

$$\frac{T_F - T_E}{\Delta x} = 0 \quad \text{or} \quad T_F = T_E \quad \text{at} \quad x = L$$

and

$$h(T_\infty - T_{A'}) = -k\frac{(T_B - T_A)}{\Delta x} \quad \text{at} \quad x = 0$$

or

$$\frac{T_\infty - T_{A'}}{k/h} = -\frac{T_A - T_B}{\Delta x}$$

The details of the graphical construction are best obtained by inspection of Fig. 9. Note that the line (0,0,0') used to initiate the graphical construction, is the only one drawn to the slab boundary A'. The numbered points indicate temperatures at the sub-slab boundaries at 1,2,3, etc., time intervals ($\Delta\theta$) after the slab is exposed to the high temperature.

For transient heat flow in two dimensions, and also, for steady-state conduction, numerical methods of solution are available in the literature.[5,10,12,19] These numerical methods are applicable to three-dimensional problems, although the calculations involved normally become too tedious for most applications of the method. An additional technique of solution for one- and two-dimensional problems in transient conduction results from the analogy of electrical resistance-capacitance networks to thermal systems.[24]

For two-dimensional problems in steady state conduction, additional techniques of solution are found in flux plotting,[5,19] and in the use of a potential tank.[5,19] These methods are of

Table 7 .... Analytical Solutions for Heat Conduction in Variously Shaped Solids

Shape of Solid	Boundary Conditions	Data Available in Graphs
Semi-infinite. 	Surface temperature suddenly changed.	Temperature distribution in solid as a function of time. References: (4) p. 37; (5) p. V-28; (10) p. 254; (13) p. 46.
		Heat flow from surface as a function of time. References: (10) pp. 256, 267; (13) p. 47.
	A steady flow of heat is suddenly applied to the surface.	Temperature distribution as a function of time. Reference: (10) p. 257.
	The surface temperature has been varying sinusoidally with time for a long time.	Temperature distribution as a function of time. Reference: (10) p. 296.
		Heat flow from surface as a function of time. Reference: (10) p. 296.
Semi-infinite with fluid at free surface. 	The temperature of the fluid in contact with the surface has a sudden change. (The surface conductance is constant.)	Temperature distribution as a function of time. References: (4) p. 37; (5) pp. V-45, 46, 47.
	The temperature of the fluid in contact with the surface has been varying sinusoidally with time for a long time. (The surface conductance is constant.)	Temperature distribution as a function of time. Reference: (10) p. 298.
		Heat flow from the surface as a function of time. Reference: (10) p. 298.
Slab. 	The temperatures t_1 and t_2 are suddenly changed from the initial uniform slab temperature to a new temperature. (The case where the surface on one side is insulated is treated by taking the case of a slab of twice the given thickness since the midplane has no heat flow due to symmetry.)	Temperature distribution as a function of time. References: (5) p. V-12; (10) p. 265.
	The temperatures t_1 and t_2 suddenly begin to increase as linear functions of time. The slab is initially at uniform temperature. (The case where one surface is insulated against heat flow is treated as noted above.)	Temperature distribution as a function of time. Reference: (10) p. 268.
	The temperature at both surfaces has been varying sinusoidally for a long time.	Temperature distribution as a function of time. Reference: (10) p. 300.
		Heat flow from the surface. Reference: (10) p. 303.

Table 7 .... Analytical Solutions for Heat Conduction in Variously Shaped Solids (Continued)

Shape of Solid	Boundary Conditions	Data Available in Graphs
Slab immersed in a fluid with constant conductance between fluid and slab surface. 	The temperature of the fluid is suddenly changed from the initial uniform slab temperature. (If one surface is insulated against heat flow, see above.)	Temperature distribution as a function of time. References: (4) pp. 32, 33, 34, 35; (5) pp. V-9, 10, 35, 42; (10) pp. 274, 284; (13) p. 106.
		Heat flow from the surface as a function of time. References: (5) p. V-10; (10) p. 274; (13) p. 107.
	The temperature of the fluid at one surface varies as a periodic function of time while the temperature of the fluid at the other surface is constant. The conductances need not be the same on both sides. (The variations in temperature are expressible as a Fourier series.)	Temperature distribution as a function of time. Reference 14.
		Heat flow at the surface as a function of time. Reference 14.
Cylinder of infinite axial dimension. 	The surface temperature is suddenly changed from the initial (uniform) temperature.	Temperature distribution as a function of time. Reference: (10) p. 265.
		Heat flow from surface as a function of time. Multiply temperature difference between surface and fluid by surface conductance.
	The surface temperature suddenly begins to increase linearly with time.	Temperature distribution as a function of time. Reference: (10) p. 269.
Cylinder of infinite axial dimension immersed in a fluid. 	The surrounding fluid suddenly changes from the initial (uniform) temperature of the cylinder.	Temperature distribution as a function of time. References: (4) p. 36; (5) pp. V-16, V-35, V-43, V-48; (10) pp. 278, 286; (15).
		Heat flow from the surface as a function of time. References: (5) p. V-16; (10) p. 278.
	The temperature of the surrounding fluid changes sinusoidally.	Temperature distribution as a function of time. Reference: (5) p. VI-34.
		Heat flow from the surface as a function of time. Reference: (5) p. VI-36.
Sphere. 	The temperature of the surface is suddenly changed from the initial uniform temperature.	Temperature distribution as a function of time. References: (5) p. V-23; (10) pp. 264, 265.
	The temperature at the surface suddenly begins to change as a linear function of time.	Temperature distribution as a function of time. Reference: (10) p. 269.

Table 7 . . . . Analytical Solutions for Heat Conduction in Variously Shaped Solids (*Concluded*)

Shape of Solid	*Boundary Conditions*	*Data Available in Graphs*
Sphere immersed in fluid. 	The temperature of the surrounding fluid suddenly changes from the initial uniform sphere temperature.	Temperature distribution as a function of time. References: (4) p. 36; (5) pp. V-21, V-35, V-44; (10) pp. 281, 282.
		Heat flow as a function of time. References: (5) p. V-21; (10) p. 281.
Rectangular bar of infinite length.	Any of the above noted boundary conditions for a slab.	Temperature distribution as a function of time. Combine solutions as indicated in Refs. 16 and 17.
Parallelopiped (rectangular).	Any of the above noted boundary conditions for a slab.	Temperature distribution as a function of time. Combine solutions as indicated in Refs. 16 and 17.
Cylinder of finite length.	Any of the boundary conditions given above for a cylinder and a slab.	Temperature distribution as a function of time. Combine solutions as indicated in Refs. 16 and 17.
Hollow cylinder of infinite exterior radius.	The temperature of the surface suddenly changes from the initial (uniform) temperature.	Temperature distribution as a function of time. Combine solutions as indicated in Refs. 16 and 17.
		Heat flow at the surface as a function of time. Reference: (10) p. 267.

most direct use in problems in which the boundaries of the two-dimensional shape are made up of isothermal and adiabatic surfaces.

REFERENCES

[1] T. K. Sherwood: *Absorption and Extraction* (McGraw-Hill Book Co., New York, 1937).

[2] Griffith and Davis: *The Transmission of Heat by Radiation and Convection* (*Department of Scientific and Industrial Research Special Report* No. 9, His Majesty's Stationery Office, London, 1933).

[3] A. P. Colburn: A method of correlating forced convection heat transfer data and a comparison with fluid friction (*American Institute of Chemical Engineers Transactions*, Vol. 29, 1933, p. 174).

[4] W. H. McAdams: *Heat Transmission* (McGraw-Hill Book Co., New York).

[5] L. M. K. Boelter, V. H. Cherry, H. A. Johnson, and R. C. Martinelli: *Heat Transfer Notes* (University of California Press, Berkeley, 1946).

[6] V. S. Touloukian, G. A. Hawkins, and Max Jakob: Heat transfer by free convection from heated vertical surfaces (*ASME Transactions*, 1948, p. 13).

[7] R. H. Heilman: Heat insulation in air conditioning (*Industrial and Engineering Chemistry*, Vol. 28, 1936, p. 782).

[8] L. M. K. Boelter, J. T. Gier, and F. A. Ryder: Photoelectric photometer for rapid comparison of two light sources (*Illuminating Engineering Society Transactions*, 1939).

[9] Perry Moon: *Scientific Basis of Illuminating Engineering* (McGraw-Hill Book Co., New York, 1936).

[10] Max Jakob: *Heat Transfer* (John Wiley & Sons, New York, Vol. 1, 1949).

[11] L. E. Grinter: *Numerical Methods in Engineering* (The Macmillan Co., New York, 1949).

[12] G. M. Dusinberre: *Numerical Analysis of Heat Flow* (McGraw-Hill Book Co., New York, 1949).

[13] Max Jakob: *Elements of Heat Transfer and Insulation* (John Wiley & Sons, New York, 1942).

[14] H. A. Johnson: Periodic heat transfer at the inner surface of a homogeneous wall (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 143).

[15] M. P. Heisler: Temperature charts for induction and constant temperature heating (*ASME Transactions*, Vol. 69, 1947, p. 227).

[16] F. C. W. Olsen: Temperatures in solids during heating and cooling (*Industrial and Engineering Chemistry*, Vol. 34, 1942, p. 874).

[17] T. K. Sherwood and C. E. Reed: *Applied Mathematics in Chemical Engineering* (McGraw-Hill Book Co., New York, 1939).

[18] H. S. Carslaw: *Introduction to the Mathematical Theory of the Conduction of Heat in Solids* (Dover Publications, New York, 1945).

[19] L. R. Ingersoll, A. C. Ingersoll, and O. J. Zobel: *Heat Conduction* (McGraw-Hill Book Co., New York, 1948, p. 209).

[20] H. P. Gurney and J. Lurie: Charts for estimating temperature distribution in heating or cooling solid shapes (*Industrial and Engineering Chemistry*, Vol. 15, 1923, p. 1170).

[21] T. K. Sherwood: *Applied Mathematics in Chemical Engineering* (McGraw-Hill Book Co., New York, p. 241).

[22] A. Nessi and L. Nissolle: *Méthodes Graphiques pour L'étude des Installations de Chauffage et de Réfrigération en Régime Discontinu* (Dunod, Paris, 1949).

[23] R. L. Perry and W. P. Berggren: *Transient Heat Conduction in Hollow Cylinders after Sudden Change of Inner Surface Temperature* (University of California, *Publications in Engineering* 5, Vol. 59, 1944).

[24] V. Paschkis: Method for determining unsteady-state heat transfer by means of electrical analogy (*ASME Transactions*, Vol. 64, 1942, p. 105).

CHAPTER 6

PHYSIOLOGICAL PRINCIPLES

Chemical Vitiation of Air, Physical Impurities in Air, Thermal Interchanges Between the Body and Its Environment, High Temperature Hazards, Acclimatization, Upper Limits of Heat for Men at Work, Application of Physiologic Principles to Air-Conditioning Problems, Effective Temperature Index and Comfort Zones

VENTILATION is defined in part as *the process of supplying air to, or removing air from, any space by natural or mechanical means.* The word in itself implies quantity, but air must be of the proper quality also. The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. The ASHVE *Code of Minimum Requirements for Comfort Air Conditioning*[1] defines it "as the process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. If an installation cannot perform all of these functions, it shall be designated by a name that describes only the function or functions performed." See definition of air conditioning in Chapter 1.

CHEMICAL VITIATION OF AIR

People living indoors bring about certain physical and chemical changes in the air about them. The oxygen content of the air diminishes and the carbon dioxide increases, but these changes are too slight to be significant except in airtight spaces as in submarines. Organic matter which is usually perceived as odors, comes from the body or clothes. Moisture and heat are given off by the body. There is no evidence of any toxic volatile material given off by man to the ambient air. Stale air may be offensive because of odors and may induce loss of appetite and loss of energy. Objectionable body odors have the same effects. These reasons, whether esthetic or physiological, usually make it desirable in the design of air-conditioning systems to provide for the elimination or control of odors arising from occupancy, cooking, or other sources. This may be accomplished by introducing odor-free air in sufficient quantities to reduce odor concentrations by dilution to a level which is not objectionable. Odor-free air may be outdoor air or air which has been cleared of odors by sorption, washing, or other appropriate means.

In the case of vitiation by a few hazardous gases such as carbon monoxide from heating, cooking, and certain industrial processes, no satisfactory chemical treatment for the elimination of the impurity has been found. The only satisfactory solution is elimination at the source by local exhaust ventilation; or, if this is impossible, reduction to a safe concentration by dilution. (See Chapter 7.) In the case of contamination by other matter, including volatile vapors and gases, chemical treatment for the removal or reduction of the impurities has been made available through air cleaning methods, which are discussed in Chapter 24.

When the only source of contamination is the human occupant, and overheating is not a problem, the minimum quantity of outdoor air needed appears to be that required to remove objectionable body odors, or tobacco smoke. The concentration of body odor in a room, in turn, depends upon a number of factors, including the dietary and hygienic habits of the occupants (frequently reflecting their socio-economic status), the outdoor air supply, air space allowed per person, odor adsorbing capacity of air-conditioning processes, and temperature and relative humidity. The intensity of odor sensation has been found to vary as the logarithm of the concentration of the odoriferous substance in the air, or inversely with the logarithmic function of the amount of outdoor air supplied and the air space per person.

The relation between air supply and occupancy has been reported by the Harvard School of Public Health[2] (Table 1) and the ASHAE Research Laboratory.[3] Outdoor air require-

Table 1 . . . . Minimum Outdoor Air Requirements to Remove Objectionable Body Odors Under Laboratory Conditions[2]

Type of Occupants	*Air Space per Person Cu Ft*	*Outdoor Air Supply CFM per Person*
Heating season with or without recirculation. Air not conditioned.		
Sedentary adults of average socio-economic status	100	25
	200	16
	300	12
	500	7
Laborers	200	23
Grade school children of average socio-economic status	100	29
	200	21
	300	17
	500	11
Grade school children of lower socio-economic status	200	38
Children attending private grade schools	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person.		
Sedentary adults	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Spray water changed daily. Total air circulation 30 cfm per person.		
Sedentary adults	200	<4

ments for removal of objectionable tobacco smoke odors are not accurately known, but available information and current practice indicate the need of 15 cfm per person or more.[4]

The total quantity of outdoor air to be circulated through an enclosure is often governed chiefly by the physical considerations for controlling temperature, air distribution, and air velocity. Other factors which must be taken into consideration, include the type and usage of the building, locality, climate, height of rooms, floor area, window area, extent of occupancy, and the operation of the system distributing the air supply. Frequently, some of these factors, particularly the need for air movement and good distribution, may be satisfied by recirculation of inside air rather than outside air.

It will be noted that, with adequate air space, the rate of air change indicated in Table 1 is from 4 to 30 cfm per person. In rooms occupied by only a few persons, such an air change will be automatically attained in cold weather by normal leakage around doors and windows, and can easily be secured in warm weather by the opening of windows. With a space allotment of 400 cu ft per person, only 1½ air changes per hour are necessary to provide a ventilation rate of 10 cfm per person.

Therefore, in the ordinary dwelling with adequate cubic space allotment, no special provision for controlling chemical purity of the air is necessary (aside from removal of fumes from heating appliances). For such conditions, the control of air temperature is the major consideration.

In more crowded rooms (large offices, large workrooms, auditoriums), where the cubic space per person is less and it is usually impossible to admit untempered outside air without creating drafts, mechanical ventilation is essential.

The present data regarding the effect of cubic space on outdoor air requirements are not universally accepted. *The Code of Minimum Requirements for Comfort Air Conditioning*[1] prescribes definite minimum requirements which should be familiar to the designing engineer. It should be emphasized, however, that the code fixes *minimum*, rather than *adequate* requirements. (See also Tables 1 and 3 of Chapter 13.)

Notwithstanding the rapid advance made in air conditioning, some persons still believe there is a stimulating quality in outdoor air (particularly country, mountain, and seashore air) under ideal weather conditions, which is lacking in artificially conditioned air. It is apparent, however, that modern air conditioning insures control of the phenomena of nature for the service and comfort of man independently of weather conditions. Freedom of movement, action, and thought, together with the variability of stimuli experienced by persons under ideal conditions in the country, mountains, or seashore, undoubtedly has some stimulating effect. Various experimenters have attempted to duplicate the invigorating qualities of outdoor air by the use of ozone, ionization, or ultraviolet light, but results to date have been inconclusive or negative.[5]

Ozone in amounts of 0.01 to 0.05 ppm of air is allowable in comfort air conditioning. Above this limit there is a pungent, unpleasant odor and perhaps respiratory distress, depression, and stupor.[6]

PHYSICAL IMPURITIES IN AIR

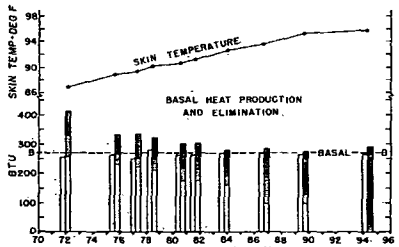
Dust particles of almost any type can produce irritation of the mucous membranes of the nose and throat, if present in high concentrations. Certain dusts may be very harmful, but coal dust is tolerated well. The effects of various industrial dusts, pollens, etc., are discussed in Chapter 7.

A certain part of the dissemination of disease in confined spaces may be due to pathogenic organisms floating in the air like dust particles. (See Chapter 8.)

While in some instances it may be possible to reduce the physical impurities of the air by dilution from a non-contaminated source, such a source is rarely available. Frequently, outdoor air contains a higher concentration of physical impurities than indoor air. Therefore, it is usually desirable to reduce the concentration of physical impurities by air cleaning methods. (See Chapter 24.)

THERMAL INTERCHANGES WITH ENVIRONMENT

Body temperature depends upon the balance between heat production and heat loss. Heat resulting from oxidation in the body (metabolism) maintains the body temperature well above that of the surrounding air in a cool or cold environment. At the same time, heat is constantly lost from the body by radiation, convection, and evaporation. Since, under ordinary conditions, the body temperature is maintained at its



** Normal control, naked, in calorimeter at temperatures from 72.8 to 94.1 F. First column in each experiment represents heat production as determined by indirect calorimetry, the second column, heat elimination. The portion marked with vertical lines represents vaporization, the dotted area convection, the unmarked area, radiation. The skin temperature represents the average reading of 18 spots on the surface.*

Fig. 1 .... Heat Loss from Human Being by Evaporation, Radiation, and Convection*

normal level of about 98.6 F, the heat production must be balanced by the heat loss. During work, the body temperature may rise; in fact, afternoon temperatures of normal persons average 1 deg above the resting value of the morning whether working or not.

The fundamental thermodynamic processes concerned in heat interchanges between the body and its environment may be described by the equation:

$$M = \pm S + E \pm R \pm C \qquad (1)$$

where

M = rate of metabolism, heat produced within the body.
S = rate of storage, change in intrinsic body heat.
E = rate of evaporative heat loss.
R = rate of radiative heat loss or gain.
C = rate of convective heat loss or gain.

The rate of metabolism, M, is always positive. The storage, S, may be either positive or negative, depending upon whether

heat is being stored or depleted owing to a rise or fall in body temperature. Under ordinary circumstances (when the dew point of the air is below the body surface temperature) the evaporation loss, E, is always positive; that is, heat from metabolism supplies this loss. R and C are positive when body surface temperature is above that of walls and air, and negative when it is below.

DuBois,[7] after careful calorimeter studies on a fasting, nude man, plotted the partition of body heat loss and heat production as a function of temperature. Fig. 1 shows some disparity between heat production and heat loss. This disparity is S in Equation 1. In the central range of the experiments S was quite low and no increase in heat loss by vaporization was apparent.

Within the range of 81–86 F air temperature, with still air, there is, for a resting nude man, a point at which his body

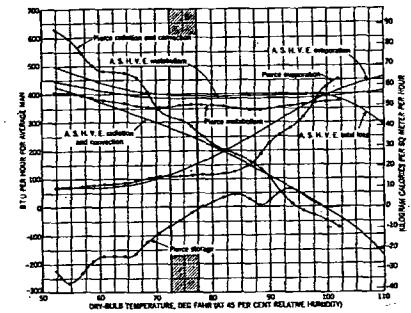


Fig. 2 .... Relation Between Metabolism, Storage, Evaporation, Radiation Plus Convection, and Temperature for the Clothed Subject

has to take no particular action to maintain its heat balance. If he is clothed, or if he is active, this point will naturally lie at a lower level. At this point, which may be termed the *neutral point for that individual*, conditions are neither too hot nor too cold. If, through a fall in air temperature, or a rise in air movement, the rate of heat loss from the skin to the environment is increased, then the body must do something to counteract this heat loss. Over a certain range, the body can achieve this by decreasing the flow of blood through the skin. This will result in some cooling of the skin and subjacent tissues, but the temperature of the deep tissues will be preserved. The range of external conditions over which this may be achieved may be termed the *zone of vaso-motor regulation against cold*. Beyond this range, the temperature of the superficial tissues will fall still further, and that of the deep tissues will fall as well, unless some other steps are taken. The body normally does react; it increases heat production by increasing muscular tension, by shivering, or by spontaneous increase in activity.[8] As long as these are adequate to meet the increased rate of heat loss to the environment, a fall in deep body temperature may be prevented. Such conditions may be said to lie in the *zone of metabolic regulation against cold*. Beyond this point, the body enters the *zone of inevitable body cooling*. Once body temperature starts to fall, man is headed for disaster.

It will be seen that, in man, deep body temperature is preserved over an important range of cold external conditions, at the expense of (1) a fall in the temperature of the peripheral tissues, and (2) an increased expenditure of energy. As regards the first of these, the farther away superficial tissue lies from the central body mass, the more readily will its temperature fall.

On the hot side of the neutral point, there exists a *zone of vaso-motor regulation against heat*, corresponding to that against cold. The blood flow through the skin is increased when the opportunities for heat loss to the environment are restricted.[9] This increase in blood flow may double the conductance of the superficial tissues over that characteristic of the neutral point, and the temperature of the skin surface may rise until it is only three degrees below that of the deep tissues. If this increase in blood flow is unable to balance the restriction in heat loss, the body has entered the next zone. Once again, the normal body takes steps to prevent a change in its deep temperature; but they are not the counterpart of the steps taken under similar circumstances on the cold side. There is, in fact, very little change in heat production, beyond that resulting from a disinclination for exertion. The second line of defense, on the hot side, is a new and powerful method of promoting heat loss—the provision of water, by the operation of the sweat glands, for evaporative cooling. As long as evaporation is adequate to restore the desired heat loss, the body is in the *zone of evaporative regulation against heat*. When this ceases to be adequate, the body is in the *zone of inevitable body heating*. The body enjoys a little more latitude in this zone than it did in the corresponding zone on the cold side, but when the deep temperature rises more than 4 deg F, it loses its efficiency. All factors which affect the evaporation of water from the skin affect heat regulation in the zone of evaporative regulation. Atmospheric vapor pressure and air motion are most important. With dry-bulb temperature above body temperature, air motion facilitates evaporative heat loss by removing hot humid air from contact with the skin and replacing it with relatively drier air.

Heat regulation in man requires an intact set of sensory nerves, a normal sympathetic nerve supply to sweat glands and blood vessels, a great many sweat glands, and a circulatory system capable of carrying heat from muscles and viscera to the skin by circulation of the blood.

Some of the phenomena of body temperature control are shown graphically in Fig. 2. The dotted curves, from a study at the John B. Pierce Laboratory of Hygiene,[10] are for subjects lightly clothed in a semi-reclining position, and give the relation between the dry-bulb temperature of the environment (with about 45 percent relative humidity) and the metabolic rate (heat production), the rate of heat dissipation by radiation and convection combined, and the latent heat loss due to evaporation from the skin and the respiratory tract. The smooth line curves from the work of the ASHAE Research Laboratory[11] give the same relationships for healthy, male subjects (18 to 24 years of age), seated at rest and dressed in customary winter indoor clothing. The Pierce Laboratory data for the semi-reclining subjects also include the rate of heat storage (either positive or negative) due to a rise or fall in body temperature. For the normally clothed subjects, a curve gives the total heat loss (that is, the sum of the radiation, convection, and evaporative losses). Here,

Table 2 .... Physiological Responses to Heat of Men at Rest and at Work[a]

Effective Temp	Actual Cheek Temp (Fahr Deg)	Men at Rest			Men at Work 90,000 ft-lb of Work per Hour			
		Rise in Rectal Temp (Fahr Deg per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-Lb)	Rise in Body Temp (Fahr Deg per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt by Perspiration (Lb per Hr)
60	....	....	....	....	225,000	0.0	6	0.5
70	....	0.0	0	0.2	225,000	0.1	7	0.6
80	96.1	0.0	0	0.3	209,000	0.3	11	0.8
85	96.6	0.1	1	0.4	190,000	0.6	17	1.1
90	97.0	0.3	4	0.5	153,000	1.2	31	1.5
95	97.6	0.9	15	0.9	102,000	2.3	61	2.0
100	99.6	2.2	40	1.7	67,000	4.0[b]	103[b]	2.7[b]
105	104.7	4.0	83	2.7	49,000	6.0[b]	158[b]	3.5[b]
110	....	5.9[b]	137[b]	4.0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a] Data by ASHAE Research Laboratory.
[b] Computed value from exposures lasting less than one hour.

storage is given by the difference between the metabolism and total heat loss.

The small difference between the metabolic rates for the two groups of subjects may be accounted for by difference in activity. Heat exchange between the body and the environment by radiation and convection is greater for the lightly clothed subject, both for cool conditions where there is excessive heat loss, and for very warm conditions where there is transfer of heat from the atmosphere to the body. The two curves for evaporative loss serve to show how physiological control uses evaporation of sweat to maintain equilibrium at high temperatures. Below 75 F, for the normally clothed subject, and below 85 F for the lightly clothed subject, evaporation loss is minimal and constant. Burch[12] has shown that this insensible perspiration reflects the permeability of the skin to the moisture of the body. Above these temperatures, control is obtained by the availability of sweat for evaporation. The difference in the curves above 75 F is probably largely determined by the difference in clothing and activity.

In the zone of evaporative heat regulation, air movement facilitates heat loss if the temperature of the air is not above that of the skin.[13] Under hot, dry conditions air movement may be of little advantage, or even of disadvantage, if it increases the addition of heat to the skin by conduction more than it promotes the loss of heat from the skin by evaporation.

HIGH TEMPERATURE HAZARDS

Studies at the ASHAE Research Laboratory[14] and elsewhere during the past two decades have made available much information dealing with the physiological effects of hot atmospheres on workers, and means of alleviating the distress and hazards associated therewith. Table 2 gives some of the physiological responses of men, at rest and at work, to hot environments. Frequent and continued exposure of workers to hot environments results in physiological derangement affecting the leucocyte count of the blood, and other factors dealing with man's mechanism of defense against infection.

Wherever S (Equation 1) becomes strongly positive and body temperature rises progressively, men will continue to work until body temperature reaches 103 F. When these body temperatures are exceeded, men work with declining efficiency and may be subject to heat stroke.

Heat exhaustion is a circulatory failure is which the venous return to the heart is reduced so that fainting results.[15] Early symptoms of heat exhaustion may include fatigue, headache, dizziness when erect, loss of appetite, nausea, abdominal distress, vomiting, shortness of breath, flushing of face and neck, pulse rate above 150, glazed eyes, and mental disturbances such as apathy, poor judgment, and irritability which usually precede fainting (syncope). Recovery is usually prompt when the man is removed to a cool place and kept lying down for a time, unless he has some other illness such as heart disease.

Heat cramps are painful muscle spasms in extremities, back, and abdomen due, at least in part, to excessive loss of salt in sweating. Formerly common in hot industries, this manifestation of illness due to heat is now greatly reduced by drinking water containing 0.1 percent salt, or by proper use of salt tablets. Heat cramps are readily alleviated by administration of salt solution intravenously.

Heat stroke is a serious effect of exposure to great heat. The body temperature climbs rapidly to excessive levels often above 105 F when for unknown reasons free sweating suddenly stops. At such high temperatures, coma appears and death may be imminent. Emergency measures are required to reduce the excessive body temperature by cooling quickly to avoid irreparable damage to the brain.[16]

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness, and mortality. Both laboratory and field data show that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises.

ACCLIMATIZATION

When men move to deserts or to jungles some adaptation to the climate takes place. If work is gradually increased day by day, and if the men can get plenty of water and salt, and can sleep each night, acclimatization may be complete in 7 to 10 days. The acclimatized man works with a lower heart rate, lower skin and rectal temperature, and more stable blood pressure than when unacclimatized. The process of acclimatization requires work in the heat.[17] During the recent war, white troops lived and did hard physical work for long periods in tropical conditions when disease hazards were controlled.

Table 3 .... Upper Limits of Environmental Conditions for Acclimatized, Healthy, Young Men in Military Service

Environment	Reactions at the end of 4 hr	
	Rectal Temp F	Pulse Rate
Relatively easy	Below 101	Below 130
Difficult	101 to 102	130 to 145
Impossible	Above 102	Over 145

In recent tests made at the ASHAE Research Laboratory,[18] subjects were required to perform light work under very hot conditions for a 4-hr period each day. It was found that the ability of a new subject to endure these conditions showed daily improvement for a period of at least 2 weeks. However, after acclimatization was completed, a recess of several days had no effect on the endurance of the subject. Individuals differ widely in their capacity to acclimatize. Acclimatized men lose most of these improvements in a few weeks of temperate climate, even though they are vigorously active. In the course of acclimatization, the sweat glands come to secrete fluid less rich in salt.[18] For all except those carrying out really hard work in hot, dry atmospheres, this effects an important saving in salt loss, and makes all the difference between being exposed and not being exposed to the risk of heat cramps.

The adaptive level changes somewhat with the season. There are also marked differences between the sexes. In the cold zone the thickness of thermal insulating tissues of women is almost double that of men, although the sensory responses to cold are similar. In the hot zone, the threshold of sweating is higher for women. The thickness and insulating value of the clothing worn are also important factors in the determination of the comfort level.

UPPER LIMITS OF HEAT FOR MEN AT WORK

In very hot conditions humidity is the limiting factor, and the wet-bulb temperature assumes great importance. In 1905 Haldane recognized that 88 F wet-bulb was the limit of endurance for coal miners, and later observers have concurred.

A study was made at the Armored Medical Research Laboratory[20] to determine the upper limits of environmental conditions under which a man can perform certain work. Thirteen enlisted men, thoroughly acclimatized to the hot conditions, served as subjects. During each test, the subjects were required to march for 4 hr at the rate of 3 mph, carrying 20 lb packs under a wide range of environmental conditions which were rated as relatively easy, difficult, and impossible, on the basis of the physiological reactions of the subjects at the end of the 4-hr period as shown in Table 3 and Fig. 3.

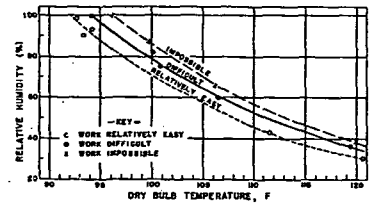


Fig. 3 .... Heat Endurance of Acclimatized Subjects Working at a Specific Rate[20]

Recognition of the need of air conditioning for workers in hot industries is growing rapidly. The choice of the type of system to be used in any given instance, must be determined by the air-conditioning engineer after a study of conditions (see Chapter 51, Control of Industrial Environment). In some hot industries where few people are working in large spaces the worker himself, rather than the atmosphere, can be cooled by placing him in a small booth, and blowing cooled air over him, or by circulating cooled air through a loose-fitting suit.[21]

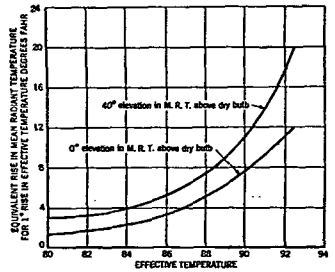
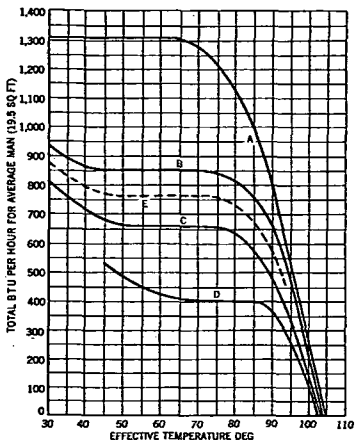


Fig. 4 .... Effect of MRT Elevation in Terms of Effective Temperature

The ASHAE Laboratory has studied the effects of walls of higher temperature than the air.[22] The findings are in part shown in Fig. 4. It will be seen that the importance of mean radiant temperature, as compared with that of the effective temperature, decreases as the effective temperature rises; and also, to a certain extent, as the mean radiant temperature itself rises. The lower of the two curves relates to conditions in which the MRT was kept approximately at the level of the DBT. If this curve is followed, it will be seen that, at 80 ET, a little more than 1 deg rise in MRT produces the same effect as 1 deg rise in ET; but when the ET is 92 deg, it takes a rise of 11 F deg in the MRT to produce the same effect as 1 deg rise in ET. The upper curve relates to conditions in which the MRT was kept about 40 F deg higher that the DBT. It will be seen that under these conditions, a rise in MRT is less effective, even at low values of ET; and that it loses its relative effectiveness more rapidly as the ET rises. It should not be assumed, however, that MRT does not matter much. All that these comparisons indicate is that unit rise in MRT becomes less important *as compared with unit rise in ET*, as conditions get hotter. This may be due more to a growing importance of unit rise in ET than to a diminishing



[a] *Curve A—Persons working, metabolic rate 1310 Btu per hour. Curve B—Persons working, metabolic rate 850 Btu per hour. Curve C—Persons working, metabolic rate 660 Btu per hour. Curve D—Persons seated at rest, metabolic rate of 400 Btu per hour. Curves B and D based on test data covering a wide temperature range. Curves A and C based on test data at an Effective Temperature of 70 and extrapolation of Curves B and D. All curves are averages of values for high and low relative humidities; variation due to humidity is small.*

Fig. 5 .... Relation Between Total Heat Loss from the Human Body and Effective Temperature for Still Air[a, 14]

importance of unit rise in MRT. Under ordinary *still air* conditions the effects of air temperature and MRT appear to be interdependent. Various authorities give 0.3 to 1 deg increase of room temperature to compensate for 1 deg depression of the MRT.

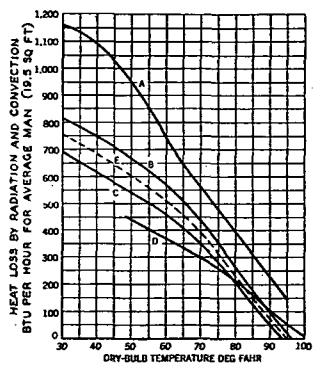
APPLICATION OF PHYSIOLOGIC PRINCIPLES TO AIR-CONDITIONING PROBLEMS

In order to estimate cooling loads in occupied spaces it is necessary to know the metabolic rate (heat production) of man. This has been studied extensively, and found to remain relatively constant per unit of body surface area in a subject fasting and resting quietly after a good night's sleep. The rate is high in children, and diminishes gradually with age; it increases in certain diseases and in the presence of fever. The metabolic rate is somewhat lower in women. Heat production goes up sharply with work and varies widely in different persons doing the same work. Figs. 5, 6, and 7 and Table 27 of Chapter 13 give sufficient basic data for estimating heat production and heat loss under various conditions.

EFFECTIVE TEMPERATURE INDEX AND COMFORT ZONES

There is no precise physiologic observation by which comfort can be evaluated. Mean skin temperature offers some promise. The zone of thermal neutrality differs with clothing, season, activity, and all the other factors controlling heat production (Table 4). The comfort zone is very similar to the zone of thermal neutrality.

Sensations of warmth or cold depend not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by a wet-bulb thermometer, upon air movement, and upon radiation effects. Dry air at a relatively high temperature may feel cooler than air of lower temperature with a high moisture content. Air motion makes any moderate condition feel cooler. Radiation to cold or from warm surfaces is another important factor under certain conditions affecting the comfort reaction of the individual.



[a] *See footnote a, Fig. 5.*

Fig. 6 .... Relation Between Radiation and Convection Loss from the Human Body and Dry-Bulb Temperature for Still Air[a, 14]

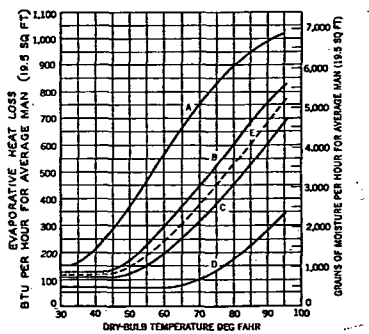
Combinations of temperature, humidity, and air movement which induce the same feeling of warmth are called thermo-equivalent conditions. A series of studies[22] at the ASHAE Research Laboratory established the equivalent conditions for practical use. This scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also to a considerable degree determines the physiological *effects* on the body induced by heat or cold. For this reason, it is called the *effective temperature* scale or index, and it denotes sensory heat level.

Effective temperature is an empirically determined index of the degree of warmth perceived on exposure to different combinations of temperature, humidity, and air movement. It was determined by trained subjects who compared the relative warmth of various air conditions in two adjoining conditioned rooms by passing back and forth from one room to the other.

The numerical value of the index for any given air conditions is fixed by the temperature of slowly moving (15 to

25 fpm air movement) saturated air which induces a like sensation of warmth or cold. Thus, any air condition has an effective temperature of 60 deg when it induces a *sensation* of warmth like that experienced in slowly moving air at 60 F saturated with moisture. The *effective temperature index* cannot be measured directly, but is determined from dry- and wet-bulb temperatures and air motion observations by reference to an Effective Temperature Chart (see Figs. 8, 9, and 10) or tables.

Fig. 8 gives the effective temperature for any combination of dry- and wet-bulb temperatures for *still air* (15 to 25 fpm) conditions. Charts similar to Fig. 8 for air velocities of 100 and 300 fpm have been presented in earlier editions of THE GUIDE, e.g., 1939. Fig. 9 is another form of effective



[a] *See footnote a, Fig. 5.*

Fig. 7 .... Evaporative Heat and Moisture Loss from the Human Body in Relation to Dry-Bulb Temperature for Still Air Conditions[a, 14]

temperature chart embodying all three variables: dry-bulb and wet-bulb temperatures, and air velocity.

As stated previously, effective temperature is an index of the degree of warmth experienced by the body. An effective temperature line is, therefore, a line defining the various combinations of conditions which *will induce like sensations* of warmth. It does not necessarily follow that like sensations of comfort will also be experienced along the entire length of an effective temperature line. Some degree of discomfort is likely to be experienced at very high or very low relative humidities, regardless of the effective temperature. It has also been found that the optimum effective temperature varies with the season, and is lower in winter than in summer.

Tests[14] made at the ASHAE Research Laboratory in very hot conditions, with subjects doing light work, were in very close agreement with the effective temperature chart. Other work[23] under similar environmental conditions, but with subjects walking 3 mph and carrying 20 lb packs, indicated that the effective temperature lines should be more nearly horizontal. It therefore appears that the slope of the ET lines may vary, depending upon the rate of work being performed.

Table 4 .... Comparison of Comfort Ranges With Zone of Thermal Neutrality

Investigators	Effective Temperature: Optimum Line	Effective Temperature: Range	Operative Temp Range	Remarks
Comfort Zone				
Houghten and Yaglou	66	63–71	...	Winter nonbasal; at rest, normally clothed. Men and women.
Yaglou and Drinker	71	66–75	...	Summer nonbasal; at rest and normally clothed. Men.
Yaglou	72.5	66–82	...	Entire year; nonbasal; at rest and stripped to waist. Men.
Keeton et al	75	74–76	...	Entire year; basal, nude. Steady state (9 hr exposure). Men and women.
Zone of Thermal Neutrality				
DuBois and Hardy	75	73.2–76.9	...	Basal; nude; men.
	71.5	64.8–76.0	...	Basal; clothed; men.
Winslow, Herrington, and Gagge	...	...	84.0–87.8	Nonbasal; at rest; nude; men.
	...	...	74 –84	Nonbasal; at rest; clothed; men.

Fig. 10 shows the ASHAE Comfort Chart[24] as published since 1950. The areas and arrows indicating the summer and winter comfort zones on the previous charts have been removed. The summer comfort zone was removed because it extended to temperatures where too large a percentage of the people would be uncomfortable. The winter comfort zone was removed for the same reason, and because of inadequate data in later studies.

The distribution curve, showing the percent of people feeling comfortable at various effective temperatures in *summer, indicates* that a maximum of 98 percent of the people were comfortable at 71 ET. The study was conducted with relative humidities between 30 and 70 percent.

The distribution curve shown on the previously used chart, showing the percent of people feeling comfortable at various effective temperatures in the *winter*, was based on research prior to 1932. This curve indicated that at 66 ET a maximum number of people were comfortable. Later studies[25] by the ASHAE Research Laboratory indicated that a maximum of 97.7 percent of the people were comfortable at 68 ET, and this finding has been confirmed by current practice.[26] However, adequate data from the later studies were available only for the ET range of 65 to 69, as presented in Fig. 10. The studies should be extended to cover a wider range. The lighter weight clothing, probably worn in the later studies, accounts for the higher desirable ET.

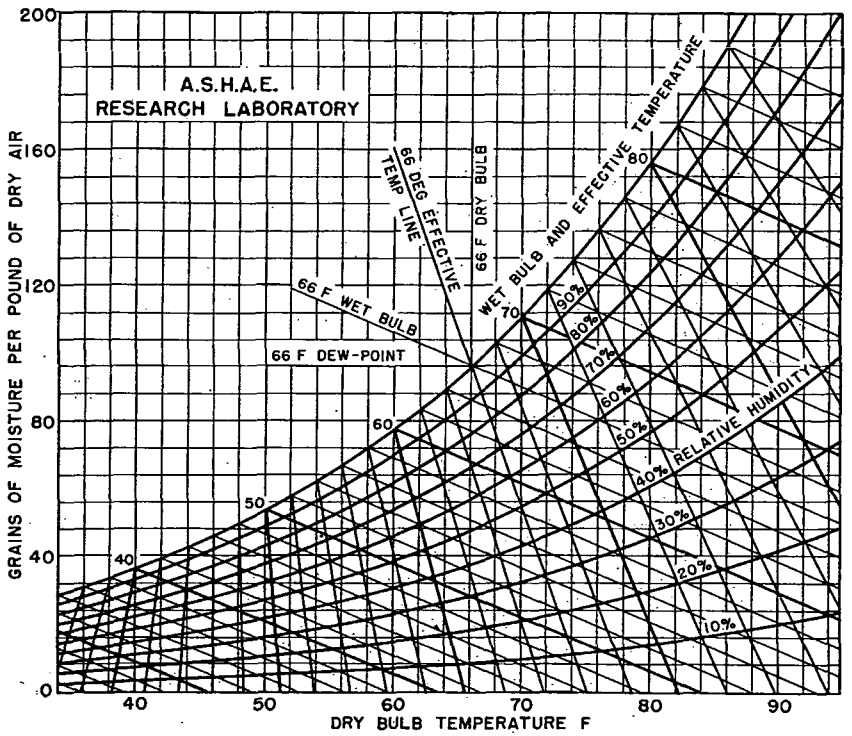


Fig. 8 .... Psychrometric Chart, Persons at Rest, Normally Clothed, in Still Air

Radiation from occupants to room surfaces, and between the occupants, has an important bearing on the feeling of warmth, and may alter to some measurable degree the optimum conditions for comfort previously indicated. Since the mean radiant temperature of a space is affected by cold walls and windows, as well as by the warm surfaces of heating units placed within the room or embedded in the walls, these factors must be compensated. Likewise, in densely occupied spaces, such as classrooms, theaters, and auditoriums, temperatures somewhat lower than those indicated by the comfort line may be desirable because of counter-radiation between the bodies of occupants in close proximity to each other. Such radiation will also elevate the mean radiant temperature of the room.

Many field studies[23] have been made to determine the optimum indoor effective temperature for both winter and summer in several metropolitan districts of the United States and Canada, in cooperation with the managements of offices employing large numbers of workers (Fig. 11). On the whole, women of all age groups studied prefer an effective temperature for comfort 1.0 deg higher than men. All men and women over 40 years of age prefer a temperature 1 deg ET higher than that desired by persons below this age. The persons serving in all of these studies were representative of office workers dressed for air-conditioned spaces in the summer season, and engaged in the customary office activity.

On the basis of present knowledge, for different geographical regions and age groups, the most popular temperature varies from a low of 66 ET in winter to a high of 73 ET in summer. The spread for summer comfort is 69 to 73 ET.

A spread of 3 deg in the optimum effective temperature for summer cooling is ascribed to geographical location. However,

variations in sensation of comfort among individuals may be greater for any given location than variations due to a difference in geographical location. The available information indicates that changes in weather conditions over a period of a few days do not alter the optimum indoor temperature.

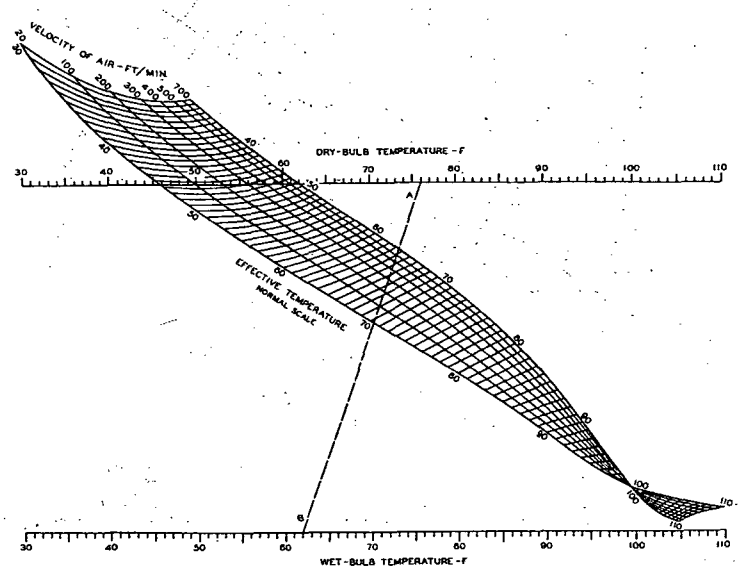
Laboratory studies of Keeton and others,[22] and years of practical field experience show that the adjustments in the heat regulatory mechanisms of healthy people on entering and leaving air-conditioned spaces are prompt, easy, and safe. Sudden sensation of coolness on entering a cooled room from the hot street is associated with sweat on the skin and in the clothing. Several minutes are required for the skin temperature to fall and the free moisture to disappear. Even ambulatory heart patients endure such changes with ease. The use of the word *shock* to describe any part of these adjustments is unwarranted.

Upon leaving an air-cooled space and re-entering a hot atmosphere, sudden warming occurs. Experiments at the ASHAE Research Laboratory[24] and elsewhere[25] indicated no demonstrable harm to a healthy individual. Adaptation occurred as soon as normal perspiration was established. Mild exercise shortened the adaptation time.

Most healthy, sedentary, and slightly active men and women, normally clothed and in uniform environments, with air velocities of the order of 25 fpm, are thermally comfortable the year around when the dry-bulb air temperature is in the range 73 to 77 F and the relative humidity is in the range of 25 to 60 percent.[27]

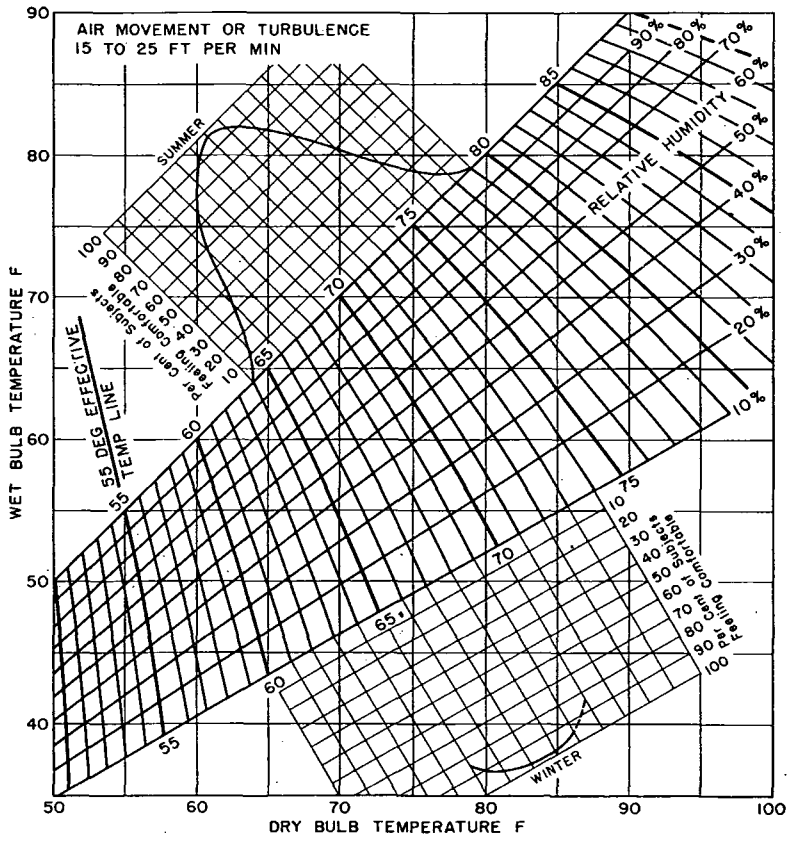
Satisfactory comfort conditions for persons at work[28] vary, depending upon the rate of work and the amount of clothing worn. In general, the greater the degree of activity, the lower the effective temperature necessary for comfort. Clothing has been evaluated for its overall insulation effects by a physical unit, the *clo* which equals 0.116 C deg per (kilogram calorie) (square meter) (hour).[29] Yaglou[30] criticizes the concept of overall insulation, and points out that different parts of the body require different amounts of insulation. The literature on effects of clothing is difficult to coordinate at the present time, as much of it is still in military service reports which are yet to be published and amplified.

For prematurely born infants, the optimum temperature varies from 100 to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 percent.[31] No data are yet available on the op-



A. Clothing: *Customary indoor clothing.* B. Activity: *Sedentary or light muscular work.* C. Heating Methods: *Convection type, i.e., warm air, direct steam or hot water radiators, plenum systems.*

Fig. 9 .... Effective Temperature Chart Showing Normal Scale of Effective Temperature, Applicable to Inhabitants of the United States Under Following Conditions:



[a] Note—Both summer and winter comfort lines apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices, and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours. The summer comfort line shown pertains to Pittsburgh and to other cities in the northern portion of the United States and Southern Canada, and at elevations not in excess of 1000 ft above sea level. An increase of one deg ET should be made approximately per 5 deg reduction in north latitude.

[b] Dotted portion of winter comfort line was extrapolated beyond test data.

Fig. 10 .... ASHAE Comfort Chart for Still Air[a,b]

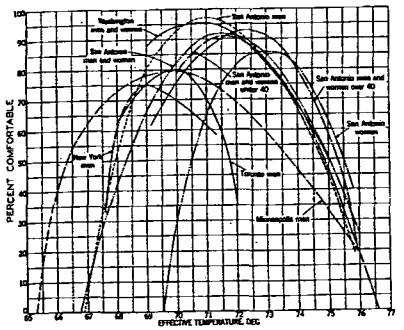


Fig. 11 .... Relation Between Effective Temperature and Percentage Observations Indicating Comfort

timum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 to 68 F with natural indoor humidities. For children (having high metabolism) at school, in winter clothes, 70 F has been considered correct, with 55 F recommended for gymnasiums.

REFERENCES

[1] Code of minimum requirements for comfort air conditioning (ASHVE Transactions, Vol. 44, 1938, p. 27).

[2] C. P. Yaglou, E. C. Riley, and D. J. Coggins: ASHVE Research Report No. 1031—Ventilation requirements (ASHVE Transactions, Vol. 42, 1936, p. 133).

[3] F. C. Houghten and J. L. Blackshaw: ASHVE Research Report No. 959—Indices of air change and air distribution (ASHVE Transactions, Vol. 39, 1933, p. 261).

[4] C. S. Leopold: Tobacco smoke control—A preliminary study (ASHVE Transactions, Vol. 51, 1945, p. 255).

[5] C. P. Yaglou, L. C. Benjamin, and S. P. Choate: ASHVE Research Report No. 921—Changes in ionic content in occupied rooms ventilated by natural and mechanical methods (ASHVE Transactions, Vol. 38, 1932, p. 191). C. P. Yaglou, A. D. Brandt, and L. C. Benjamin: ASHVE Research Report No. 965—Physiologic changes during exposure to ionised air (ASHVE Transactions, Vol. 39, 1933, p. 357). C. P. Yaglou and L. C. Benjamin: ASHVE Research Report No. 985—Diurnal and seasonal variations in the small ion content of outdoor and indoor air (ASHVE Transactions, Vol. 40, 1934, p. 271). L. B. Loeb: The nature of ions in air and their possible physiological effects (ASHVE Transactions, Vol. 41, 1935, p. 101). L. P. Herrington: The influence of ionised air upon normal subjects (*Journal Clinical Investigation*, 14, January 1935). L. P. Herrington and K. L. Smith: The effect of high concentration of light negative atmospheric ions on the growth and activity of the albino rat (*Journal Industrial Hygiene*, 17, November 1935). C.-E. A. Winslow and L. P. Herrington: Subjective reactions of human beings to certain outdoor atmospheric conditions (ASHVE Transactions, Vol. 42, 1936, p. 119).

[6] Editorial (*The British Medical Journal*, June 25, 1932, p. 1182).

[7] E. F. DuBois: The mechanism of heat loss and temperature regulation, (Lane Medical Lectures, Stanford University Publications, *Medical Science*, Vol. 3, No. 4, 1937, p. 348; also *Transactions of the Association of American Physicians*, Vol. 51, 1936, p. 252)

[8] E. J. Adolph and G. W. Molnar: Exchanges of heat and tolerance to cold in men exposed to outdoor weather (*American Journal of Physiology*, Vol. 146, 1946, p. 507).

[9] F. K. Hick, R. W. Keeton, N. Glickman, and H. C. Wall: ASHVE Research Report No. 1108—Cardiac output, peripheral blood flow and blood volume changes in normal individuals subjected to varying environmental temperatures (ASHVE Transactions, Vol. 45, 1939, p. 123).

[10] C.-E. A. Winslow, T. Bedford, E. F. DuBois, R. W. Keeton, A. Missenard, R. R. Sayers, and C. Tasker: ASHVE Research Report No. 1107—Recent advances in physiological knowledge and their bearing on ventilation practice (ASHVE Transactions, Vol. 45, 1939, p. 111).

[11] F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant: ASHVE Research Report No. 830—Heat and moisture losses from the human body and their relation to air conditioning problems (ASHVE Transactions, Vol. 35, 1929, p. 245).

[12] G. E. Burch and T. Winsor: Rate of insensible perspiration (diffusion of water) locally through living and through dead human skin (*Archives of Internal Medicine*, Vol. 74, 1944, p. 437).

[13] N. A. Nelson, W. B. Shelley, S. M. Horvath, L. W. Eichna, and F. F. Hatch: The influence of clothing, work and air movement on the thermal exchanges of acclimatized men in various hot environments (*Journal of Clinical Investigations*, Vol. 27, 1948, p. 209).

[14] W. J. McConnell and F. C. Houghten: ASHVE Research Report No. 654—Some physiological reactions to high temperatures and humidities (ASHVE Transactions, Vol. 29, 1923, p. 129). W. J. McConnell, F. C. Houghten, and F. M. Phillips: ASHVE Research Report No. 672—Further study of physiological reactions (ASHVE Transactions, Vol. 29, 1923, p. 353). W. J. McConnell, F. C. Houghten, and C. P. Yaglou: ASHVE Research Report No. 690—Air motion, high temperatures and various humidities—reactions on human beings (ASHVE Transactions, Vol. 30, 1924, p. 167). W. J. McConnell and C. P. Yaglou: ASHVE Research Report No. 718—Work tests conducted in atmospheres of high temperatures and various humidities in still and moving air (ASHVE Transactions, Vol. 31, 1925, p. 101). W. J. McConnell, C. P. Yaglou, and W. B. Fulton: ASHVE Research Report No. 719—Basal metabolism before and after exposure to high temperatures and various humidities (ASHVE Transactions, Vol. 31, 1925, p. 123). F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant: ASHVE Research Report No. 908—Heat and moisture losses from men at work and application to air conditioning problems (ASHVE Transactions, Vol. 37, 1931, p. 541). W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten, and M. B. Ferderber: ASHVE Research Report No. 1106—Air conditioning in industry—Physiological reactions of individual workers to high effective temperatures (ASHVE Transactions, Vol. 45, 1939, p. 59). F. C. Houghten, A. A. Rosenberg, and M. B. Ferderber: ASHVE Research Report No. 1153—Seasonal variations in reactions to hot atmospheres (ASHVE Transactions, Vol. 46, 1940, p. 185).

[15] R. W. Keeton, F. K. Hick, Nathaniel Glickman, and M. M. Montgomery: ASHVE Research Report No. 1151—The peripheral type of circulatory failure in experimental heat exhaustion (ASHVE Transactions, Vol. 46, 1940, p. 157).

[16] M. W. Heilman and E. S. Montgomery: Heat disease: clinical and laboratory studies (*Journal of Industrial Disease and Toxicology*, 18: 651, 1936).

[17] L. W. Eichna, W. B. Bean, W. F. Ashe, and N. Nelson: Performance in relation to environmental temperature (*Bulletin of Johns Hopkins Hospital*, Vol. 76, 1945, p. 25).

[18] C. M. Humphreys, Oscar Imalis, and Carl Gutberlet: Physiological response of subjects exposed to high effective temperatures and elevated mean radiant temperatures (ASHVE Transactions, Vol. 52, 1946, p. 153).

[19] D. B. Dill: *Life, Heat and Altitude* (Harvard University Press, Cambridge, 1938).

[20] L. W. Eichna, W. F. Ashe, W. B. Bean, and W. B. Shelley: The upper limits of environmental heat and humidity tolerated

by acclimatized men working in hot environments (*The Journal of Industrial Hygiene and Toxicology*, Vol. 27, March 1945, p. 59).

[22] F. C. Houghten, M. B. Ferderber, and Carl Gutberlet: ASHVE Research Report No. 1158—Local cooling of workers in hot industry (ASHVE Transactions, Vol. 47, 1941, p. 403).

[23] F. C. Houghten and C. P. Yaglou: ASHVE Research Report No. 673—Determination of the comfort zone (ASHVE Transactions, Vol. 29, 1923, p. 361). F. C. Houghten and C. P. Yaglou: ASHVE Research Report No. 691—Cooling effect on human beings produced by various air velocities (ASHVE Transactions, Vol. 30, 1924, p. 193). C. P. Yaglou and W. E. Miller: ASHVE Research Report No. 717—Effective temperature with clothing (ASHVE Transactions, Vol. 31, 1925, p. 89). F. C. Houghten, W. W. Teague, and W. E. Miller: ASHVE Research Report No. 755—Effective temperature for persons lightly clothed and working in still air (ASHVE Transactions, Vol. 32, 1926, p. 315). C. P. Yaglou, W. H. Carrier, Dr. E. V. Hill, F. C. Houghten, and J. H. Walker: How to use the effective temperature index and comfort charts (ASHVE Transactions, Vol. 38, 1932, p. 410).

[24] Thomas Chester, N. D. Adams, C. R. Bellamy, G. D. Fife, E. P. Heckel, Dr. W. J. McConnell, F. C. McIntosh, A. B. Newton, B. F. Raber, and C. Tasker: ASHVE Research Report No. 1196—Comfort with summer air conditioning (ASHVE Transactions, Vol. 48, 1942, p. 107).

[25] C. S. Leopold: Conditions for comfort (ASHVE Transactions, Vol. 53, 1947, p. 295).

[26] A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley, and M. C. W. Tomlinson: ASHVE Research Report No. 1102—Shock experiences of 275 workers after entering and leaving cooled and air conditioned offices (ASHVE Transactions, Vol. 44, 1938, p. 571). N. Glickman, T. Inouye, S. E. Telser, R. W. Keeton, F. K. Hick, and M. K. Fahnestock: Physiologic adjustments of human beings to sudden change in environment (ASHVE Transactions, Vol. 53, 1947, p. 327). N. Glickman, Tohru Inouye, R. W. Keeton, M.D., I. W. Callen, M.D., F. K. Hick, M.D. and M. K. Fahnestock: ASHVE Research Report No. 1350—Physiological adjustments of normal subjects and cardiac patients to sudden changes in environment (ASHVE Transactions, Vol. 55, 1949, p. 27). F. K. Hick, M.D., Tohru Inouye, R. W. Keeton, M.D., N. Glickman, and M. K. Fahnestock: ASHVE Research Report No. 1447—Physiological adjustments of clothed human beings to sudden changes in environments—first hot moist and later comfortable conditions (ASHVE Transactions, Vol. 58, 1952, p. 189). Tohru Inouye, F. K. Hick, M.D., R. W. Keeton, M.D., J. Losch, and N. Glickman: ASHVE Research Report No. 1463—A comparison of physiological adjustments of clothed women and men to sudden changes in environment (ASHVE Transactions, Vol. 59, 1953, p. 35). Tohru Inouye, F. K. Hick, M.D., R. W. Keeton, M.D., and Lionel Bernstein, M.D.: ASHVE Research Report No. 1508—Physiological responses to sudden changes in atmospheric environment—Studies of normal subjects, obese, hyperthyroid and hypothyroid patients (ASHVE Transactions, Vol. 60, 1954, p. 315).

[27] F. C. Houghten, F. E. Giesecke, C. Tasker, and Carl Gutberlet: ASHVE Research Report No. 1055—Cooling requirements for summer comfort air conditioning (ASHVE Transactions, Vol. 43, 1937, p. 145).

[28] A. P. Kratz, S. Konzo, M. K. Fahnestock, and E. L. Broderick: ASHVE Research Report No. 1012—Study of summer cooling in the research residence for the summer of 1934 (ASHVE Transactions, Vol. 41, 1935, p. 207). C. P. Yaglou: ASHVE Research Report No. 1319—A method of improving the effective temperature index (ASHVE Transactions, Vol. 53, 1947, p. 307). Tohru Inouye, F. K. Hick, M.D., S. E. Telser, M.D., and R. W. Keeton, M.D.: ASHVE Research Report No. 1480—Effect of relative humidity on heat loss of men exposed to environments of 80, 76, and 72 F (ASHVE Transactions, Vol. 59, 1953, p. 329). M. K. Fahnestock and J. E. Werden: Environment, comfort, health and people (*Refrigerating Engineering*, February 1956, p. 43).

[29] F. C. Houghten, W. W. Teague, and W. E. Miller: ASHVE Research Report No. 755—Effective temperature for persons lightly clothed and working in still air (ASHVE Transactions, Vol. 32, 1926, p. 315).

[30] A. P. Gagge, A. C. Burton, and H. C. Gazett: A practical system of units for the description of the heat exchange of man with his environment (*Science*, Vol. 94, 1941, p. 428).

[31] C. P. Yaglou: Thermal insulation of clothing (ASHVE Transactions, Vol. 54, 1948, p. 291).

[32] C. P. Yaglou, Philip Drinker, and K. D. Blackfan: Application of air conditioning to premature nurseries in hospitals (ASHVE Transactions, Vol. 36, 1930, p. 383).

CHAPTER 7

AIR CONTAMINANTS

Classification of Air Contaminants; Sizes of Airborne Particles; Air Pollution by Smoke, Ash, and Cinders; Smoke Abatement and Air Pollution Control; Odor Nuisance; Industrial Air Contaminants, Flammable Gases and Vapors; Combustible Dusts; Atmospheric Pollen; Airborne Bacteria; Radioactive Air Contaminants

THE normal constituents of the earth's atmosphere are oxygen, nitrogen, carbon dioxide, water vapor, argon, small or negligible amounts of other inert gases, hydrogen, variable traces of ozone, and small quantities of microscopic and submicroscopic solid matter, sometimes called *permanent atmospheric impurities*. From the viewpoint of the air-conditioning engineer, all other airborne substances may be termed *contaminants*. This term is applied preferably, however, to undesirable or chance impurities, since the occasion may arise for adding to the air controlled amounts of solid or gaseous diluents for the prevention of explosions; germicidal vapors or mists (aerosols) for bacteria control; masking substances for odor control; or a substitute for one of the normal gases, as, for example, when helium is used to replace nitrogen in atmospheres for compressed air workers or divers.

Since control of the chemical quality of air is one of the functions of complete air conditioning, some knowledge of the composition, concentration, and properties of air contaminants under various circumstances is essential to the air conditioning engineer.

Air contaminants arise from the normal processes of wear, erosion, windstorm, sea-spray evaporation, thermal disintegration, earthquake, volcanic eruption, combustion, manufacturing, transportation, agriculture, and the biochemical or biological processes of life. They are classified at various times as organic and inorganic, visible or invisible, microscopic or macroscopic, particulate or gaseous, toxic or harmless, beneficial or destructive. The following classification is based chiefly upon the origin or method of formation of air contaminants.

CLASSIFICATION OF AIR CONTAMINANTS

Air contaminants may be classified in the following groups: (1) dust, fumes, and smokes which are solid particulate contaminants, (2) mists and fogs which are liquid particulate contaminants, and (3) vapors and gases which are nonparticulate contaminants.

Dust, Fumes, and Smokes

Dusts are solid particles projected into the air by natural forces, such as wind, volcanic eruption, or earthquake, and by mechanical processes, such as crushing, grinding, milling, drilling, demolition, shovelling, conveying, screening, bagging, and sweeping. Some of these forces *produce* dust from larger masses, while others simply *disperse* materials that are already pulverized. Generally, particles are not called *dust* unless they are smaller than about 100 microns. Dusts may be of *mineral* type, such as rock, ore, metal, sand; *vegetable*, such as grain, flour, wood, cotton, pollen; or *animal*, such as wool, hair, silk, feathers, leather.

Fumes are solid particles commonly formed by the condensation of vapors from normally solid materials such as molten metals. Metallic fumes generally occur as the oxides in air because of the highly reactive nature of finely divided matter. Fumes may also be formed by sublimation, distillation, calcination, or chemical reaction, whenever such processes create airborne particles predominantly below the 1 micron size. Fumes permitted to *age* tend to flocculate into clumps or aggregates of larger size, thereby facilitating removal from air.

Smokes are the extremely small solid particles produced by incomplete combustion of organic substances such as tobacco, wood, coal, oil, tar, and other carbonaceous materials. The term *smoke* is commonly applied to the mixture of solid, liquid, and gaseous products of combustion, although the technical literature prefers to distinguish between such components as soot or carbon particles, fly-ash, cinders, tarry matter, unburned gases, and gaseous combustion products. The finest particulate constituents are much less than 1 micron in size, often in the range of 0.1 to 0.3 micron.

Mists and Fogs

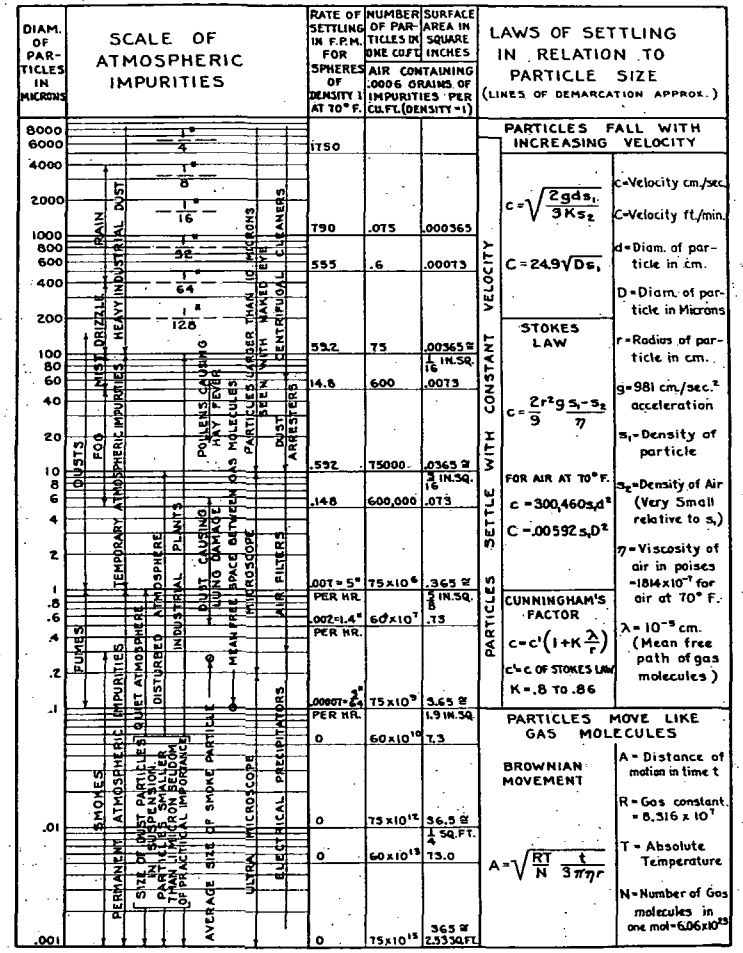
Mists are very small airborne droplets of materials that are ordinarily liquid at normal temperatures and pressures. They may be formed by atomizing, spraying, splashing, mixing, violent chemical reaction, electrolytic evolution of gas from a liquid, or escape of a dissolved gas upon release of pressure. Very small droplets expelled or atomized into the air by sneezing constitute mists containing micro-organisms that become air contaminants.

Fogs are limited by some classifications to airborne droplets formed by condensation from the vapor state. This arbitrary distinction between mist and fog is of minor importance, as both terms are used to indicate the particulate state of airborne liquids (occasionally termed *aerosols*). *Fog nozzles* are so named because of their ability to produce extra fine droplets as compared to the mist from ordinary spray devices. The highly volatile nature of some liquids quickly reduces their airborne droplets from the mist to the fog range, and eventually to the vapor phase until the air becomes saturated with that liquid. Many droplets in fogs or clouds are microscopic and submicroscopic in size, and may be conceived as the transition state between the larger mists and the vapors.

Vapors and Gases

Vapors and Gases are nonparticulate air contaminants.

Vapors are the gaseous phase of substances that are either liquid or solid in their commonly known state, examples being gasoline, kerosine, benzene, carbon tetrachloride, mercury, iodine, camphor. Vapors may be changed to the solid or



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Fig. 1 .... Sizes and Characteristics of Airborne Particulate Matter

liquid form by increasing the pressure, decreasing the temperature, or applying both processes simultaneously. They are removed from the air by condensation with less difficulty than are the gases.

Gases are normally formless fluids which tend to occupy a space or enclosure completely and uniformly at ordinary temperatures and pressures. The following substances qualify as gases: oxygen, nitrogen, carbon dioxide, carbon monoxide, hydrogen, ammonia, sulfur dioxide. Gases, likewise, may be solidified or liquefied by the proper control of temperature and pressure.

The preceding classification is not suitable for the airborne living organisms, which range in size from the submicroscopic *viruses* to the largest pollen grains, not considering the smallest insect life. Bacteria range from about 0.2 to 5 microns in size, fungus spores from 1 to 10 microns, and pollen from 5 to 150 microns.

SIZES OF AIRBORNE PARTICLES

Fig. 1 is a graphic tabulation of the properties of airborne solids and liquids arranged according to size on the micron scale. There are 25,400 microns in 1 inch.

Particles larger than 10 microns are unlikely to remain suspended in air currents of moderate strength, but settle out by gravity at speeds dependent upon the shape, size and specific gravity of the particle, wind velocity, orientation of the collecting surface, and topography. These larger particles are of major interest to the engineer in the solution of nuisance problems, but it is usually the smaller particles, or those below 10 microns, that remain in the air long enough to be of hygienic as well as economic significance.

Industrial dust particles are predominantly of the order of 1 micron in size. Tremendous numbers are also present in the submicroscopic range below 0.5 micron, but those below 0.1 micron are not believed at present to be of practical importance, possibly due to their exceedingly small mass in comparison with the balance of airborne matter. In fact, particles this small may become the *permanent* atmospheric impurities that have little, if any, opportunity of settling because of the continual motion imparted to them by air currents and the molecular activity of gases (Brownian Movement).

The survey[1] of atmospheric pollution in 14 American cities conducted from 1931 to 1933 indicated the average size of outdoor dust particles to be 0.5 micron, as collected by the Owens jet dust counter and measured under the microscope. Inability of the *light field* microscope to reveal particles in the 0.1 micron vicinity may have influenced the determination of average particle size.

The lower limit of particle size visible to the naked eye cannot be stated definitely. It depends not only upon the individual eye, but also upon the shape and color of the particle, intensity and quality of the light, and nature of the background or opportunity for contrast. Under ideal conditions a particle of 10-micron size might be recognized, while under less favorable conditions it may be impossible to distinguish a particle smaller than 50 microns. The lower limit of visibility *probably ranges from 10 to 50 microns.*

Dusts, powders, and granular materials are frequently classified by reference to the size of screens used for separation. Particles above 40 microns are said to be the *screen sizes* and those below, the *subscreen* or microscopic sizes. Approximate or theoretical sizes of particles corresponding to the mesh scale of the U. S. Standard Sieve Series are given in Table 1.

Table 1 .... Relation of Screen Mesh to Particle Size

U. S. Standard sieve mesh	400	325	200	140	100	60	35	18
Nominal sieve opening in microns	37	44	74	105	149	250	500	1000

Microscopic examination of screened dust indicates that the average diameter of a sample of irregular particles may be substantially larger than the openings of the screen through which it has passed, if the particle shapes deviate considerably from the spherical form.[2] The smallest dimension of many such particles will correspond to the maximum permissible distance between the wires of commercial screens made to ASTM Standard Specifications. Screening does not give sharp separation into size groups, and accordingly, such a classification is statistical rather than absolute.

AIR POLLUTION BY SMOKE, ASH, AND CINDERS

Total airborne solids settling in urban areas are usually reported as *soot fall* in tons per (square mile) (month). Such data published for the cities in this country range from 20 to 200 tons per (square mile) (month). To the air-conditioning engineer this information may indicate the effectiveness of smoke abatement or fuel combustion control methods in his locality, but it does not provide a suitable index of the suspended dust that air cleaners in a ventilating system are expected to capture.[3, 4, 5] Gravimetric or weight data of the type given in Table 2 are preferable. In some cases airborne particle counts may be necessary, as for pollen, bacteria, spores, and insoluble dusts causing illness or lung disease.

Table 2 .... Dust Concentration Ranges

Location	Grains per 1000 Cu Ft[a]	Milligrams per Cubic Meter
Rural and suburban districts	0.02–0.2	0.05–0.5
Metropolitan districts	0.04–0.4	0.1–1.0
Industrial districts	0.1–2.0	0.2–5.0
Ordinary factories or workrooms	0.2–4.0	0.5–10
Excessively dusty factories or mines	4–400	10–1000
Minimum explosive concentrations	4000–200,000	10,000–500,000

[a] 1 grain per 1000 cu ft = 2.3 milligrams per cubic meter.
1 oz per cubic foot = 1 gram per liter = 1000 grams per cubic meter.

Dust concentrations by weight cannot be converted readily to concentrations by particle count because of the variability of particle size, shape, and specific gravity, and the inherent characteristics of dust counting and weighing procedures. One milligram of dust per cubic meter of air may represent dust counts from 1 million to 100 million particles per cubic foot of air (light field microscope technic) according to the size distribution of the airborne dust sample. Information of this type for a specified application is best obtained by simultaneous sampling for both counting and weighing and noting carefully at the time all factors that might affect the reproducibility of the count-weight ratio.

Smoke Abatement and Air Pollution Control

Successful abatement of atmospheric pollution caused by smoke requires the combined efforts of the combustion engi-

neer, industrial executive, public health officer, city planning commission, and the community at large. Electrification of industry and railroads, increases in the use of domestic oil and gas furnaces, and segregation of industrial districts are gradually providing effective aid in the solution of this problem. In the large cities where nuisance from smoke, fly-ash, and cinders is more serious, limited areas obtain some relief by the use of district heating. (See Chapters 33, 34, and 35 for further discussion on fuel-burning technic.)

Many present ordinances limit the number of minutes in any one hour that smoke of a specified density (determined by comparison with a Ringelmann Chart which is described in Chapter 44) may be discharged.

There is now considerable interest and activity in the control of air pollution factors in addition to smoke. Difficulty in the establishment of acceptable criteria for certain corrosive and irritant gases, such as fluorides and the oxides of sulfur and nitrogen discharged with the gases of combustion, and the frequently complicated technical and economic problems encountered in control, have delayed the drafting and enforcement of legislative measures. Recent reports of an increased incidence of diseases, such as pneumonia and lung cancer, in areas high in certain air contaminants, require further critical investigation before acceptance. The values finally adopted will undoubtedly be lower than the MAC (Maximum Allowable Concentration) limits for use in industry, because the exposure is continuous compared with the 8-hour day, 5- or 6-day week upon which MAC values are based, and because the exposed population contains individuals with greater variation in age and health status.

In foggy weather, or during an *inversion* of atmospheric conditions, accumulation of gaseous contaminants may cause irritation of eyes, nose, and respiratory passages, and possibly cause even more serious physiological effects. The Meuse Valley fog disaster (Belgium 1930) and the Donora *smog* (Pennsylvania 1948) are classic examples in the history of gaseous air pollution. In both instances it is believed that irritant gases, principally from industrial plants, accumulating during periods of exceptionally prolonged meteorological *inversion* and fog, contributed to the illness of many persons, and to the death of some who were especially susceptible.

Absorption of Solar Radiation

Absorption of solar ultra-violet light by smoke and soot is recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore[6] by actinic methods demonstrated that ultra-violet light intensity in the country was 50 percent greater than in the city. In New York City[7] a loss as great as 50 percent in visible light was found by photoelectric measurements.

ODOR NUISANCE

A problem companionate with smoke abatement is the control of odor nuisance in the neighborhood of industrial plants discharging noxious or offensive air contaminants. Community planning and zoning will avoid much of the difficulty in the future, but meanwhile many industrial cities must resort to corrective measures by requiring installation of air cleaning devices, alteration of manufacturing processes, or termination of the offensive operation in residential or commercial districts.

Control of outdoor odor nuisance is especially troublesome because of the extremely minute quantities of contaminant that are capable of offending through a wide area. New industrial chemicals with strange or unfamiliar odors tend to receive more attention from the neighborhood than the customary odors generated by well-known processes and raw materials. Methods of odor control currently in use include charcoal adsorption, scrubbing towers and air washers, chlorination, condensation, masking, passage of the odorous air through combustion chambers, dispersion through a tall stack, and best of all, substitution of less offensive materials whenever possible.[8, 9, 10, 11]

Control of air quality within buildings ventilated for human occupancy is discussed in Chapter 6. Tobacco smoke odors, cooking odors, and body odors are air contaminants of the nuisance type which now command a decisive position in the standards of air quality for indoor comfort. However, the engineer will find, at times, that odors originating *outside* buildings in industrial or business districts may determine the kind and capacity of equipment he must provide for a high quality air-supply installation.

INDUSTRIAL AIR CONTAMINANTS

Many industrial processes are sources of contaminants. Their control is an important function of the ventilating or air-conditioning engineer, because the atmosphere within buildings is the medium whereby such finely divided matter is dispersed and transported from the source to remote locations where it may cause property damage, nuisance, fire, explosion, disease, and even death.

Tables 3, 4, and 5 give maximum allowable concentration values for many industrial air contaminants. Some of these values have been proposed by ASA Sectional Committee Z37 on Allowable Concentrations of Toxic Dusts and Gases, and others by the Committee on Threshold Limits of the American Conference of Governmental Industrial Hygienists. The values have been adopted by the American Conference of Governmental Industrial Hygienists and by many of the official industrial hygiene agencies.

It must be emphasized that these limits in the great majority of instances are only *suggested maximum working levels* since they are estimates, based in many cases upon incomplete environmental and medical studies. Where there is agreement between the *ASA Standard* Z37 and ACGIH values, reliability of the ASA Standard is increased. The values are not fixed, but are subject to revision, upward or downward, with the development of new information. There is by no means complete agreement regarding these values among responsible industrial hygienists.

In applying these guide limits, the following factors must be considered:

1. The duration of exposure is 8 hr a day for 5 or 6 days a week.

2. The measurements are indicative of the concentration in the breathing zone of the exposed person.

3. The MAC is usually accepted as the average exposure value when the upper limits do not greatly exceed the MAC value. For example, it cannot be assumed that if 100 ppm is considered *safe* for 8-hour exposure, that 800 ppm for one hour will be permissible.

4. Two substances with similar MAC values may be quite different as to physiological effects at other concentrations. In one instance the selection of a limit may be predicated upon a discomfort factor, with a wide margin of safety before systemic effects would be encountered, while in another case the value represents an appreciable fraction of the concentra-

Table 3 .... Maximum Allowable Concentration of Gases and Vapors

Substance	ASA Standards MAC ppm[b] by volume	Threshold Limit Values ACGIH[a] 1958 ppm[b]	Threshold Limit Values ACGIH[a] 1958 g/cu m or oz/1000 cu ft
Acetaldehyde		200	0.360
Acetic acid		10	0.025
Acetic anhydride		5	0.020
Acetone		1000	2.400
Acrolein		0.5	0.0012
Acrylonitrile		20	0.045
Allyl alcohol		5	0.012
(x) Allyl chloride		5	0.015
Allyl propyl disulfide		2	0.012
Ammonia		100	0.070
Amyl acetate		200	1.050
Amyl alcohol (iso)		100	0.360
Aniline		5	0.019
Arsine		0.05	0.0002
Benzene (benzol)	100	25	0.080
Benzyl chloride		1	0.005
Bromine		1	0.007
Butadiene (1,3-butadiene)		1000	2.200
Butanone (methyl ethyl ketone)		250	0.740
Butyl acetate (n-butyl acetate)		200	0.950
Butyl alcohol (n-butanol)		100	0.300
Butylamine		5	0.015
* Butyl cellosolve (2-butoxyethanol)		50	0.240
Carbon dioxide		5000	9.000
Carbon disulfide	20	20	0.060
Carbon monoxide	100	100	0.110
Carbon tetrachloride		25	0.160
Cellosolve (2-ethoxyethanol)		200	0.740
Cellosolve acetate (2-ethoxyethyl acetate)		100	0.540
Chlorine		1	0.003
Chlorine trifluoride		0.1	0.0004
Chlorobenzene (monochlorobenzene)		75	0.350
Chloroform (trichloromethane)		100	0.490
1-Chloro-1-nitropropane		20	0.100
(x) Chloropicrin		1	0.007
Chloroprene (2-chloro-1,3-butadiene)		25	0.090
Cresol (all isomers)		5	0.022
Cyclohexane		400	1.400
Cyclohexanol		100	0.410
Cyclohexanone		100	0.400
Cyclohexene		400	1.350
Cyclopropane		400	0.690
(x) Decaborane		0.05	0.0003
Diacetone alcohol (4-hydroxy-4-methyl-2-pentanone)		50	0.240
Diborane		0.1	0.0001
o-Dichlorobenzene		50	0.300
Dichlorodifluoromethane		1000	4.95
1,1-Dichloroethane		100	0.400
1,2-Dichloroethane (ethylene dichloride)		100	0.400
1,2-Dichloroethylene		200	0.790
Dichloroethyl ether		15	0.090
Dichloromonofluoromethane		1000	4.200
1,1-Dichloro-1-nitroethane		10	0.060
Dichlorotetrafluoroethane		1000	7.00
* Diethylamine		25	0.075
Difluorodibromomethane		100	0.860
Diisobutyl ketone		50	0.290
Dimethylaniline (N-dimethylaniline)		5	0.025
Dimethylsulfate		1	0.005
Dioxane (diethylene dioxide)		100	0.360
Ethyl acetate		400	1.400
(x) Ethyl acrylate		25	0.100
Ethyl alcohol (ethanol)		1000	1.900
Ethylamine		25	0.045
Ethyl benzene		200	0.870
Ethyl bromide		200	0.890
Ethyl chloride		1000	2.600
Ethyl ether		400	1.200
Ethyl formate		100	0.300
Ethyl silicate		100	0.850
Ethylene chlorhydrin		5	0.016
Ethylenediamine		10	0.030
Ethylenedibromide (1,2-dibromoethane)		25	0.190
Ethylene imine		5	0.009
Ethylene oxide		50	0.090
Fluorine		0.1	0.0002
Fluorotrichloromethane		1000	5.600
Formaldehyde	10	5	0.006
(x) Furfural		5	0.020
Gasoline		500	2.000
Heptane (n-heptane)		500	2.000
Hexane (n-hexane)		500	1.80
Hexanone (methyl butyl ketone)		100	0.410
Hexone (methyl isobutyl ketone)		100	0.410
Hydrazine		1	0.0013
Hydrogen bromide		5	0.017
Hydrogen chloride		5	0.007
Hydrogen cyanide		10	0.011
Hydrogen fluoride		3	0.002
Hydrogen peroxide, 90%		1	0.0014
Hydrogen selenide		0.05	0.0002
Hydrogen sulfide	20	20	0.030
* Iodine		0.1	0.001
Isophorone		25	0.140
Isopropylamine		5	0.012
* Mesityl oxide		25	0.100
Methyl acetate		200	0.610
Methyl acetylene		1000	1.650
(x) Methyl acrylate		10	0.035
Methyl alcohol (methanol)	200	200	0.260
Methyl bromide		20	0.080
Methyl cellosolve (2-methoxyethanol)		25	0.080
Methyl cellosolve acetate (ethylene glycol monomethyl ether acetate)		25	0.120
Methyl chloride	100	100	0.210
Methylal (dimethoxymethane)		1000	3.100

[a] ACGIH = American Conference of Governmental Industrial Hygienists. Reprinted by permission from *AMA Archives of Industrial Health*, September 1957, Vol. 16, pp. 251-255.

[b] Parts of vapor or gas per million parts of air by volume.

(x) These values appeared in ACGIH tentative list for 1956.

* These items include changes made by ACGIH Committee on Threshold Limits in 1958.

Table 3 . . . . Maximum Allowable Concentration of Gases and Vapors (Concluded)

Substance	ASA Standards MAC ppm[b] by volume	Threshold Limit Values ACGIH[a] 1958 ppm[b]	Threshold Limit Values ACGIH[a] 1958 g/cu m or oz/1000 cu ft
Methyl chloroform (1-1-1 trichloroethane)		500	2.700
Methylcyclohexane		500	2.000
Methylcyclohexanol		100	0.470
Methylcyclohexanone		100	0.460
Methyl formate		100	0.250
Methyl isobutyl carbinol (methylamyl alcohol)		25	0.100
Methylene chloride (dichloromethane)		500	1.750
Naphtha (coal tar)		200	0.800
Naphtha (petroleum)		500	2.000
Nickel carbonyl		*0.001*	*0.000007*
*(x) Nitric acid		10	0.025
p-Nitroaniline		1	0.006
Nitrobenzene		1	0.005
Nitroethane		100	0.310
Nitrogen dioxide		5	0.009
Nitroglycerine		0.5	0.005
Nitromethane		100	0.250
2-Nitropropane		50	0.180
Nitrotoluene		5	0.030
Octane		500	2.350
Ozone		0.1	0.0002
Pentane		1000	2.950
Pentanone (methyl propyl ketone)		200	0.700
Perchlorethylene (tetrachloroethylene)		200	1.350
Phenol		5	0.019
Phenylhydrazine		5	0.022
Phosgene (carbonyl chloride)		1	0.004
Phosphine		0.05	0.00007
Phosphorus trichloride		0.5	0.003
Propyl acetate		200	0.840
Propyl alcohol (isopropyl alcohol)		400	0.980
Propyl ether (isopropyl ether)		500	2.100
Propylene dichloride (1,2-dichloropropane)		75	0.350
Propylene imine		25	0.060
Pyridine		10	0.030
Quinone		0.1	0.0004
Stibine		0.1	0.0005
Stoddard solvent		500	2.90
Styrene monomer (phenylethylene)	400	100	0.420
Sulfur dioxide		5	0.013
Sulfur hexafluoride		1000	6.000
Sulfur monochloride		1	0.006
Sulfur pentafluoride		0.025	0.00025
p-Tertiarybutyltoluene		10	0.060
1,1,2,2-Tetrachloroethane		5	0.035
(x) Tetrahydrofuran		200	0.590
Tetranitromethane		*1*	*0.008*
Toluene (toluol)	200	200	0.750
o-Toluidine		*5*	0.022
Trichloroethylene	200	200	1.050
Trifluoromonobromomethane		1000	6.100
Turpentine		100	0.560
Vinyl chloride (chloroethylene)		500	1.30
Xylene (xylol)		200	0.870

tion which may be associated with severe and irreversible injury.

5. These values are *upper* limits; it is desirable to operate well below the levels if the engineering and economic factors permit. The prudent engineer will incorporate a reasonable margin of safety in his estimates of ventilation capacity.

In Table 3, Column 1 lists the ASA Sectional Committee Z37 MAC values; Column 2, those of the ACGIH Committee on Threshold Limits. Column 3 gives the value in grams per cubic meter or ounces per 1000 cu ft for the corresponding ACGIH limits.

In Table 4, Columns 2 and 3 are respectively the MAC values from the ASA and ACGIH Committees.

In Table 5, the MAC values for industrial dusts as given were obtained from data of the ACGIH Committee on Threshold Limits.

ACGIH limits for either gamma or Roentgen radiations are 0.3 roentgens per week.

Information on the properties and effects, with respect to health, of specific industrial air contaminants is available in publications listed at the end of this chapter.

FLAMMABLE GASES AND VAPORS

Adequate ventilation is a primary requirement for minimizing the hazard of fire or explosion due to gases and vapors. The need for good ventilation is not removed by the use of other precautions, such as the elimination of known ignition sources, segregation of hazardous operations, adoption of safe building construction, and installation of automatic alarms. Some safety engineers regard overventilation of an operation employing flammable liquids as a legitimate operating charge for the privilege or necessity of using a dangerous process. However, it is not possible to apply a reasonable safety factor to the ventilation estimate without consideration of the concentrations of gases or vapors that approach the danger point. Safety engineers prefer to limit the concentration to ¼ or ⅕ of the lower explosive limit, and this fact should be given full weight in determining the capacity and design of ventilating equipment. Rarely should consideration be given to *operation above the upper explosive limit* in the open areas of buildings or rooms—even though unoccupied—because the danger of temporary drop of gas concentration to a point within the explosive range is too great.

Ability of a flammable liquid to form explosive mixtures is determined largely by its vapor pressure, volatility, or rate of evaporation. *Flash point* is a convenient method of expressing this property in terms of the temperature scale. It may be defined as the temperature to which a combustible liquid must be heated to produce a flash when a small flame is passed across the surface of the liquid. The higher the flash point, the more safely can the liquid be handled. Liquids with flash points under 70 F should be regarded as highly flammable.

Upper and lower limits of flammability of gases and vapors, and the flash points of the corresponding liquids are given in Table 6.

Methods for estimating the flammable limits of mixtures of gases or vapors must be applied with caution; the reader is referred to other publications for this information.[13, 18]

Design of equipment for the control of combustible anesthetics is outlined in Chapter 8. Construction of equipment for handling air containing flammable substances, or operating in atmospheres so contaminated, is discussed in Chapter 52.

It is customary to report the concentrations of flammable

Table 4 . . . . Limits for Toxic Dusts, Fumes, and Mists

Substance	ASA Standards MAC, g/cu m[a]	Threshold Limit Values, ACGIH 1958 g/cu m[a]
Aldrin (1,2,3,4,10,10-hexachloro-1,4,4a,5,8,8a-hexahydro-1,4,5,8-dimethanonaphthalene)		0.00025
Ammate (ammonium sulfamate)		0.015
Antimony		0.0005
(x) ANTU (alpha naphthyl thiourea)		0.0003
Arsenic		0.0005
Barium (soluble compounds)		0.0005
Cadmium oxide fume		0.0001
(x) Calcium arsenate		0.0001
Chlordane (1,2,4,5,6,7,8,8-octachloro-3a,4,7,7a-tetrahydro-4,7-methanoindane)		0.002
(x) *Chlorinated camphene (60%)*		*0.0005*
Chlorinated diphenyl oxide		0.0005
Chlorodiphenyl (42% chlorine)		0.001
(x) Chlorodiphenyl (54% chlorine)		0.0005
Chromic acid and chromates (as CrO_3)	0.0001	0.0001
Crag herbicide (sodium 2-[2,4-dichlorophenoxy] ethanol hydrogen sulfate)		0.015
Cyanide (as CN)		0.005
2,4-D (2,4-dichlorophenoxyacetic acid)		0.010
(x) DDT (2,2-bis[p-chlorophenyl]-1,1,1,trichloroethane)		0.001
Dieldrin (1,2,3,4,10,10-hexachloro-6,7-epoxy-1,4,4a,5,6,7,8,8a-octahydro-1,4,5,8-dimethanonaphthalene)		0.00025
(x) Dinitrobenzene		0.001
Dinitrotoluene		0.0015
Dinitro-o-cresol		0.0002
EPN (O-ethyl O-p-nitrophenyl thionobenzenephosphonate)		0.0005
(x) Ferbam (ferric dimethyl dithiocarbamate)		0.015
Ferrovanadium dust		0.001
Fluoride		0.0025
Hydroquinone		0.002
Iron oxide fume		0.015
Lead	0.00015	0.0002
(x) Lead arsenate		0.00015
Lindane (hexachlorocyclohexane, gamma isomer)		0.0005
Magnesium oxide fume		0.015
Malathion (O,O-dimethyl dithiophosphate of diethyl mercaptosuccinate)		0.015
Manganese	0.006	0.006
Mercury	*0.0001*	*0.0001*
Mercury (organic compounds)		0.00001
Methoxychlor (2,2-di-p-methoxyphenyl-1,1,1-trichloroethane)		0.015
Molybdenum (soluble compounds)		0.005
(insoluble compounds)		0.015
(x) Nicotine		0.0005
Parathion (O,O-diethyl O-p-nitrophenyl thiophosphate)		0.0001
Pentachloronaphthalene		0.0005
Pentachlorophenol		0.0005
Phosphorus (yellow)		0.0001
Phosphorus pentachloride		0.001
Phosphorus pentasulfide		0.001
Picric acid		0.0001
Pyrethum		0.002
(x) Rotenone		0.005

Table 4 . . . . Limits for Toxic Dusts, Fumes, and Mists (Concluded)

Substance	ASA Standards MAC, g/cu m[a]	Threshold Limit Values, ACGIH 1958 g/cu m[a]
Selenium compounds (as Se)		0.0001
(x) Sodium fluoroacetate (1080)		0.0001
Sodium hydroxide		0.002
(x) Strychnine		0.00015
Sulfuric acid		0.001
TEDP (tetraethyl dithionopyrophosphate)		0.0002
TEPP (tetraethyl pyrophosphate)		0.00005
Tellurium		0.0001
Tetryl (2,4,6-trinitrophenylmethylnitramine)		0.0015
(x) Thiram (tetramethyl thiuram disulfide)		0.005
(x) Thallium (soluble compounds)		0.0001
Titanium dioxide		*0.015*
Trichloronaphthalene		0.005
Trinitrotoluene		0.0015
Uranium (soluble compounds)		0.00005
(insoluble compounds)		0.00025
Vanadium (V_2O_5 dust)		0.0005
(V_2O_5 fume)		0.0001
(x) Warfarin (3-[α acetonylbenzyl]-4-hydroxycoumarin)		0.0005
Zinc oxide fumes		0.015
Zirconium compounds (as Zr)		0.005

[a] Grams of dust, fume, or mist per cubic meter of air.

(x) These values appeared on the ACGIH tentative list, for 1956.

Radioactivity: For permissible concentrations of radioisotopes in air, see *Maximum Permissible Amounts of Radioisotopes in the Human Body and Maximum Permissible Concentrations in Air and Water*, Handbook 52, U. S. Department of Commerce, National Bureau of Standards, March, 1953. In addition, see *Permissible Dose from External Sources of Ionizing Radiation*, Handbook 59, U. S. Department of Commerce, National Bureau of Standards, September 24, 1954.

Table 4 values reprinted by permission from *AMA Archives of Industrial Health*, September 1957, Vol. 16, pp. 261-265. No changes were made by ACGIH Committee on Threshold Limits at 1958 meeting.

Table 5 . . . . Limits for Mineral Dusts

Substance	Threshold Limit Values ACGIH 1958 mppcf[a]
Aluminum oxide	50
Asbestos	5
Cement, Portland	50
Dust (nuisance; no free silica)	50
Mica (below 5% free silica)	20
Silica—high (above 50% free SiO_2)	5
Silica—medium (5 to 50% free SiO_2)	20
Silica—low (below 5% free SiO_2)	50
Silicon carbide	50
Soapstone (below 5% free SiO_2)	20
Talc	20

[a] mppcf—million particles per cubic foot of air, standard light field count.

Table 5 values reprinted by permission from *AMA Archives of Industrial Health*, September 1957, Vol. 16, pp. 261-265. No changes were made by ACGIH Committee on Threshold Limits in 1958.

Table 6 .... Approximate Limits of Flammability of Single Gases and Vapors In Air at Ordinary Temperatures and Pressures[a]

Gas or Vapor	Lower Limit Percent by Volume	Upper Limit Percent by Volume	Closed Cup[b] Flash Point Fahrenheit	NBFU Classification[c]
Acetaldehyde	4.0	57	−17	I
Acetone	2.5	12.8	0	I
Acetylene	2.5	80	....	
Allyl alcohol	2.5	....	70	II
Ammonia	15.5	26.6	....	
Amyl alcohol	1.2	....	100	III
Amyl chloride	1.6	....	....	
Amylene	1.6	7.7	....	
Benzene (benzol)	1.3	6.8	12	I
Benzyl chloride	1.1	....	140	II
Butane	1.8	8.4	....	
Butyl acetate	1.4	15.0	84	III
Butyl alcohol	1.7	....	....	
Butylene	2.0	9.7	....	
Carbon disulfide	1.2	50	−22	I
Carbon monoxide	12.5	74.2	....	
Crotonaldehyde	2.1	15.5	55	II
Cyclohexane	1.3	8.4	1	I
Cyclopropane	2.4	10.5	....	
Decane	0.67	2.6	115	III
Dichloroethylene (1, 2)	9.7	12.8	57	II
Diethyl selenide	2.5	....	....	
Dioxane	2.0	22.2	54	II
Ethane	3.1	12.5	....	
Ether (diethyl)	1.8	36.5	−49	I
Ethyl acetate	2.2	11.5	28	II
Ethyl alcohol	3.3	19.0	54	II
Ethyl bromide	6.7	11.3	....	
Ethyl cellosolve	2.6	15.7	104	III
Ethyl chloride	4.0	14.8	−58	I
Ethylene	2.7	28.6	....	
Ethylene dichloride	6.2	15.9	56	II
Ethyl formate	2.7	16.5	−4	I
Ethyl nitrite	3.0	50	−31	I
Ethylene oxide	3.0	80	....	
Furfural (125 C)	2.1	....	140	III
Gasoline (variable)	1.4–1.5	7.4–7.6	−50	I
Heptane	1.0	6.0	25	II
Hexane	1.2	6.9	−15	I
Hydrogen cyanide	5.6	40.0	....	
Hydrogen	4.0	74.2	....	
Hydrogen sulfide	4.3	45.5	....	
Illuminating gas (coal gas)	5.3	33.0	....	
Isobutyl alcohol	1.7	....	82	III
Isopentane	1.3	....	....	
Isopropyl acetate	1.8	7.8	43	II
Isopropyl alcohol	2.0	....	53	II
Methane	5.0	15.0	....	
Methyl acetate	3.1	15.5	14	I
Methyl alcohol	6.7	36.5	52	II
Methyl bromide	13.5	14.5	....	
Methyl butyl ketone	1.2	8.0	....	
Methyl chloride	8.2	18.7	....	
Methyl cyclohexane	1.1	....	25	II
Methyl ethyl ether	2.0	10.1	−35	I
Methyl ethyl ketone	1.8	9.5	30	II
Methyl formate	5.0	22.7	−2	I
Methyl propyl ketone	1.5	8.2	....	
Natural gas (variable)	4.3	13.5	....	
Naphtha (benzene)	1.1	6.0	20–110	III

Table 6 .... Approximate Limits of Flammability of Single Gases and Vapors In Air at Ordinary Temperatures and Pressures[a] (Concluded)

Gas or Vapor	Lower Limit Percent by Volume	Upper Limit Percent by Volume	Closed Cup[b] Flash Point Fahrenheit	NBFU Classification[c]
Naphthalene	0.9	....	176	III
Nonane	0.83	2.9	88	III
Octane	0.95	3.2	56	II
Paraldehyde	1.3	....	....	
Pentane	1.4	7.8	....	
Propane	2.1	10.1	....	
Propyl acetate	1.8	8.0	58	II
Propyl alcohol	2.1	13.5	59	II
Propylene	2.0	11.1	....	
Propylene dichloride	3.4	14.5	60	II
Propylene oxide	2.0	22.0	....	
Pyridine	1.8	12.4	74	III
Toluene (toluol)	1.3	7.0	40	II
Turpentine	0.8	....	95	III
Vinyl ether	1.7	27.0	....	
Vinyl chloride	4.0	21.7	....	
Water gas (variable)	6.0	70	....	
Xylene (xylol)	1.0	6.0	63	II

[a] Adapted from: *Fire and Explosion Hazards of Combustible Gases and Vapors*, by G. W. Jones; Chapter 13, *Industrial Hygiene and Toxicology*, edited by F. A. Patty (Interscience Publishers, 1948); *Properties of Flammable Liquids, Gases and Solids* (Associated) Factory Mutual Fire Ins. Cos., January (1940); and *National Fire Codes for Flammable Liquids, Gases, Chemicals and Explosives*—1945 (National Fire Protection Association).

[b] *Closed cup* refers to the equipment used in flash point determinations.

[c] From *Standards for Storage, Handling and Use of Flammable Liquids* (National Board of Fire Underwriters, No. 30, July 1954).

gases or vapors in percent by volume, or *volume percent*. Comparison with concentrations on the *part per million* scale used in chemical, medical, or industrial hygiene literature is readily made by the conversion: 1 percent = 10,000 ppm (parts of contaminant per million parts of air, by volume, or in other words, cubic feet of contaminant per million cubic feet of air). It will be noted in Table 6 that nearly all of the substances listed have lower explosive limits above 1.0 percent, while the *maximum allowable concentrations* for gases and vapors in Table 3 are below 1000 ppm or 0.1 percent in most cases. Therefore, control of toxic or injurious vapors to levels below their maximum allowable concentrations for health usually requires much more effective ventilation than for the prevention of a fire hazard.

COMBUSTIBLE DUSTS

A dust explosion is essentially a sudden pressure rise caused by the very rapid burning of airborne dust. The primary explosion often originates from a small amount of dust in suspension exposed to a source of ignition and the pressure and vibration it creates may be sufficient to dislodge large accumulations of dust on horizontal ledges or surfaces of the building and equipment, thereby creating a secondary explosion of great force. Thus the air-conditioning engineer is involved for two reasons: (1) to obtain a movement of dust-laden air into exhaust hoods or openings, and through ventilating or pneumatic conveying ducts, in a manner that will prevent accumulation of highly flammable dust at points

where it could ignite inside the equipment; and (2) to so design *process ventilation* as to prevent the escape of dust which might settle on horizontal surfaces and become a potential source of disaster at some distance from the dusty operation. (See Chapter 52.)

Intensity of a dust explosion depends upon: chemical and thermal properties of the dust; particle size and shape; concentration in air; proportion of inert dust in the air; moisture content and composition of the air; size and temperature of the ignition source; and degree of dispersion of the dust cloud. Investigations on the explosibility of dusts require determination of the maximum pressure developed during explosion of a known air concentration, as well as determination of the rate of pressure rise. Investigators frequently experience difficulty in obtaining dust suspensions of uniform dispersion, and this should be kept in mind when comparing results from several sources.[14]

Minimum explosive concentrations of airborne dusts already tested range from 0.01 to 0.5 oz per cubic foot, or 10 to 500 grams per cubic meter of air. Maximum pressures generated have been reported as high as 500 psig, although they are more likely to be of the order of 50 psig. Investigations on the flammable characteristics of dusts are currently made at 0.1 and 0.5 oz per cubic foot.[15–21]

ATMOSPHERIC POLLEN

Properties of pollen grains discharged by weeds, grasses, and trees and responsible for hay fever, are of special interest to designers of air cleaning equipment (see Allergic Disorders in Chapter 8, and Air Cleaning, Chapter 24). Whole grains and fragments transported by the air range chiefly between 10 and 50 microns in size, but some have been measured as small as 5 microns, and others over 100 microns in diameter. Ragweed pollen grains are fairly uniform in size within the range of 15 to 25 microns. Pollen grains can be removed from the air more readily than the particles of dust prevalent in outdoor air or produced by dusty processes, since the latter predominate in the range of 0.1 to 10 microns in size.

Most grains are quite hygroscopic and therefore vary in weight with the humidity. Illustrations and data on individual pollen grains are available in the botanical literature.[22, 23, 24] Geographical distribution of plants known to produce hay fever is also recorded.[25, 26]

The quantity of pollen grains in the air is generally estimated by exposing an adhesive-coated glass plate outdoors for 24 hr, and then counting calibrated areas under the microscope. Methods are available for determining the number of grains in a measured volume of air,[25, 27, 28] but their greater accuracy has not caused them to replace the more simple *gravity slide method* used for most pollen counts. Counting technics vary somewhat, but the daily pollen counts reported in local newspapers during the hay fever season usually represent the number of grains found on 1.8 sq cm of a 24-hr *gravity slide*.

Hay fever sufferers may notice the first symptoms when the pollen count is 10 to 25, and in some localities the maximum figures for the seasonal peak may approach 1000 for a 24-hr period, depending upon the sampling and reporting methods of the laboratory. Translation of gravity counts by special formulas to a volumetric basis, or the number of grains per cubic yard or per cubic foot of air, is unreliable because of the complexity of the modifying factors. When such information is important, it is best obtained directly by a volumetric instrument. The number of pollen grains per cubic yard of air evidently varies from 2 to 20 times the number found on 1 sq cm of a 24-hr gravity slide, depending on grain diameter, shape, specific gravity, wind velocity, humidity, and physical placement of the collecting plate.[25, 26, 27]

AIRBORNE BACTERIA

Study of the occurrence and significance of micro-organisms in the atmospheres of the indoor world is absorbing the energies of a substantial number of physicians, bacteriologists, aerobiologists, physicists, public health workers, engineers, and hospital personnel. Some data are available on the types and quantities of bacteria found in a variety of spaces, but it is not possible at present to use this information as a conclusive index of the potential health hazard of a given environment. The reported number of airborne organisms may vary from 1 to 1000 per cubic foot of air, influenced somewhat by the method of testing.[29] Many are attached to the dust particles present in the air.

Where it seems advisable or desirable to control the bacterial content of rooms, public conveyances, or buildings, highly effective methods are available (see Chapter 8), and their extended use may do much to assist the workers in this field in accumulating the necessary mass of evidence that will decide the practical value of air sterilization for the control of communicable disease. It is now well established that ultra-violet radiation is feasible for the protection or preservation of pharmaceuticals, cosmetics, and food products.

RADIOACTIVE AIR CONTAMINANTS[32]

Radioactive air contaminants are physically similar to ordinary industrial and chemical plant contaminants. They differ from ordinary contaminants, however, in that they may be highly toxic when ingested and may present a direct radiation hazard when accumulated in appreciable quantities. Another factor which must be considered in the filtration of these materials is the possibility of their being concentrated in plant and animal life in the area of contamination. Concentrations resulting from absorptions by plants and subsequent feeding of animals in the magnitude of several thousand times, is not unusual. Tolerances for radioactive contaminants have been established and are under continuing study and review. Guidance in regard to tolerance levels for discharge of contaminated air in different types of localities can best be had from the classified literature which is available through proper clearance procedures.

REFERENCES

[1] J. E. Ives et al: *Atmospheric Pollution of American Cities for the Years 1931 to 1933* (*U.S. Public Health Service Bulletin* No. 224, March 1936).

[2] J. M. DallaValle: *Micrometrics, The Technology of Fine Particles* (Pitman Publishing Corporation, New York, 1943).

[3] A. C. Stern: Atmospheric pollution due to smoke (*Heating and Ventilating*, May 1945). A. C. Stern: Atmospheric pollution due to dust and cinders (*Heating and Ventilating*, July 1945).

[4] J. Siegel and B. Feiner: Sootfall studies for New York City (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, September 1945, p. 495).

[5] A. C. Stern: The use of fuel consumption and equipment data in the abatement of atmospheric pollution (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, August 1945, p. 447).

[6] J. H. Shrader, M. H. Coblentz, and F. A. Korff: Effects of atmospheric pollution upon incidence of solar ultra-violet light (*American Journal of Public Health*, Vol. 19, 1929, p. 7).

[7] J. E. Ives: *Studies in Illumination III: A Study of the Loss*

of Light Due to Smoke on Manhattan Island (*U. S. Public Health Service Bulletin* No. 197, June 1930).

[8] F. M. Stead: Study and control of industrial atmospheric pollution nuisances (*American Journal of Public Health*, Vol. 35, May 1945, p. 491).

[9] J. M. DallaValle and H. C. Dudley: *Evaluation of Odor Nuisance in the Manufacture of Kraft Paper* (*U. S. Public Health Service Reprint* No. 2022, *Public Health Reports*, Vol. 54, January 13, 1939, p. 35).

[10] *Disposal of Refinery Wastes, Section II: Waste Gases, Vapors, Sludges and Dusts* (American Petroleum Institute, 1938).

[11] David Ronald: *Offensive Trades* (William Hodge and Co., London, 1935).

[12] H. F. Coward and G. W. Jones: *Limits of Inflammability of Gases and Vapors* (*U. S. Bureau of Mines Bulletin* No. 279, 1939).

[13] G. W. Jones: Inflammation limits and their practical application in hazardous industrial operations (*Chemical Reviews*, Vol. 22, February 1938).

[14] H. R. Brown: Private communication (U. S. Bureau of Mines).

[15] P. W. Edwards and L. R. Leinbach: *Explosibility of Agricultural and Other Dusts as Indicated by Maximum Pressure and Rates of Pressure Rise* (*U. S. Department of Agriculture Technical Bulletin* No. 490, October 1935).

[16] H. R. Brown: *Dust Explosion Hazards in Plants Producing or Handling Aluminum, Magnesium, or Zinc Powder* (*U. S. Bureau of Mines Information Circular* No. 7148, March 1941).

[17] I. Hartman, J. Nagy, and H. R. Brown: *Inflammability and Explosibility of Metal Powders* (*U. S. Bureau of Mines Report of Investigation* No. 3722, October 1943).

[18] I. Hartman and J. Nagy: *Inflammability and Explosibility of Powders Used in the Plastics Industry* (*U. S. Bureau of Mines Report of Investigation* No. 3751, May 1944).

[19] H. R. Brown: *Industrial Dust Explosions* (*U. S. Bureau of Mines Information Circular* No. 7309, January 1945).

[20] *Proposed Code for the Prevention of Dust Explosions in the Plastics Industry* (National Fire Protection Association, May 1945).

[21] *National Fire Codes for the Prevention of Dust Explosions* (National Fire Protection Association, 1944). Contains codes for aluminum, magnesium, coal, pulverized fuel, flour, spice, starch, sugar, cocoa, sulfur, and wood.

[22] G. Erdtman: *An Introduction to Pollen Analysis* (Chronica Botanica Co., Waltham, Massachusetts, 1943).

[23] R. P. Wodehouse: *Pollen Grains* (McGraw-Hill Book Co., New York, 1935).

[24] R. P. Wodehouse: Atmospheric pollen (*Aerobiology*, Publication No. 17, 1942, p. 8).

[25] R. P. Wodehouse: *Hayfever Plants* (Chronica Botanica Co., Waltham, Massachusetts, 1945).

[26] *Hay Fever: A Geographical and Botanical Survey* (E. R. Squibb and Sons, New York, 1937).

[27] Techniques for appraising air-borne populations of microorganisms, pollen and insects (*Phytopathology*, Vol. 31, March 1941, p. 201).

[28] B. J. Cody, W. F. Kinney, and N. A. Kerstein: *Apparatus for Determining the Pollen Concentration of the Atmosphere* (Research Department, Detroit Edison Company, Detroit).

[29] O. C. Durham: The volumetric incidence of atmospheric allergens (*Journal of Allergy*, Vol. 14, September 1943, p. 455).

[30] O. C. Durham: The volumetric incidence of atmospheric allergens, II: Simultaneous measurements by volumetric and gravity slide methods (*Journal of Allergy*, Vol. 15, May 1944, p. 226).

[31] O. C. Durham: Air-borne fungus spores as allergens (*Aerobiology*, Publication No. 17, 1942, p. 32).

[32] H. G. DuBuy and A. Hollaender: Sampling devices (*American Journal of Medical Science*, Vol. 209, February 1945, p. 172).

[33] W. W. McIntosh: *Ventilation Problems in Safe Handling of Radioactive Materials* (Paper presented at ASME Spring Meeting, March 1952).

BIBLIOGRAPHY

Monthly, annual, and special abstracts and bulletins (Industrial Hygiene Foundation, Inc.).

Aerobiology (Publication No. 17, 1942).

W. N. Witheridge: Air sanitation and industrial ventilation (Detroit, Michigan, 1945).

American Industrial Hygiene Association Quarterly.

M. B. Jacobs: *Analytical Chemistry of Industrial Poisons, Hazards and Solvents* (Interscience Publishers, Inc., New York, 1941).

Bibliography of Industrial Hygiene, 1900–1943 (*U. S. Public Health Service Bulletin* No. 289, 1945).

W. E. Gibbs: *Clouds and Smokes* (P. Blakiston's Son & Co., Philadelphia, 1924).

Bloomfield and DallaValle: *Determination and Control of Industrial Dust* (*U. S. Public Health Service Bulletin* No. 217, 1935).

S. C. Blacktin: *Dust* (The Sherwood Press, Cleveland, 1934).

The Environment and Its Effect upon Man (Harvard School of Public Health, Boston, 1937).

Drinker and Hatch: *Industrial Dust* (McGraw-Hill Book Co., New York, 1936).

A. D. Brandt: *Industrial Health Engineering* (John Wiley and Sons, New York, 1947).

F. A. Patty (ed.): *Industrial Hygiene and Toxicology* (Interscience Publishers, Inc., New York, Vol. I, 1948 and Vol. II, 1949).

Journal of Industrial Hygiene and Toxicology. See cumulative abstract and subject indexes.

W. M. Gafafer et al: *Manual of Industrial Hygiene* (U. S. Public Health Service, W. B. Saunders Co., Philadelphia, 1943).

Henderson and Haggard: *Noxious Gases and the Principles of Respiration Influencing Their Action* (American Chemical Society Monograph Series No. 35, Reinhold, New York, 1943).

Occupation and Health (International Labor Office, Washington, D.C., 2 volumes).

Lehmann and Flury: *Toxicology and Hygiene of Industrial Solvents* (Translated by Eleanor King and H. F. Smyth, Jr., Williams & Wilkins, Baltimore, 1943).

CHAPTER 8

AIR CONDITIONING IN THE PREVENTION AND TREATMENT OF DISEASE

Sanitary Ventilation, Control of Airborne Infection, Value of Air Cooling Under Tropical Conditions, Treatment of Disease, Operating Rooms, Nurseries for Premature Infants, Fever Therapy, Cold Therapy, Allergic Disorders, Oxygen Therapy, General Hospital Air Conditioning

THE Second World War caused an increase of interest in the preventive aspects of air conditioning. It re-emphasized the importance of the control of airborne infection and demonstrated the value of air cooling under tropical conditions for the prevention of heat rash, for promoting proper rest and sleep, and in the convalescence of patients.

The problem of air conditioning or air purification in shelters, hospitals and other occupied buildings, to reduce the hazard from radioactive particles following an atomic explosion, merits the most careful consideration. Shelters should be without windows, underground, strong enough to resist blast and have sufficient cover to protect against initial radiation. They should be closed to outdoor air in the period immediately following a blast but may eventually have to be provided with ventilation from outdoors to limit the increase of temperature, moisture or odor, resulting from the occupancy, or in extreme cases to avoid serious oxygen deficiency and increase of carbon dioxide. When air must be drawn from outdoors and power is available, a mechanical ventilation system can be used. This will make it possible to pressurize the shelter, to utilize special filtration apparatus and to control the entry of outdoor air at will.

A surface or subsurface burst of an atomic bomb will result in a wide dispersion of airborne radioactive particles.[1] Windows of buildings within a three-mile radius from the point of detonation of a 20 KT bomb will be broken, and consequently, no appreciable protection will be afforded by the use of conventional air-conditioning systems under these conditions. Beyond the three-mile radius many structures may be found with windows undamaged and otherwise capable of providing some degree of protection from airborne radioactive particles by the utilization of an air-conditioning system. However, conventional air-conditioning systems should not be relied upon for protection in urban structures which may be utilized in the accomplishment of specific missions following an atomic detonation. There are special techniques and devices available which will provide this protection.

The smog incident in Donora, Pa., in 1948 focused the attention of health authorities and engineers on the problem of air contamination by the gaseous wastes discharged into the air above cities. The U. S. Public Health team investigating the incident[2] found that in this city several previous incidences of lesser severity had occurred during the preceding 30 years as indicated by an increase of mortality rates during certain periods. A high percentage of the population suffered to some extent during the 1948 smog. Older individuals particularly were seriously affected with cardio-respiratory symptoms.

The exact way in which the contaminants due to gaseous wastes affect the human being is unknown.[3] It is believed that the combination of contaminants rather than any single substance produces the toxic effects. The medical profession has much to learn about the effects of contaminants on human beings as a basis for determining when precise controls are needed.[3]

SANITARY VENTILATION

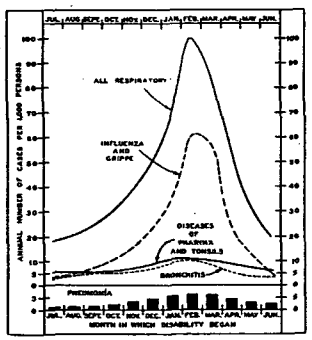
During the last 20 years great popular interest has been aroused in the spread of respiratory infection indoors, and in its control by ventilation or its sanitary equivalent, air disinfection. In Europe three important documents have appeared in English, Swedish, and French literature.[4, 5, 6] In the United States where the study began, the vast literature has been consolidated, and definitions, formulations, and factors of sanitary ventilation codified in a volume now being edited for publication.

The Council of Physical Medicine of the American Medical Association approved the radiant disinfection of air in 1943, and two sub-committees of the Committee on Research and Standards of the American Public Health Association have reported favorably upon the control of airborne infection by sanitary ventilation and on air sanitation respectively.[7, 8, 9, 10] The ASHAE Technical Advisory Committee on Air Sterilization from 1944 to 1949 reported progress in THE ASHAE JOURNAL, and recommended a set of definitions, formulations and factors for joint adoption by the American Public Health Association and the Society. The following section on Control of Airborne Infection gives some idea of the scope of the subject.

CONTROL OF AIRBORNE INFECTION

The majority of airborne diseases are spread indoors where people gather. Any program of air sanitation is influenced by a number of factors. In the winter months, the closing of doors, windows, and other means of access to the outdoor air to conserve warmth, as well as the crowding of persons indoors, provides conditions conducive to a high incidence of contagion. This seasonal phenomenon, illustrated in Fig. 1 which represents a study made by the U. S. Public Health Service, will concern the ventilating engineer insofar as air quality (determined by temperature, humidity, air replenishment, type of air movement, and freedom from contamination) is a major intrinsic factor. Apart from the

seasonal picture of airborne contagion, are such extrinsic factors as rate of turnover of personnel, and the marked susceptibility of the recruit as compared with permanent personnel[21] as shown (see Fig. 2) by studies of military personnel housed in barracks. These extraneous variables and the factor of contact infection (direct spray) tend to complicate any evaluation of the effectiveness of air sanitation for elimination of micro-organisms in droplet-nuclei and droplet-dust. Thus, control measures may eliminate consistently 90 percent of airborne organisms in laboratory tests, but may not effect a decrease in actual incidence of infection exceeding 30 percent. Thirty percent may be the maximal reduction in infection possible by air treatment methods. The distinction should be clearly drawn, therefore, between the effectiveness of a procedure in laboratory tests



Occurrence of diseases causing disability for 8 consecutive days or longer in a group of 100,000 wage earners (10 percent women) in different industries.

Fig. 1.... Study of Average Monthly Frequency (1921–1926 Inclusive) of Specified Respiratory Diseases

and its effectiveness and applicability in actually reducing the incidence of airborne disease. On the other hand, recent studies suggest that inhalation of dust-borne bacteria is more important than direct inhalation of infectious droplets or droplet nuclei in the spread of respiratory tract infections.[18]

The following sequence of events has been postulated as occurring in a large proportion of intra-ward infections: (*a*) ejection of relatively large protected infective particles from patients; (*b*) rapid venting or settling of these particles so that those remaining airborne are in low concentration; (*c*) survival of infective particles to permit the accumulation of high concentrations on surfaces; (*d*) repeated reintroduction of infective particles into the air under the stimulus of ward activities or by air currents of the order of 50 fpm over the floor; and (*e*) extension of infective areas by air turbulence throughout the ward or hospital. The most important link in this probable infection chain has been demonstrated to be the reintroduction of particles into the air.[23]

Intensive studies on air disinfection have indicated two distinct control measures: (*a*) suppression of dust and lint, and (*b*) disinfection of droplet-nuclei. A third measure, control of relative humidity, is important. It has been shown that the viability of certain organisms sprayed into the atmosphere from a liquid suspension is dependent on relative humidity. The mortality rate of the organisms is very high at a relative humidity of 50 percent,[24] and decreases at humidities above and below this figure. It has also been reported that the influenza virus loses much of its virulence when the relative humidity is 50 percent.[13]

Well controlled, large scale tests of the various methods of air sterilization conducted in barracks[25, 27] have confirmed the importance of dust control in minimizing the spread of airborne disease, a consideration which has guided the practices of ventilating engineers for a number of years. The importance of the dust factor has been emphasized by many engineers, and has been convincingly demonstrated by subsequent bacteriologic studies aboard ships.

Treatment of floors and bedclothes with oil emulsions has proved effective in reducing bacterial dispersion by as much as 90 percent in Army barracks and station hospitals.[17] The incidence of acute respiratory infections was from 10 to 30 percent lower in barracks with oiled floors and bedclothes than it was in control barracks which received no special treatment. More recent studies, however, have yielded inconsistent results.

An emulsifying mixture, *Fixanol C* containing cetyl pyridinium bromide, when incorporated in an oil-in-water emulsion imparted a bactericidal action. Blankets treated with this emulsion became bactericidal and retained this property for as long as three months. The possibility of hypersensitivity of an occasional individual to bromide drugs should be borne in mind when exposing large groups to treated garments or blankets.[13]

No simple method for disinfecting droplet-nuclei has yet been devised. Under favorable laboratory conditions, propylene glycol in concentrations of 0.07 to 0.14 milligrams per liter, and triethylene glycol in a concentration of 0.0045 milligrams per liter were highly germicidal for most airborne bacteria in clean air when the relative humidity was between 40 and 60 percent.[26, 29, 30] A humidity of 50 percent, without the use of glycol vapor, has been reported as destructive to some bacteria. However, maximum rates of bactericidal action of triethylene glycol vapor will be secured at humidities between 20 to 50 percent.[31]

In a recent report on the effect of triethylene glycol vapor in air disinfection, it is pointed out that the rate of ventilation, as determined by the number of air changes per hour, is important and that continuous vaporization is needed to maintain effective concentrations of glycol vapor.[31] In the absence of an apparatus to measure the concentration of vapor in the room, a slight fog is an indication of adequate concentration. Absence of such a fog indicates a non-bactericidal concentration. The report stated: "However, under experimental conditions we have observed no apparent decomposition of triethylene glycol when temperatures up to 290 F were maintained at the site of vaporization, provided the vaporizing unit was designed so that heat was applied to liquid glycol only at the point of vapor formation. On the other hand, decomposition was frequently observed when the vaporizing temperature was raised above 290 F, or when liquid glycol was kept above 120 F for long periods of time. The precise nature of the decomposition products is as yet unknown, but they may well be irritant or toxic and their formation should be avoided. For this reason, it is felt that a limit of 290 F should be applied to the temperature employed in vaporization, and furthermore, that liquid glycol should not be heated above 120 F for long periods of time.[27]

It was also recommended that vaporizers be adjustable to a wide range of output as measured by grams of liquid glycol vaporized per hour; for instance, from 0.5 g to at least 2.0 g per 1000 cu ft of volume treated, for small vaporizing units. Such high rates secure bactericidal concentration in less than an hour. Each room should be furnished with a single small vaporizer, and one vaporizer should not be expected to handle a volume greater than 4000 cu ft. Apparatus for the production of glycol vapor, and an independent duct system for carrying this vapor and diluting air for large rooms or spaces, and unit type vaporizers for small spaces, have been described.[32] There is also available a device called the glycostat for the automatic regulation of glycol vapor in the air. This instrument has been calibrated to measure the degree of saturation of the air with glycol vapor by direct reading of the variations in the intensity of

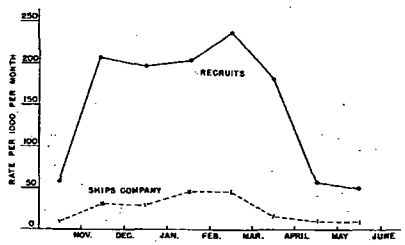


Fig. 2.... Monthly Incidence of Acute Respiratory Illness Among Naval Recruits and Ship's Company (Permanent Personnel)[21]

light reflected from the glycol condensing surface of the wheel of the instrument.[28]

Under practical conditions, particularly in the presence of dust in the air, glycol effectiveness is much reduced. The use of other chemical aerosols that have been tried is limited by their toxicity, odor, or destructiveness to fabrics and metals. A recently reported controlled experiment in the offices of the Metropolitan Life Insurance Company showed that under ordinary working conditions triethylene glycol vapor failed to reduce the number of airborne bacteria and the incidence rate of minor respiratory infections.[26]

Ultraviolet radiation of floors and upper air has been studied extensively at the Naval Training Center, Sampson, N. Y. In barracks housing naval recruits, hospital admissions for respiratory infections (mostly *catarrhal fever*) were 25 percent lower in a group of men exposed to ultraviolet radiation—2537 Angstrom Units, 1 to 7 ergs per (cm^2) (sec) at bed level—than they were in adjacent control barracks without ultraviolet radiation.[33]

A controlled study over a six-year period on the evaluation of ultraviolet radiation of sleeping quarters as a supplement to accepted methods of respiratory disease control has been reported.[34] The amount of radiation over the last two years of the period was believed to be about five times that recommended commercially. No significant effect on the incidence of disease could be detected in about 400 inmates during the six-year period. This conclusion may not be applicable to present ultraviolet equipment installed in accordance with manufacturers' recommendations. The data imply that air layers were not sufficiently mixed. However, more efficient mixing would have been obtained at the expense of increased circulation of dust and lint. Sources of ultraviolet radiation should be so situated as to protect the eyes of the occupants of the room from direct or reflected rays. A combination of ultraviolet radiation and dust control measures is believed to be more effective than either one of the two used alone, but the proof for this has yet to come.

There is no doubt that these methods will reduce the number of bacteria in the air of an enclosed space, but there is still considerable question as to whether they are practical measures for reducing the total number of respiratory infections among personnel, since exposure by contact is an important factor. Bourdillon and his group concluded that there is justification for attempting to reduce the load of airborne micro-organisms by the methods now available.[4] They have explored the properties of a number of compounds, and their conclusions are in agreement with those of American investigators.

The present status is admirably reviewed by the Committee on Sanitary Engineering of the National Research Council,[27] and by a subcommittee of the American Public Health Association.[3] Both committees believe that the problem of air disinfection is still in the experimental stage. Since knowledge of the effectiveness of glycol vapors has not kept pace with the development of vaporizing devices, there is real danger that commercial exploitation of the various devices may discredit the method and discourage careful research in this important field.[35] More experimentation is needed before a definite conclusion can be reached concerning its use in industry and public buildings.

VALUE OF AIR COOLING UNDER TROPICAL CONDITIONS

Although statistics are not at hand to indicate the deaths or retarded recoveries of patients due to lack of air cooling in ships operating in tropical waters, it is generally agreed among competent observers that high temperature and humidity are major factors in prolonging disability and increasing mortality of the sick and injured. Physiologic data obtained on healthy men, moreover, show the large loss of body fluids and the stress on the cardiovascular system in terms of increased pulse rate when these men are continuously subjected to high temperatures. Even at rest about 50 cc of fluid per hour are lost as sweat[36] through intact skin. In burn patients the difficulty, encountered in temperate climates, of maintaining fluid and electrolyte balance is tremendously augmented by the additional evaporative fluid loss in hot environments. Naval hospital ships having all wards, laboratories, and living spaces air cooled, represent a notable achievement in the control of environmental factors for the purpose of providing better treatment of patients, especially those suffering from extensive burns.

Patients who have such varied conditions as heart disease, thyrotoxicosis, shock from any cause, or severe hemorrhage, or those who have had an anesthetic, will invariably store heat when subjected to a hot humid environment. The gradient between the body surface temperature and the environmental temperature is such that loss by radiation is slight. The heat loss by evaporation in a warm humid environment is low whether the patient does or does not perspire. The

heat regulatory center may be temporarily deranged following an anesthetic, brain injury, or after an overdose of barbiturate. Loss of fluids and electrolytes is another influencing factor. In these circumstances there is great benefit from assisting the body in cooling, and this can best be done in a cool room of low relative humidity where conditions for heat loss are ideal. This measure is also valuable in controlling temperature of patients with various acute febrile diseases.[38]

Frequently from 50 to 75 percent of personnel aboard naval vessels operating in tropical waters are afflicted with heat rash to a degree that interferes with rest and sleep. In carefully controlled experiments[39] it was possible to produce a fulminating type of rash in all men living continuously at an effective temperature of 85 (90 F dry-bulb and 83 F wet-bulb). In the control group, 12 out of 24 hr were spent in a relatively cool atmosphere of 75 ET (80 F dry-bulb, and 70 F wet-bulb). These men either remained free from heat rash, or occasionally developed a mild form. Thus, intermittent cooling to a degree that prevented sweating in men at rest, eliminated a serious handicap to good performance of duty.

In both laboratory tests and aboard hospital ships a relatively cool living environment of 76 to 78 ET provided an atmosphere conducive to rest and sleep without excessive sweating. Berthing spaces tended to have extremely low odor levels. Motivation, initiative and alertness, in contrast to the usual irritability and lack of incentive incident to residence in tropical climate, were maintained.[40]

Little has been done, however, to obtain practical methods for application of air conditioning under heavy heat loads and on the enormous scale that would be needed to modify life in the tropics. Part time cooling of personnel to promote rest and sleep in tropical areas would provide a prophylactic measure of great potential importance.

TREATMENT OF DISEASE

In the past few years considerable progress has been made in using air conditioning as an adjunct in the treatment of various diseases. Among the important applications are those in operating rooms, nurseries for premature infants, maternity and delivery rooms, children's wards, clinics for arthritic patients, heat therapy, cold therapy, oxygen therapy, X-ray rooms, and the control of allergic disorders.

Normal individuals may be subjected to considerable strain in adjusting to hot humid conditions. Heat loss by radiation is reduced, as is loss by evaporation of sweat. Individuals with certain disease processes are at a still greater disadvantage since they may also have difficulty in the transport of heat from the interior to the surface of the body via the circulation.

Patients with thyrotoxicosis tolerate hot humid conditions or *heat waves* very poorly. Their metabolism is high, and therefore their heat production is excessive. They may be unable to eliminate heat from the body surface as rapidly as it is produced and transported to the skin. They develop hyperthermia or fever, and a tachycardia or rapid heart rate. The demand on the circulation for transport of heat from the interior of the body to the skin surface is increased. The increased body temperature leads to increased cell metabolism, and in turn to still greater heat production. This vicious cycle may threaten life if the cardio-vascular or transport mechanism breaks down. A cool, dry environment favors the loss of heat by radiation and evaporation from the skin, and may save the life of the patient.

Cardiac patients may be unable to maintain the circulation necessary to insure normal heat loss. Recently the importance of air conditioning hospital wards and rooms of cardiac patients, particularly those with congestive heart failure, has been stressed as a therapeutic measure.[41] It is more important in tropical or subtropical climates. Individuals with head injuries, those subjected to brain operations, and those with barbiturate poisoning may have hyperthermia, especially in a hot environment, due to a disturbance in the heat regulatory center of the brain. Obviously, one of the most important factors in recovery is an environment in which the patient can lose heat by radiation and evaporation, namely, a cool room with dehumidified air.

The patient in shock, or the patient who has had a severe hemorrhage, may have an inadequate volume of circulating blood and be unable to maintain an adequate skin circulation. This may result in heat storage or fever. Patients with extensive skin burns may be unable to lose heat adequately from the limited uninvolved skin surface, and thus develop a fever. They need adequate fluid replacement, saline solution, plasma or blood to expand the circulating blood volume and thereby improve peripheral circulation. A cool environment is valuable in aiding heat loss after adequate skin circulation is established.

A hot dry environment (89.6 F and 35 percent relative humidity) has been used over an extended period for the treatment of patients with rheumatoid arthritis, with reported improvement.[42]

OPERATING ROOMS

The widest application of air conditioning in hospitals is in operating rooms. Complete air conditioning of operating wards is important because winter humidification helps reduce the danger incident to the use of anesthetic gases; summer cooling with some dehumidification tends to eliminate excessive fatigue and to protect the patient and operating personnel; and finally, filtering aids the removal of allergens from the operating room air.

Reducing Explosion Hazard

Explosion hazards in operating rooms increase with the introduction of anesthetic gases and apparatus. Ether administered by the old drop method gives rise to an explosive mixture, but in practice this method is still regarded as comparatively safe. When ether is mixed with pure oxygen, or nitrous oxide in certain concentrations, the explosion hazard may be as great as with ethylene-oxygen, or cyclopropane-oxygen mixtures.

Of the anesthetic gases nitrous oxide alone does not explode but supports combustion. Ether, vinyl ether, ethylene, and cyclopropane are as potentially dangerous as gasoline or *illuminating gas in the home.*[43] *Chloroform does not* explode violently in contact with flame, but decomposes to liberate phosgene. All of the anesthetic gases and vapors, except ethylene, are heavier than air. Although the incidence of injury or death from explosion is negligible compared with other hazards in the operating room, the dramatic features surrounding an explosion justify continued investigation to eliminate the hazard.

During the course of ethylene anesthesia, the mixture, usually 80 percent ethylene and 20 percent oxygen, is so rich that the danger of explosion is slight in the immediate vicinity of the face mask, but leakage of ethylene into the air may result in lower concentrations, and thus introduce a serious hazard. The most dangerous period is at the end of the operation when the patient's lungs and the anesthesia apparatus are customarily *washed out* with oxygen with or without the addition of carbon dioxide. Even when this procedure is omitted, it is difficult in practice to avoid dilution of the anesthetic gas with air during the normal course of breathing following the administration. In either case the mixture would pass through the explosion range and extraordinary precaution is necessary for the safety of the patient and operating personnel.

In a study[44] of 230 anesthetic explosions and fires, 70 percent of the explosions and 60 percent of the deaths were caused by igniting agents other than static sparks. The National Fire Protection Association[45] made certain recommendations for safe practice based on available information in Pamphlet No. 56, *Recommended Safe Practice for Hospital Operating Rooms* (July 1952). These recommendations outline ways and means for eliminating or correcting hazardous conditions which experience and investigation have shown to contribute to the hazards in question. They are divided into three parts: Part I, General, deals with the nature of the hazards, Part II, Construction and Equipment, deals with physical standards for features incorporated into the construction and equipment of the surgical suite, and Part III, Administration, contains precautions to be observed by hospital personnel. The requirements and recommendations are interdependent and each will be ineffective unless coordinated with the other. To approach complete success in the prevention of anesthetic explosions, all persons—the surgical staff, the nursing staff, the maintenance staff and administrative personnel—must be educated and periodically reminded of the explosive nature of combustible anesthetic agents.

Experience has shown that neither high humidity nor intercoupling devices have eliminated the danger from static electric discharge. The removal of gas concentrations from the operating table area, by means of specially devised exhaust ventilation, should be thoroughly tested. Portable duct systems as installed aboard ship should be acceptable. Serious explosions can occur in a closed system, but proper precautions will reduce this hazard to a minimum.

A comprehensive study of the explosion problem and of the general causes and prevention of operating room hazards, by the University of Pittsburgh, the ASHAE Research Laboratory, and the U. S. Bureau of Mines has led to a fruitful attempt to eliminate the explosive range of cyclopropane, one of the best but most difficult gases to handle. The use of helium as a diluent in the total gaseous mixture controls the oxygen concentration by displacement and, because of its flame quenching properties, it is the ideal gas for this purpose. In addition, a gaseous mixture containing helium is more difficult to ignite by electric discharges, and this quality also increases the safety factor of anesthetic administration.

Operating Room Conditions

Little is known about optimum air conditions for maintaining normal body temperatures during anesthesia and the immediate post-operative period. An anesthetized patient displays dilation of blood vessels in the skin resulting in profuse sweating and (it has been believed) inability to regulate body temperature. From this it was concluded that all anesthetized patients suffered considerable heat loss, although there may be little more than 0.8 F deg variation in the rectal temperature during the course of the operation.[47] The severe physiological effects, such as excessive sweating and rapid pulse, of high operating room temperatures on attendants and patients during the hot months signify the need for proper cooling. Statements of surgeons who operate in both air-conditioned and non-air-conditioned rooms strongly indicate that the recuperative power of the patient is greater when operated upon in air-conditioned rooms.[47]

Although the comfortable air conditions for the operators are not identical with those for the patient, it is usually not difficult to compromise within a range of 55 to 60 percent relative humidity and 72 to 80 F temperature. It is generally reported that dry-bulb temperatures of 76 to 78 F with 55 percent relative humidity not only furnished comfort for the operating room workers, but apparently prevented exhaustion of patients as evidenced by rapid convalescence in the recovery ward. Additional heat may be furnished to patients locally or by suitable covering, according to body temperature in individual cases.

In the control of airborne infection in the operating room, the prevention of dispersal of infectious materials into the air, control of dust, and proper ventilation supersede attempts to remove or kill pathogenic organisms. The bacterial content of conditioned operating rooms is generally lower than that of non-conditioned rooms.

Bacterial counts aboard an air-conditioned submarine were found to be exceptionally low and not cumulative with time, although all of the air was recirculated for more than 12 hours[48] without replenishment. The removal of bacteria by the process of air cooling and condensation of moisture out of air, merits further study.[49]

The degree of air contamination can be reduced by proper ventilation if velocity of air over the floor does not exceed 50 fpm. Research is in progress on the use of filtered air flowing through a system of mechanical cleaners which protect the patient against infection from attendants, and from bacteria-containing air in the corridor or ward.[46]

Operations are frequently postponed on allergic patients during asthmatic manifestations through fear of complications. The removal of airborne allergens, therefore, is in some cases an important function of the air-conditioning system in preparing patients for operation.

The best practice in air conditioning hospital operating rooms is the use of all outdoor air with arrangements for preheat, reheat, and the control of humidity, coupled with a mechanical exhaust system that removes the air from both the high and low levels of the operating room. However, there is some evidence of lack of any cumulative effect on bacteria count due to recirculation of air through an air conditioning unit in operating rooms. This consideration and the fact that relief cooling frequently provides improved, if not ideal conditions, has led to the use of central systems employing 50 percent recirculated air. Also, perhaps as a recognition of practice, the NBFU Pamphlet No. 56 outlines the conditions under which room air conditioners may be used in operating rooms. These conditions are so written in the pamphlet that the recommendation of separate mechanical ventilation of such rooms is clearly inferred.

Stated in the reverse order, room air conditioners for relief cooling of mechanically ventilated rooms are now considered permissible when used with the necessary precautions. These precautions are not generally assured by the use of standard room air conditioners as produced by the manufacturers, and are frequently costly of accomplishment in the field. The use of larger systems employing 50 percent

recirculated air is a recognized practice. It is certain that such a system must have the necessary precautions (regarding the electrical and air moving equipment) against the hazard of gross spillage of the anesthetics. Systems using 100 percent outdoor air and adequate air removal means are regarded as following the best practice. The choice between the systems is usually made on the basis of first costs. As compared with the other daily usage costs of an operating room, the differences between the operating costs of the various air conditioning systems is not a significant item.

Good thermal construction of the operating rooms is a recommended practice. Much can be gained by thermal insulation of sterilizing equipment, and by exhaust ventilation of sterilizing rooms adjoining the operating rooms. The amount of glass surface should be kept to a minimum, particularly in walls exposed to the sun. Double windows are desirable and often necessary to prevent condensation on the glass in cold weather. The equipment capacity and control methods must be properly selected for the type of system employed, and for the loads encountered. The resulting air flow rates are usually in the range of 8 to 12 changes per hour. The method of air introduction should be selected to keep air movement in the operating area under 50 fpm. Where all outdoor air or a large percentage of outdoor air is used, the air introduction and exhaust arrangements should be designed to provide a thorough air change in all parts of the room. This may be accomplished by introducing the air at one side and exhausting it from the other side of a small room or by introducing the air at the center and exhausting from the sides of a large room. The supply and exhaust arrangements can be exchanged in the case of large rooms. An air-conditioned recovery ward in connection with the air-conditioned operating room, is of great value in stabilizing peripheral circulation, and in reducing excessive loss of fluids on hot humid days.

NURSERIES FOR PREMATURE INFANTS

One of the most important requirements in the care of premature infants is the stabilization of body temperature. This is necessary because the infant's heat regulatory system is not fully developed, with the resultant tendency for environmental temperature to influence body temperature. The younger the premature infant, the greater is the tendency. As the infant's metabolism is low, heat production is not adequate to maintain a normal body temperature in a cool environment. The resistance to infection is low, and the mortality rate is high. In general, the younger the age of the premature infant, the higher the mortality rate.

Nurseries constructed for metabolic research should be air-conditioned so that conditions are reproducible. Results of such studies may be invalid if environmental conditions are not identical, since fluid and electrolyte loss may vary greatly with change in environmental conditions.

Air-Conditioning Requirements

The optimum air conditions for growth and development of premature infants were determined by extensive research[1] at the Children's Hospital, Boston, Mass., using four valid criteria, namely, stability of body temperature, gain in weight, incidence of digestive syndromes, and mortality. Individual temperature requirements varied widely (from 72 to 100 F) according to the constitutional state of the infants and body weights. The optimum relative humidity was about 65 percent, and the air movement less than 20 fpm.

A single nursery conditioned to 77 F and 65 percent relative humidity was found to fulfill satisfactorily the requirements of the majority of premature infants. Additional heat for weak (or debilitated) infants may be furnished in the cribs or by means of electric incubators placed inside the conditioned nursery, with temperature adjusted according to individual requirements. In this way a multiplicity of chambers and of air-conditioning apparatus is obviated; the infants in the heated beds derive the benefit of breathing cool humid air, and the nurses and doctors need not expose themselves to extreme conditions.

Importance of Humidity: Although external heat is an important factor in the maintenance of normal body temperature, humidity appears to be of equal or greater importance. When the premature nurseries at the Children's Hospital were kept at relative humidity between 25 and 50 percent for two weeks or longer, the body temperature became unstable, gain in weight diminished, the incidence of gastro-intestinal disturbances increased, and the mortality rose. On the other hand, continuous exposure to air conditions with 55 to 65 percent relative humidity gave satisfactory results over a period of years. The initial physiologic loss of body weight (loss occurring within first four days of life) was found to vary inversely with the humidity. In the old nurseries with natural humidity it averaged 12.4 percent of the birth weight; in the conditioned nurseries it was 8.9 percent with 25 to 49 percent relative humidity, and 6.0 percent with 50 to 75 percent relative humidity. The number of days required to regain the birth weight was correspondingly maximum in the old nursery, and minimum in the conditioned nurseries under high humidity.

Maximum gains in body weight occurred in the conditioned nurseries under high humidity (55 to 65 percent) in infants weighing less than 5 lb. The gains were less under low humidity (25 to 50 percent) in the same nurseries, and in the old nurseries prior to use of air-conditioning apparatus.

The incidence and severity of digestive syndromes, with diarrhea, persistent vomiting, diminishing gain or loss of body weight, and other symptoms, were generally from two to three times as high under low as under high humidity.

Summarizing, the best chances for life in premature infants are created by maintaining a relative humidity of 65 percent in the nursery, and by providing a uniform environmental temperature just sufficiently high to keep the body temperature within normal limits. Medical and nursing care are, of course, factors of equal and sometimes of greater importance.

Air-Conditioning Equipment

Many of the installations now in use are of the central system type providing for filtration, for humidification and heating in cold weather, and for cooling and dehumidification in hot weather. A ventilation rate between 8 and 12 air changes per hour is desirable to remove odors and maintain uniformity of temperatures in extremes of weather. Recirculation should not be used in these wards owing to odors and the possibility of infection. There should be a frequent change in spray water.

Control of Airborne Infection

The protection of the premature and older infant against infection is of the utmost importance. It was found in one installation equipped with air conditioning, germicidal lights and mechanical barriers that air conditioning alone did not prevent respiratory cross-infections.[2] Bacterial ultraviolet barriers, air conditioning and mechanical barriers are efficient. However, infections are brought in by, and often spread by, ward personnel in spite of these measures.

FEVER THERAPY

Artificial production of high fever in man can be considered an imitation of nature's way of overcoming invading pathogenic organisms. The action may be direct and specific by destruction of the invading organism within the safe limit of human temperatures, or indirect in the case of heat resistant organisms, by general mobilization of the defensive mechanisms of the body.

Although the action may be direct and specific by destruction of the invading organisms within the safe human limits, fever therapy exerts much of its benefit through the improvement of the mechanism of bodily defense. A serious challenge to the theory on which fever therapy is based comes from the demonstration that high fever causes a reduction in the concentration of circulating antibodies in experimental animals.

Patients for fever therapy should be carefully selected. The most serious complications which may arise are heat stroke, heat exhaustion and circulatory collapse. The chief minor complications are heat cramps, fever blisters and mild dehydration.

The limits of induced systemic fever are usually between 104 and 107 F (rectal), and the duration from 3 to 8 hours at a time. The total period of fever treatment varies with the type of the organism involved.

The diseases which respond favorably to artificial fever therapy are gonorrhea and its complications (which include arthritis, pelvic infections in women, and involvement of the eye), syphilis and chorea.

The most striking results are seen in gonorrhea and syphilis, since the causative organisms can be destroyed at temperatures compatible with human life. However, the use of fever therapy has decreased since penicillin has been found so effective in the treatment of gonorrhea and syphilis. Mild fever, up to 101 F for one hour, has reventy been used in the treatment of rheumatoid arthritis. This degree of fever is not bactericidal, but is believed to stimulate the body defense mechanism.

Equipment for Production of Fever

Artificial fever can be induced by injections of various crystalloid or colloid substances, bacterial products of typhoid and malarial organisms, or by physical methods using hot baths, radiant heat cabinets, hot humidified air cabinets, or by short wave diathermy in combination with a cabinet.

The relative advantages of various methods have been evaluated clinically.[3] Among the devices for the production of fever by physical means, the one most widely used is the hot humid air or *air-conditioned* cabinet. This apparatus was developed at the Kettering Institute for Medical Research at Miami Valley Hospital in Dayton, Ohio.

In the earlier studies of the Society, temperatures were elevated more easily using saturated atmospheres. A fever therapy apparatus[4] using these same principles has proved efficient as a means of inducing and maintaining fever in a body, with small likelihood of burns because of the comparatively low dry-bulb temperatures.

When heat is necessary in treating legs or arms, such media as short- or long-wave diathermy, micro-waves, infrared, water baths, etc., have been used extensively. A recent development, a saturated atmosphere heating unit, similar to one previously described has proven satisfactory, because heat may be administered over longer periods which render deep heating possible without fear of burns or shocks. Local heating has been somewhat satisfactory in relieving the painful symptoms of peripheral vascular disease. This procedure, however, is not without danger. Elevation of tissue temperature increases cell metabolism and the need for oxygen. The inadequate blood supply and oxygen deficiency may lead to tissue death or gangrene. Application of heat to the trunk or abdomen, with consequent reflex dilatation of the vessels of the extremities, eliminates this danger of local heat application.

Short-wave diathermy within the cabinet during the induction phase has been used. When the desired body temperature has been reached by electrical induction, the atmosphere of the enclosure is kept at saturation to prevent heat loss, thus maintaining the patient's temperature at the desired point. The two underlying principles in the production of fever by the hot humid air cabinet are: (1) the transfer of heat by conduction from the circulating hot air to the body, and (2) prevention of heat loss. The latter is more important. In an atmosphere of high humidity, the heat loss by evaporation is markedly decreased.

COLD THERAPY

Cold as an anesthetic agent was advocated by Allen several years ago.[5] Freezing of the tissue must be avoided. For certain patients, in whom amputation of an extremity is indicated, the application of a tourniquet with cooling of the affected extremity down to near freezing (5 C or 40 F) is of value. The patient, following this procedure can be prepared for surgery without the handicap of absorption of septic products and severe pain. This procedure has proven especially valuable in the neglected diabetic patient with an infected gangrenous extremity. Time for treatment of coma and hydration of the patient is gained. However, if amputation of an extremity is not indicated, the application of a tourniquet and packing in ice are dangerous procedures, since loss of the limb usually results. An extremity with inadequate blood supply can be readily cooled without the use of a tourniquet, but such an extremity is also usually eventually lost. Cooling is said to reduce the metabolism of the tissue with suspension of the vital processes. It also reduces the blood flow to practically zero, and few extremities with inadequate blood supply remain viable or recover.

Packing in ice, or use of low temperatures, is contraindicated in the treatment of patients with frostbite, immersion foot or trench foot. The affected extremities should be exposed to the air in a cool room and not rubbed with snow or packed in ice. The lowering of temperature by packing the body in ice for treatment of cancer has not proven successful.

The methods used for refrigeration, depending upon available facilities, are as follows:[6]

(1) Cracked or shaved ice which is simple and has the advantage of not freezing tissues. However, it is cumbersome and sloppy to handle and is unsuited to prolonged treatments.

(2) Use of ice in a pail for immersion of local parts.

(3) Special boxes for holding ice with padded or curtained openings for the limb.

(4) Bare ice bags and cloth bags for iced wet dressings for prolonged treatments and convenience.

(5) A double-chambered cabinet using dry ice has been constructed.

(6) Electrical refrigerating apparatus, consisting of a compact noiseless unit that pumps fluid to various types of applications, is available. The applicators may be in the form of blankets containing rubber tubes suitable for covering the entire body, or all or part of a limb. Special applicators are available for insertion into various body cavities, and for inducing dental anesthesia.

(7) An air chamber at regulated temperature for treatments of frostbite and immersion foot, and amputation stumps.

The electrical apparatus is costly, but has the advantages of thermostatic regulation, light weight, freedom of movement, and permits prolonged treatments with heat, as well as cold over the range of temperatures therapeutically desirable.

ALLERGIC DISORDERS

Hay fever, asthma, eczema and contact dermatitis are classified as allergic disorders. The allergic individual responds to contact with a variety of substances, which are innocuous to a non-allergic person, with severe manifestations of hypersensitivity.

These substances are known as allergens and consist of airborne irritants such as dusts, molds, feathers, pollens, animal dander and others; of food protein such as milk, wheat, eggs, etc., or of simple chemicals brought in contact with the skin. They may enter the body by various routes of which inhalation is the most common type. Ingestion of offending food substances is not infrequent.

The offending substance reacts with the sensitized cells of the mucous membranes or skin. During this reaction, histamine or a histamine-like substance is released and causes (a) increased capillary permeability, (b) secretion of mucus and (c) muscular contraction. In the eyes and nose this produces itching, redness and lacrimation or rhinorrhea, in short, the symptoms of hay fever. In the lungs it causes, in addition to the secretory response, a contraction of the smooth muscles of the bronchi resulting in a bronchial asthma.

It is commonly known that non-specific environmental factors such as dust, irritating gases, change of temperature and humidity may precipitate asthmatic attacks in allergic subjects, even in the absence of exposure to specific allergens. It is assumed that the presence of frequent allergic bronchial constriction renders the smooth muscles of the bronchi so sensitive to various non-specific stimuli that the threshold of their response to such irritation is considerably lower than that of a non-allergic individual.

Air-Conditioning Apparatus

Of all the measures to relieve a specifically sensitive individual, elimination of exposure to the responsible allergen is the most efficient, though not always a practical, form of treatment. In recent years considerable effort has been made to accomplish such elimination by removal of respiratory allergens from enclosures by filtration or other air-conditioning processes.

Paper or cloth filters, mounted in inexpensive window or floor units, prove quite satisfactory in many cases, but since dust and smoke frequently cause asthmatic attacks, it is desirable that an air filter, to be of full value in the treatment of asthma, should remove all possible dusts and pollens regardless of size or amount. Electrostatic air cleaners are more efficient than most commonly used types for capturing very fine dust.

Although the chief remedial factor in the treatment by conditioned air is the filtration of pollen, a certain amount of cooling and dehumidification appears to be desirable. A comfortable temperature between 70 and 75 F, and a relative humidity well below 50 percent has proved satisfactory.[6] Direct drafts, overcooling or overheating are apt to initiate or aggravate the symptoms.

Limitations of Air-Conditioning Methods

The results obtained with air filtration, or other air conditioning processes, in the control of allergic conditions, are fairly comparable to those obtained by desensitization treatment, so long as the patients remain in the pollen-free atmosphere. For all practical purposes filtration gives only temporary relief. In mild cases sleeping in an air-conditioned space may make it possible for the individual to pass more comfortable nights. With rare exceptions, the symptoms recur on exposure to pollen-laden air. Moreover, the usefulness of air-conditioning methods is limited, because all cases are not caused by airborne substances. Cases of bacterial asthma do not respond to treatment with filtered air.

Despite these limitations, air-conditioning methods possess definite advantages in the simplicity of treatment, convenience, and under certain conditions, almost immediate relief.[7] Pollen cases are usually relieved of most of their symptoms within 1 to 3 hr after exposure to properly filtered air. A pollen-free atmosphere is especially valuable when desensitization has given little or no relief, and when desensitization is not advisable owing to intercurrent illness.

OXYGEN THERAPY

Oxygen therapy is used to prevent or relieve anoxia. Some of the more important clinical conditions in which oxygen treatment is beneficial include pneumonia, severe anemia, cardiac decompensation, pulmonary atelectasis, asphyxia and asthma. The effectiveness of oxygen therapy is dependent on the concentration of the oxygen in the inspired air, or the partial pressure of oxygen in the pulmonary alveoli.

Oxygen is usually administered by nasal catheter, face mask or tent.[8] The necessity of air conditioning in oxygen therapy arises from the fact that oxygen is too expensive a gas to waste in the ventilation of oxygen tents and oxygen chambers. Air conditioning is applied to the oxygen tent or chamber through reconditioning of the atmosphere in a closed circuit. Excessive heat, moisture and carbon dioxide are removed.

Oxygen Tents

In oxygen tents, the air enriched with oxygen is usually circulated by means of a small motor blower which sends the air over soda lime to remove carbon dioxide, and then over ice to remove excess heat and moisture. The concentration of oxygen in the tent is regulated by means of a pressure reducing valve and flow meter. In an inadequately cooled tent, high temperatures and humidities are inevitable, increasing the discomfort of the patient and imposing an added strain on an already overburdened heart. Oxygen therapy under such conditions may do more harm than good. An ice melting rate of approximately 10 lb per hr gives satisfactory results in patients with fever in a medium size oxygen tent.

Oxygen tents are confining to the patient. They may terrify the restless and delirious patient. Medical and nursing care is complicated, as the tent must be opened or removed with attendant loss of oxygen. Oxygen concentrations of 50 percent or more are difficult to maintain, and it is a problem to keep the temperature and humidity low enough in hot weather. However, with attention to details, the patient can be made quite comfortable. In fact, during hot humid weather an oxygen tent may be very valuable in controlling a patient's temperature, since the upper part of the body within the cooled tent loses heat rapidly.

Oxygen Chambers

The conventional oxygen chamber is an air-tight sheet-metal enclosure of fire-proof construction, large enough to accommodate one or two patients. Trap doors or curtains are provided for the personnel, food and service, to avoid loss of oxygen. Glass windows in the ceiling and walls admit light from outside the chamber. The air conditioning system may be of the gravity type, or of the fan type using mechanical refrigeration or air drying agents.

The temperature and humidity requirement in oxygen therapy depends primarily upon the physical condition of the patient, and secondarily upon the type of disease. In pneumonias[9] prescribed conditions should be a temperature of 60 to 75 F, humidity 50 to 55 percent, moderate air movement, oxygen concentration of 50 percent, and carbon dioxide of less than one percent.

Oxygen in Aviation

An important application of the principle of oxygen therapy is in aviation. High altitude military airplanes in this country are normally provided with gaseous oxygen equipment, and military personnel are required to utilize oxygen at all times while in flight above 15,000 ft, or between 12,000 to 15,000 ft for longer than two hours, or between 10,000 to 12,000 ft for longer than six hours.

Today all large passenger aircraft are pressurized to provide passenger comfort in flight at altitudes of 20,000 feet or more. Smaller civilian aircraft are usually not pressurized and must carry gaseous oxygen equipment where appreciable altitudes must be attained.

GENERAL HOSPITAL AIR CONDITIONING

Complete conditioning of a large hospital involves a capital investment and running expenses which may not be justified. In clean and quiet districts, the requirements of almost all general and private wards during the cool season of the year can be satisfactorily fulfilled by the use of conventional heating equipment, in conjunction with window air supply and gravity or mechanical exhaust. Insulation against heat and sound is much more important than humidification in winter; thermal insulation will also help in keeping the building cool in warm weather. Excessive outside noise and dust may require the use of silencers and air filters in the openings.

Cooling and dehumidification in warm weather are important. In new hospitals particularly, the desirability of cooling certain sections of the building should be given serious consideration. Financial reasons may preclude the cooling of the entire building, but the needs of the average hospital can be met by the use of built-in room coolers and a few portable units which can be wheeled from ward to ward when needed.

In the North, and certain sections of the Pacific Coast, cooling is needed but a few days during summer, while in the South, it can be used to advantage from May to October, and in tropical climates almost continuously throughout the year.

F. L. Grocott of the Anglo-Iranian Oil Co. states that in Iran, the medical staff after 10 years' experience with air conditioning, demand a uniform environment of 75 F and 50 percent relative humidity under all summer outdoor conditions for general wards and treatment rooms, and 70 F with 30 to 50 percent relative humidity for winter conditions. In the operating rooms, 70 F with 50 percent relative humidity is demanded all the year 'round, although the annual external range is 40 F to 120 F. No ill effects have been noted in the medical personnel, though they are exposed to changes from external to internal conditions many times daily. Temperature shock in either direction seems to create discomfort for a short interval, but if the individual is in good health, no injury results.[10]

Aside from comfort and recuperative power of the patients, cooling is of great assistance in the treatment of fevers in the new-born and in post-operative cases, in enteric disorders, fevers, heat stroke, heart failure, thyroid crisis, and in a variety of other ailments which often accompany summer heat waves.

Problem of Odors

Experiences in the evacuation of battle casualties in aircraft and their subsequent hospitalization have stimulated efforts to find ways of minimizing odors arising from draining wounds, old odorous casts, and gangrenous wounds. For aircraft, chemical sprays and vapors, perfumes, oxidizing gases and simple exhaust methods are unsatisfactory. An ideal deodorant would purify the air by means of odor adsorption so that subsequently the air could be recirculated. Based upon the effectiveness of activated carbon commercially and industrially to adsorb odors, individual adsorption units have been used successfully. In hospital wards the feasibility of adsorption methods over other methods for elimination of odors remains to be answered.[11]

REFERENCES

[1] *The Effects of Atomic Weapons* (U. S. Government Printing Office, revised September 1950).

[2] J. G. Townsend, M.D.: Investigation of the smog incident in Donora, Pa., and vicinity (*American Journal of Public Health* 40: 183, February 1950).

[3] C. P. McCord, M.D.: The physiologic aspects of atmospheric pollution (*Industrial Medicine and Surgery* 19:97, March 1950).

[4] R. B. Bourdillon, O. M. Lidwell, and J. E. Lovelock with W. C. Cawston, L. Colebrook, F. P. Ellis, M. V. Ende, R. E. Glover, A. M. Macfarlan, A. A. Miles, W. F. Raymond, E. Schuster, and J. C. Thomas: *Studies in Air Hygiene* (Medical Research Council of Great Britain, 1948, *Special Report Series* No. 262).

[5] H. E. Ronge: *Ultraviolet Irradiation with Artificial Illumination* (Uppsala, Sweden, 1948).

[6] P. A. Burrucand: *L'Hygiene du Batiment per L'Ultraviolet* (Centre Scientific et Technique du Batiment, Paris, 1949).

[7] Council of Physical Therapy: Acceptance of ultraviolet lamps for disinfecting purposes (*Journal of American Medical Association* 122: 503, 1943).

[8] J. E. Perkins: Evaluation of methods to control airborne infection (*American Journal of Public Health* 35:891, 1945).

[9] Committee on Evaluation of Methods to Control Airborne Infections of the American Public Health Association: The present status of the control of airborne infections (*American Public Health Association Journal* 37:13, 1947).

[10] Subcommittee on Air Sanitation: Progress in the control of airborne infections (*American Public Health Association Yearbook* 40:5, May 1950, p. 82).

[11] S. M. Wheeler and T. D. Jones: Factors in the control of the spread of acute respiratory infections with reference to streptococcal illness and acute rheumatic fever (*American Journal of the Medical Sciences* 209:58, 1945).

[12] C. G. Loosli: Dust control, report on suppressive measures for the control of dust to the Standards Methods Committee for the Examination of Germicides and Antibacterial Agents (*American Journal of Public Health*, March 1948, p. 409).

[13] A. P. Krueger et al: Laboratory and field studies of glycols and floor-oiling in the control of air-borne bacteria (*U. S. Naval Medical Bulletin* 42:1288, 1944).

[14] E. W. Dunklin and T. T. Puck: The lethal effect of relative humidity on air-borne bacteria (*Journal of Experimental Medicine* 87:87, February 1948).

[15] Quotation by William Lester, Jr. in *Health on the Job* No. 22 (Institute of Industrial Medicine, New York University, February 1949).

[16] H. M. Lemon, H. Wise, and M. Hamburger: Bacterial content of air in army barracks (*War Medicine* 6:92, 1944).

[17] O. H. Robertson, M. Hamburger, C. G. Loosli, T. T. Puck, and H. M. Lemon: A study of the nature and control of airborne infection in army camps (*Journal of the American Medical Association* 126:993, 1944).

[18] O. H. Robertson et al: Lethal effects of triethylene glycol vapor on air-borne bacteria and influenza virus (*Science* 97: 142, 1943).

[19] M. Hamburger, Jr., O. H. Robertson, and T. T. Puck: The present status of glycol vapors in air sterilization (*American Journal of the Medical Sciences* 209:162, 1945).

[20] T. N. Harris and J. Stokes, Jr.: Summary of a 3-year study of the clinical applications of the disinfection of air by glycol vapors (*American Journal of the Medical Sciences* 209:152, 1945).

[21] William Lester, Jr., Saul Kaye, O. H. Robertson, and Edward Dunklin: Factors of importance in the use of triethylene glycol vapor for aerial disinfection (*American Journal of Public Health* 40:813, July 1, 1950).

[22] Edward Bigg, B. H. Jennings, and F. C. W. Olson: Triethylene glycol vapor distribution for air sterilization (ASHVE Transactions, Vol. 53, 1947, p. 393).

[23] Glycol vapors for disinfecting purposes (*Journal of the American Medical Association* 133:696, March 8, 1947).

[24] W. J. McConnell: An experiment with triethylene glycol vapor for the control of colds among office employees (*Industrial Medicine* 18:5, page 192).

[25] S. M. Wheeler, H. S. Ingraham, A. Hollaender, N. D. Lill, J. Gershon-Cohen, and E. W. Brown: Ultra-violet light control of airborne infections in a naval training center (*American Journal of Public Health*, Vol. 35, 1945, p. 457).

[26] H. G. DuBuy, J. E. Dun, F. S. Brackett, W. C. Dressen, P. A. Nealand, and I. Possner: An evaluation of ultraviolet radiation of sleeping quarters as supplement of accepted methods of disease control (*American Journal of Hygiene*, September 1948, p. 207).

[27] Recent studies on disinfection of air in military establishments (*American Journal of Public Health*, February 1947, p. 189).

[28] Commercial exploitation of glycol vaporizers (*American Journal of Public Health*, February 1949, p. 222).

[29] N. Pace, M. B. Fisher, J. E. Birren, G. C. Pitts, W. A. White, Jr., W. V. Consolazio, and L. J. Pecora: *A Comparative Study of the Effect on Men of Continuous Versus Intermittent Exposure to a Tropical Environment* (Research Project X-205, *Naval Medical Research Institute Report* No. 2, May 1945).

[30] F. K. Hick and M. M. Montgomery: *Mechanism of Heat Retention* (unpublished).

[31] A. R. Behnke: *Environmental and Physiologic Studies Aboard an Air-Cooled Hospital Ship En Route from Norfolk, Virginia, to Canal Zone* (*U.S.S. Tranquility*, AH-14, June 6-13 1945) (Research Project X-205, *Naval Medical Research Institute Report* No. 4, September 1945).

[32] G. E. Burch: The influence of environmental temperature and relative humidity on the rate of water loss through the skin in congestive heart failure in a subtropical climate (*American Journal of Medical Science*, Vol. 211, 1946, p. 181). G. Berenson and G. E. Burch: The responses of patients with congestive heart failure to rapid elevation in atmospheric temperature and humidity (*American Journal of Medical Science*, Vol. 223, 1952, p. 45).

[33] Gunner Edstrom, G. Lundin, and T. Wramer: Investigation into the effect of hot, dry microclimate on peripheral circulation in arthritic patients (*Annals of Rheumatic Diseases* 7:76, June 1948).

[34] *Fundamentals of Anesthesia* (American Medical Association Press, Chicago, 1944, 2nd ed., p. 204).

[35] B. A. Green: The hazard of fire and explosion in anesthesia (*Anesthesiology* 2:144, 1941).

[36] R. M. Tovell and A. W. Friend: Control of physical hazards of anesthesia (*Canadian Medical Association Journal* 46: 560, 1942).

[37] F. C. Houghten and W. L. Cook, Jr.: ASHAE Research Report No. 1111—Air conditioning requirements of an operating room and recovery ward (ASHVE Transactions, Vol. 45, 1939, p. 161).

[38] A. R. Behnke and O. Schneider: Unpublished naval studies (1940).

[39] C. P. Yaglou and Ursula Wilson: *Disinfection of Air by Air Conditioning Processes* (American Association for the Advancement of Science, Publication No. 17, p. 129).

[40] J. A. Reyniers: The control of cross-contamination by the use of mechanical barriers (American Association for the Advancement of Science, *Aerobiology* 17:254, 1942).

[41] K. D. Blackfan, C. P. Yaglou, and K. McKenzie: The premature infant: a study of the effect of atmospheric conditions on growth and on development (*American Journal Diseases of Children* 46:1175, 1933).

[42] I. Rosenstern: Observations on the control of respiratory contagion in the cradle (American Association for the Advancement of Science, *Aerobiology* 17:242, 1942).

[43] F. H. Krusen: *Physical Medicine* (W. B. Saunders Co., Philadelphia, 1941).

[44] F. C. Houghten, M. B. Federber, and Carl Gutberlet: ASHVE Research Report No. 1054—Fever therapy induced by conditioned air (ASHVE Transactions, Vol. 46, 1940, p. 131). M. B. Federber, F. C. Houghten, and Carl Gutberlet: ASHVE Research Report No. 1161—Fever therapy locally induced by conditioned air (ASHVE Transactions, Vol. 46, 1940, p. 307).

[45] L. W. Crossman and S. K. Safford: Refrigeration for anesthesia and therapy (*The Modern Hospital* 64:86, 1945).

[46] B. Z. Rappaport, T. Nelson, and W. H. Welker: The effect of low relative humidity and constant temperature on pollen asthma (*Journal of Allergy* 6:111, 1935).

[47] C. P. Yaglou: Hospital air conditioning (*The Environment and Its Effect Upon Man*, Harvard School of Public Health, 1939, p. 244).

[48] A. L. Barach: *Principles and Practices of Inhalational Therapy* (J. B. Lippincott Co., Philadelphia, 1944).

[49] J. G. M. Bullowa: *The Management of Pneumonia* (Oxford University Press, 1937, p. 260).

[50] Cyril Tasker: What are the right conditions for comfort cooling? (*Heating, Piping and Air Conditioning*, August 1948, p. 84).

[51] C. P. McCord and W. R. Witheridge: *Odors Physiology and Control* (McGraw-Hill Book Co., New York, 1949).

BIBLIOGRAPHY

R. P. Gaulin: Air conditioning the hospital (*Reference Section, Air Conditioning, Heating & Ventilating*, January 1957, p. 73).

W. W. Treichler, Jr.: Air conditioning the operating room (*Air Conditioning, Heating & Ventilating*, April 1957, p. 73).

CHAPTER 9

HEAT TRANSMISSION COEFFICIENTS OF BUILDING MATERIALS

Heat Transfer Symbols; Calculating Overall Coefficients; Conductivity of Homogeneous Materials; Soil Conductivity and Specific Heat; Surface and Air Space Conductance; Overall Coefficients and Their Practical Use; Computed Coefficients of Walls, Roofs, Ceilings, and Floors; Effect of Insulation; Combined Ceiling, Roof, and Floor Coefficients; Glass Coefficients; Calculating Surface Temperatures

THE design of air-conditioning or heating systems for buildings requires a knowledge of the thermal properties of the walls enclosing the space. (The term *walls* in this case, includes windows, doors, ceilings, floors, roofs, and skylights.) The rate of heat flow through the walls under steady-state conditions at design temperatures is usually the basis for calculating the heat required. For a given wall under standard conditions the rate is a specific value designated as U, the *overall coefficient of heat transmission* or thermal transmittance. It may be determined by test in a guarded hot box apparatus, or it may be computed from known values of the thermal conductance of the various components. Because it is impracticable to test all combinations of building materials, the procedure and necessary data for calculation of the value of U are given in this chapter, together with convenient tables of computed values for a large number of the more common constructions.

HEAT TRANSFER SYMBOLS

U = overall coefficient of heat transmission or thermal transmittance (air-to-air); the time rate of heat flow expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference between air on the inside and air on the outside of a wall, floor, roof, or ceiling). The term is applied to the usual combinations of materials, and also to single materials, such as window glass, and includes the surface conductance on both sides. This term is frequently called the U value.

k = thermal conductivity; the time rate of heat flow through a homogeneous material under steady conditions per unit temperature gradient through unit area perpendicular to the temperature gradient. Its value is expressed in Btu per (hour) (square foot) (Fahrenheit degree per inch of thickness). Materials are considered homogeneous when the value of k is not affected by variation in thickness or size of sample within the range normally used in construction.

C = thermal conductance; the time rate of heat flow through a unit area of a material from one of its surfaces to the other per unit temperature difference *between the two surfaces*. Its value is expressed in Btu per (hour) (square foot) (Fahrenheit degree). The term is applied to specific materials as used, either homogeneous or heterogeneous, for the thickness or construction stated, not per inch of thickness.

f = film or surface conductance; the time rate of heat flow between a unit area of a surface and the surrounding air. Its value is expressed in Btu per (hour) (square foot of surface) (Fahrenheit degree temperature difference). Subscripts i and o are used to differentiate between inside and outside surface conductances, respectively.

a = thermal conductance of an air space; the time rate of heat flow through a unit area of an air space per unit temperature difference *between the boundary surfaces*. Its value is expressed in Btu per (hour) (square foot of area) (Fahrenheit degree). The conductance of an air space is dependent on the temperature difference, the height, the depth, the position, character, and temperature of the boundary surfaces. Since the relationships are not linear, accurate values must be obtained by test and not by computation.

e = emissivity; the ratio of the total radiant flux emitted by a surface to that emitted by an ideal black body at the same temperature.

E = effective emissivity; the combined effect of the surface emissivities e of the boundary surfaces of an air space; the boundaries assumed to be parallel and of large dimensions as compared to the distance between them.

r = surface reflectivity; the ratio of the radiant flux reflected by an opaque surface to that falling upon it.

R = thermal resistance. Its value is obtained from the reciprocal of heat transfer as expressed by U, k, C, f, or a. It may be expressed in (Fahrenheit degrees per Btu)/(hour) (square foot). For example, a wall with a U value of 0.25 would have a resistance value of $R = 1/0.25 = 4.0$ *ru*. The word *ru* has been suggested as an abbreviation for resistance unit.

CALCULATING OVERALL COEFFICIENTS

From Chapter 5, Equation 7, the total resistance to heat flow through a wall is equal numerically to the sum of the resistances in series.

$$R_T = R_1 + R_2 + R_3 + R_4 + \cdots + R_n \qquad (1)$$

where R_1, R_2, etc., are the individual resistances of the wall components, and R_T is total resistance.

For a wall of a single homogeneous material of conductivity k and thickness x with surface coefficients f_i and f_o,

$$R_T = \frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o} \qquad (2)$$

Then by definition, $U = 1/R_T$

For a wall with air space construction and consisting of two homogeneous materials of conductivities k_1 and k_2, thicknesses x_1 and x_2, respectively, and separated by an air space of conductance a,

$$R_T = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o} \qquad (3)$$

and $U = 1/R_T$

For types of building materials having non-uniform or irregular sections such as hollow clay tile or concrete blocks, it is necessary to use the conductance C of the section unit as manufactured instead of a conductivity k. The resistance of the section $1/C$ is therefore substituted for x/k in Equations 2 and 3.

It will be noted that in order to compute the U value of a construction it is first necessary to know the conductivity

and thickness of the homogeneous material, the conductance of non-homogeneous materials (such as concrete blocks), the surface conductances of both sides of the construction, and the conductances of any contained air spaces. These items are discussed in the pages that follow.

Conductivities and Conductances

The method of calculating the overall coefficient of heat transmission for a given construction is comparatively simple, but accurate values of conductivities and conductances must be used to obtain satisfactory results. In addition, there are sometimes parallel heat-flow paths of different resistances in the same wall, and these may necessitate modification of the formula. In such cases calculated results should be checked by test.

The determination of the fundamental conductivities and conductances requires considerable skill and experience to obtain accurate results. It is recommended that thermal conductivities of homogeneous materials be determined by means of the Guarded Hot Plate.[1] For determination of conductances, a Guarded Hot Box method[2] is generally used.

Conductivity of Homogeneous Materials

Thermal conductivity is a property of a homogeneous material and of types of building materials such as lumber, brick, and stone, which may be considered homogeneous. Most insulating materials, except reflective types, are of a porous nature and consist of combinations of solid matter with small voids. Such materials including fibrous, cellular, or granular matter are generally known as mass or bulk insulations. The thermal conductivity of these materials will vary with density; mean temperature; size of voids, fibers, or particles; degree and extent of bond between particles; moisture present; and the arrangement of fibers or particles within the material.

The effect of density upon conductivity (at constant mean temperature) is illustrated for two fibrous materials in Fig. 1. Typical variation of conductivity with mean temperature is shown in Fig. 2.

Thermal Conductivity of Soil

The following statements are based largely on results of a study[3] made in the Engineering Experiment Station, University of Minnesota, and published in *Bulletin* No. 28. Tests were made on nineteen different soils which represented a wide textural variety, including gravel, sand, sandy loam, silt loam, and clay, as well as some crushed rocks and a fibrous peat. Moisture contents in tests varied from air-dried values

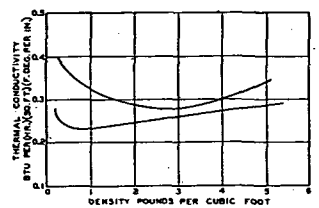


Fig. 1 . . . . Typical Variation of Thermal Conductivity with Density—for Fibrous Material

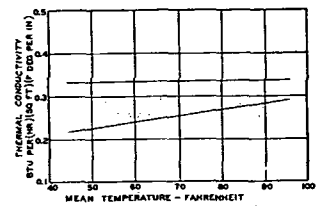


Fig. 2 . . . . Typical Variation of Thermal Conductivity with Mean Temperature

to those greater than the optimum moisture content; densities varied from a loosely-poured condition to the maximum density obtainable by heavy ramming. The general findings of the investigation are as follows:

Effect of Temperature. Soils were tested at several mean temperatures. The degree of influence of temperature depends upon whether it is above or below freezing. For increases of moisture content exceeding about 6 to 12 percent, the conductivity of frozen soil becomes progressively greater than that of the unfrozen soil.

Effect of Density. Density affects the thermal conductivity of a soil in about the same manner for all soils, at any moisture content, and for either the frozen or unfrozen condition. On the average, each one pound per cubic foot increase in dry density increases the thermal conductivity by about 3 percent.

Effect of Moisture. An increase in moisture content, up to the point of saturation, causes an increase in thermal conductivity. The rate of increase in typical soils was as follows: average conductivities, in Btu per (square foot) (hour) (Fahrenheit degree per inch), of four sands at a density of 110 lb per cu ft were: 6.8 at 2.5 percent moisture, 8.9 at 5 percent moisture, 11.2 at 10 percent moisture. Five soils of a fine texture at a density of 100 lb per cu ft, gave average conductivities of 6.7 at 10 percent moisture, and 9.5 at 20 percent. Thus, the doubling of moisture content within the ranges cited increases the conductivity by approximately 30 or 40 percent. At higher moisture contents the percentage increase would be less.

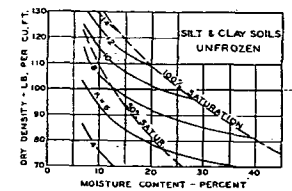
Effect of Soil Characteristics. The thermal conductivity of the soil, at a given density and moisture content, varies in general with the texture of a soil, being relatively high for coarse-textured soils and relatively low for fine-textured soils. The mineral composition of the soils also affects the conductivity. Quarts tends to give high values, whereas minerals such as plagioclase-feldspar and pyroxene, which are constituents of basic rocks tend to give low values of thermal conductivity. These points are illustrated by the values in Table 1 which lists seventeen soils in approximate order of their magnitude of thermal conductivity from greatest to least for seven different density-moisture content conditions. Some of the values in this table have been determined by extrapolation and are consequently approximate. Blank spaces in the table indicate that the density or moisture content, or both, are such that no tests were possible for that condition or that no tests were sufficiently close to permit a reasonable extrapolation of the data. Granular soils, particularly those with high quartz contents, head the tabulation or have the greatest conductivity

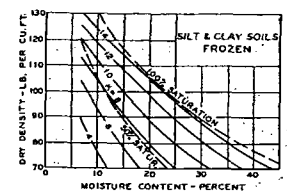
Table 1 . . . . Thermal Conductivity (*k*) Values of Soils in Approximate Order of Decreasing Values[a]

Mean Temperature—40 F

Soil Designation	Mechanical Analysis % by Weight				Moisture Content—%						
	Gravel	Sand	Silt	Clay	4			10		20	
					Dry Density-lb per cu ft						
	Over 2.0 mm	0.5 to 2.00 mm	0.005 to 0.05 mm	Under 0.005 mm	100	110	120	90	110	90	100
Fine Crushed Quartz	0.0	100.0	0.0	0.0	12.0	16.0					
Crushed Quarts	15.5	79.0	5.5		11.5	16.0	22.0				
Graded Ottawa Sand	0.0	99.9	0.1		10.0	14.0					
Fairbanks Sand	27.5	70.0	2.5		8.5±	10.5	13.5		15.0		
Lowell Sand	0.0	100.0	0.0	0.0	8.5	11.0			13.5		
Chena River Gravel	80.0	19.4	0.6			9.0±	13.0				
Crushed Feldspar	25.5	70.3	4.2		6.0	7.5	9.5				
Crushed Granite	16.2	77.0	6.8		5.5	7.5	10.0				
Dakota Sandy Loam	10.9	57.9	21.2	10.0		6.5	9.5		13±		
Crushed Trap Rock	27.0	63.0	10.0		5.0	6.0	7.0				
Ramsey Sandy Loam	0.4	53.6	27.5	18.5	4.5	6.5			10.0		
Northway Fine Sand	0.0	97.0	3.0	0.0	4.5	5.5			8.5		
Northway Sand	3.0	97.0	0.0	0.0	4.5	6.0			7.5±		
Healy Clay	0.0	1.9	20.1	78.0	4.0±			5.5	9.0±	8.0	10.0
Fairbanks Silt Loam	0.0	7.6	80.9	11.5				5.0	9.0±	7.5	10.0
Fairbanks Silty Clay Loam	0.0	9.2	63.8	27.0				5.0	9.0±	7.5	9.5
Northway Silt Loam	1.0	21.0	64.4	13.6				4.0±	7.0±	6.0±	7.0±

[a] *k* = Btu per (square foot) (hour) (Fahrenheit degree per inch).





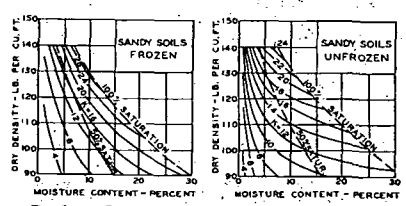


Fig. 3 . . . . Determining Thermal Conductivity of Soils from Density and Moisture Content

at a given condition. Sandy loam soils are midway in the table and fine grained soils such as clay and silt loam are last.

Estimating Thermal Conductivity. The four diagrams of Fig. 3 are presented to aid in the estimate of the thermal conductivity of any soil. Two of the charts are for sands or sandy soils, and two for silt and clay soils. One of the diagrams for each type of soils is for the frozen, and the other for the unfrozen condition. It is expected that these charts will give conductivity values with a precision of 25 percent. The effect of such factors as density, moisture content, freezing, or texture may be easily approximated by use of these graphs.

Specific Heat of Soils

Tests to determine specific heat were run on twelve soils. On five of the soils, tests were made at three or four mean temperatures varying from about 10 to 140 F. The specific heat values of all twelve soils varied by only a small amount (about 0.01), and averaged 0.19 at 140 F. The specific heat values of the soils decreased with a decrease in temperature. The average value at zero F would be about 0.16. Values at temperatures between zero and 140 F can be estimated by considering a straight-line relationship between the two values given.

Surface Conductance

The surface conductance of a wall is the combined heat transfer to or from the wall by radiation, convection, and conduction. Each of the three portions making up the total may vary, independently of the others, thus affecting the total conductance. The heat transfer by radiation between two surfaces is controlled by the character of the surfaces (emissivity), the temperature difference between them, and the solid angle through which they *see* each other. The heat transfer by convection and conduction is controlled by the roughness of the surface, by air movement, and temperature difference between the air and the surface.

The importance of the effect of temperature of *surrounding surfaces* on the surface conductance, due to the effect on radia-

Table 2 . . . . Variation in Surface Conductance Coefficient for Vertical Surfaces with Different Temperatures of Surrounding Surface

Surrounding Surface Temperature	75 F	70 F	69 F	60 F	50 F
Convection—Btu per (hr) (sq ft)	6.6	6.6	6.6	6.6	6.6
Radiation—Btu per (hr) (sq ft)	4.4	8.6	9.6	17.0	24.9
Total—Btu per (hr) (sq ft)	11.0	15.2	16.2	23.6	31.5

tion, is illustrated in Table 2, which applies to a vertical surface at 80 F, with ambient air at 70 F and with radiation exchange corresponding to an effective emissivity of 0.83.[4]

In many cases, because the heat resistance of the internal parts of the wall is high compared with the surface resistance, the surface factors are of minor importance. In other cases, e.g., single glass windows, the surface resistances constitute almost the entire resistance and are therefore very important. An analysis of various factors affecting surface conductance and the difference between surface and air temperatures will be found in Reference 5. (See also Chapter 30.)

The convection part of the surface conductance is affected markedly by air movement. This is illustrated by Fig. 4, which shows the results of tests[6] made on 12-in. square samples of different materials at a mean temperature of 20 F, and for wind velocities up to 40 mph. These conductances include the radiation portion of the coefficient which, for the conditions of the tests, was about 0.7 Btu per (hr) (sq ft) (F deg). More recent tests[7] on smooth surfaces show that surface length also affects significantly the convection part of conductance; the average value decreases as the surface length increases. Moreover, observations[8] of the magnitude of low temperature radiant energy received from outdoor surroundings show that only under certain conditions may the out-of-doors be treated as a black body radiating at air temperature.

Because of these factors, the selection of surface conductance coefficients for a practical building becomes a matter of judgment. Surface conductances are shown in Tables 3 and 4. *In calculating the overall heat transmission coefficients for the walls, etc., of Tables 5 through 16, the appropriate indoor and outdoor surface coefficients given in Table 4 for Air Surfaces have been used.* Both values combine the effects of convection and radiation, and are applicable to ordinary building materials. They should not be used for low emissivity surfaces such as bright metal. For exposed reflective surfaces refer to Table 3, Section A, and to footnotes under Table 16.

In special cases, where surface conductances become important factors in the overall rates of heat transfer, more selective coefficients may be required. Principles and data given in Chapter 5, Heat Transfer, may be applied in such cases.

Air Space Conductance

The transfer of heat across an air space involves the boundary surfaces as well as the intervening air, and depends markedly on the orientation of the air space and the direction of heat flow. The coefficients given for air space conductance represent the total conductance from one surface bounding the air space to the other. The total conductance is the sum of a component due to radiation and a component due to convection and conduction combined. These components may vary independently of each other.

The radiation portion of the coefficient is affected by the temperature of the two boundary surfaces, and by their respective surface emissivities ϵ, the combined effect of which is expressed by means of the *effective emissivity* E of the air space. The radiation component is not affected by the thickness of the space or by its orientation or direction of heat flow. The heat transfer by convection and conduction combined, however, is markedly affected by the orientation of the air space and the direction of heat flow, is significantly affected by the temperature difference across the space and in some cases by the thickness of the space, and is affected to only a small extent by the mean temperature of its surfaces. For air spaces usually employed in building construction, the radiation and convection-conduction components may vary independently of each other.

Table 3, Section C, gives the thermal conductances and resistances of air spaces of uniform thickness and moderately

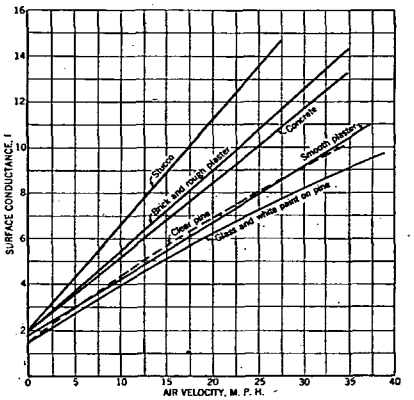


Fig. 4 . . . . Curves Showing Relation Between Surface Conductance for Different Surfaces at 20 F Mean Temperature

smooth surfaces, based on experimental measurements conducted at the National Bureau of Standards.[9] Although the conductances of air spaces vary to some extent with thickness in the range over ¾ in., average values are tabulated for the range from ¾ in. to 4 in., for all except horizontal spaces with heat flow downward. The error involved by averaging is less than 10 percent in the extreme case and less than 5 percent in most. For more exact values Reference 9 may be consulted.

For narrow air spaces, which may be defined as those for which the product of the cube of the thickness of the space in inches times the temperature difference (Fahrenheit degrees) across the space is less than 3 for heat flow horizontally or downward, or less than 1 for heat flow upward, the conductance is the sum of the radiative heat transfer coefficient and that for conduction alone through air, since convection is practically suppressed. The radiation component can be computed by means of Equation 4 and Table 4 of Chapter 5; the conduction component can be computed using the conductivity of air at the appropriate mean tempearture (see Table 1, Chapter 5).

The effects of different mean temperatures, temperature differences, and effective emissivities are indicated in Table 3 of this chapter, Section C. As indicated, use may be made of interpolation and moderate extrapolation of conductance values in the table to obtain conductances for conditions moderately different from those given. Interpolation of resistance values is not recommended, especially in relation to emissivity values.

Table 3, Section B, gives values for the surface reflectivities and emissivities of materials used as boundaries of air spaces in building construction, for total radiation at ordinary building temperatures. Effective emissivities for various combinations of these materials, for use in conjunction with Section C of Table 3, are given in the last two columns of Section B.

When considering heat transfer across air spaces in building construction, the emissivities of the boundary surfaces must be known. The possibility of change in emissivity of highly reflective surfaces due to exposure to conditions promoting chemical action, deposition of dust, soiling of the surface, or the application of coatings, even though transparent to the eye, must be considered in selecting a material for use.[10] Surface emissivity values should be obtained by tests.

OVERALL COEFFICIENTS AND THEIR PRACTICAL USE

The values in Tables 3 and 4 for component elements and materials were selected by the ASHAE Technical Advisory Committee on Insulation as representative for dry materials at 75 F mean temperature. They are based on available published data obtained by the guarded hot plate method (*ASTM* C177-45) or by the guarded hot box method (*ASTM* C236-54T). Because there are variations in commercially available materials of the same type, not all of these selected representative values will be in exact agreement with data for individual products. The exact value for a certain manufacturer's material can be secured from unbiased tests or from guaranteed data of the manufacturer.

The most exact method of determining the heat transmission coefficient for a given combination of building materials assembled as a building section is to test a representative section in a guarded hot box. However, it is not practicable to test all the combinations which may be of interest in building construction. Experience has indicated that *U* values for many constructions, when calculated by the methods given in this chapter, using accurate values for the component materials, are in good agreement with values determined by guarded hot box measurements.

Caution

Although the validity of calculating *U* values for all of the types of constructions in Tables 5 through 16 has not been fully demonstrated, calculated values are given because measured values are not available. It is emphasized that the calculated values in Tables 5 through 16 are given for the convenience of the user.

In calculating *U* values, exemplary conditions of components and installations are assumed, i.e., that insulating materials are uniformly of the nominal thickness and conductivity, that air spaces are of uniform thickness and surface temperatures, that effects due to moisture are not involved, and that installation details are in accordance with design. Some evidence of departures of measured values from calculated values for certain insulated constructions is given in *Building Materials and Structures Report* BMS 151, National Bureau of Standards. In order to provide a reasonable factor of safety to account for departures of constructions from exemplary conditions, in part due to field construction requirements and practices, some may wish, before making corrections for framing (as indicated in Fig. 6), to increase moderately the calculated *U* values of the insulated walls, floors and ceiling sections obtained from Table 16. Where reflective air spaces are involved, increases of *U* values up to 10 percent for applications where heat flow is horizontal or upward, and up to 20 percent where heat flow is downward, appear reasonable on the basis of present information.

Computed Heat Transmission Coefficients

In the analysis of any wall construction for the purpose of calculating the overall coefficient of heat transmission *U*, it is first necessary to determine the paths of heat flow, that is,

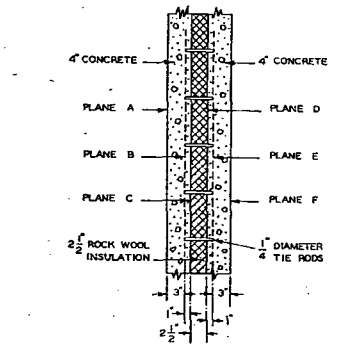


Fig. 5 . . . . Section of Concrete Wall Having Steel Tie Rods and Insulation

whether they are parallel or series, or a combination of both. This is in accordance with the basic laws of heat transfer which state that *in parallel flow the conductances are additive*, while *in series flow the resistances are additive*. Likewise, in order to determine the total resistance for the wall, the conductance must be known.

The importance of this analysis cannot be overemphasized. This is especially true in wall constructions in which there are parallel paths of heat flow, and one path has a high heat transfer while others have a low heat transfer. The method of making this calculation can best be shown by *Example 1* and Fig. 5. As this wall was tested by the hot box method at the University of Minnesota, a direct comparison can be made between calculated and tested values.

Example 1: Calculate the coefficient of heat transmission *U* for a wall shown in Fig. 5. Wall construction consists of two 4-in. concrete walls separated by a 2½-in. space filled with insulation; ¼-in. diameter metal tie rods are embedded a distance of 1 in. in each 4-in. concrete wall, and spaced 9 in. vertically and 12 in. horizontally. Values of *k* are: insulation 0.30, concrete 12.00, tie rods 400.00.

(Continued on p. 103.)

Table 3 .... Surface Conductances and Resistances for Still Air

All conductance values expressed in Btu per (hr) (sq ft) (F deg temp diff)

SECTION A. Surface Conductances for Still Air[a, b]

Position of Surface	Direction of Heat Flow	Surface Emissivity: Nonreflective ε = 0.90 C	Nonreflective ε = 0.90 R	Reflective ε = 0.20 C	Reflective ε = 0.20 R	Reflective ε = 0.05 C	Reflective ε = 0.05 R
Horizontal	Upward	1.63	0.61	0.91	1.10	0.76	1.32
Sloping—45 deg	Upward	1.60	0.62	0.88	1.14	0.73	1.37
Vertical	Horizontal	1.46	0.68	0.74	1.35	0.59	1.70
Sloping—45 deg	Downward	1.32	0.76	0.60	1.67	0.45	2.22
Horizontal	Downward	1.08	0.92	0.37	2.70	0.22	4.55

SECTION B. Reflectivity and Emissivity Values of Various Surfaces[c] and Effective Emissivities of Air Spaces

Surface	Reflectivity in percent	Average Emissivity ε	Effective Emissivity E of Air Space: With one surface having emissivity ε and other 0.90	Effective Emissivity E of Air Space: With both surfaces of emissivity ε
Aluminum foil, bright	92 to 97	0.05	0.05	0.03
Aluminum sheet	80 to 95	0.12	0.12	0.06
Aluminum coated paper, polished	75 to 84	0.20	0.20	0.11
Steel, galvanized, bright	70 to 80	0.25	0.24	0.15
Aluminum paint	30 to 70	0.50	0.47	0.35
Building materials: wood, paper, glass, masonry, nonmetallic paints	5 to 15	0.90	0.82	0.82

[a] For ventilated attics or spaces above ceilings under summer conditions (heat flow down) see Table 17.

[b] Conductances are for surfaces of the stated emissivity facing surroundings having an emissivity equal to 0.9 and at the same temperature as the ambient air. Values are based on a surface-air temperature difference of 10 deg and for surface temperature of 70 F. (See Table 4 for surface conductances for moving air.)

[c] See also Chapter 5, Table 3.

SECTION C. Thermal Conductances and Resistances of a Plane[d, e] Air Space[e]

Position of Air Space	Heat Flow	Air Space: Thickness,[e] inches	Mean temp,[f] F	Temp diff[f] deg	Thermal Conductance-C Value of E[f, g]: 0.05	0.2	0.5	0.82	Thermal Resistance-R Value of E[g]: 0.05	0.2	0.5	0.82
Horiz.	Up	¾ to 4	50	10	0.41	0.55	0.82	1.11	2.44	1.83	1.22	0.90
			50	30	0.54	0.68	0.95	1.24	1.84	1.47	1.05	0.80
			90	10	0.41	0.55	0.92	1.28	2.47	1.73	1.09	0.78
45° Slope	Up	¾ to 4	50	10	0.35	0.49	0.76	1.05	2.84	2.05	1.31	0.95
			50	30	0.48	0.62	0.89	1.18	2.08	1.62	1.13	0.85
			90	10	0.35	0.52	0.86	1.23	2.85	1.92	1.16	0.81
Vert.	Horiz.	¾ to 4	50	10	0.28	0.42	0.69	0.99	3.52	2.37	1.44	1.02
			50	30	0.38	0.52	0.79	1.08	2.64	1.94	1.27	0.92
			90	10	0.29	0.46	0.80	1.16	3.49	2.19	1.25	0.86
45° Slope	Down	¾ to 4	50	10	0.24	0.38	0.65	0.94	4.14	2.65	1.54	1.06
			50	30	0.30	0.43	0.71	1.00	3.36	2.30	1.41	1.00
			90	10	0.25	0.42	0.76	1.12	4.07	2.40	1.32	0.89
Horiz.	Down[h]	¾	50	20	0.28	0.42	0.69	0.98	3.57	2.38	1.45	1.02
		1½	50	20	0.18	0.31	0.58	0.87	5.56	3.23	1.73	1.15
		4	50	20	0.11	0.25	0.52	0.81	8.94	4.03	1.92	1.23
		¾	90	20	0.31	0.48	0.82	1.18	3.23	2.08	1.22	0.85
		1½	90	20	0.20	0.37	0.71	1.07	5.00	2.70	1.41	0.93
		4	90	20	0.13	0.30	0.64	1.01	7.82	3.35	1.56	0.99

Notes: [d] Spaces of uniform thickness, bounded by moderately smooth surfaces.

[e] Where a range of thickness is given, the given conductance is the average over the range; extreme values within the range differ therefrom by less than 10 percent.

[f] Interpolation, and moderate extrapolation, of conductance values are permissible for other values of mean temperature, temperature difference and effective emissivity E.

[g] Effective emissivity of space, E, is given by $\frac{1}{E} = \frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1$ where ϵ_1 and ϵ_2 are the emissivities of the surfaces of the air space. (See Section B, above.)

[h] The conductances of horizontal spaces with heat flow downward are substantially independent of temperature difference.

[e] Based on National Bureau of Standards data presented in *Housing Research Paper* No. 32, Housing and Home Finance Agency, 1954 (U. S. Government Printing Office, Washington, D. C.).

Table 4 .... Conductivities (k), Conductances (C), and Resistances (R) of Building and Insulating Materials (Design Values)[a]

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and, conductances (C) are for thickness or construction stated, not per inch thickness

Material	Description	Density (lb per Cu Ft)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance (R): Per inch thickness (1/k)	Resistance (R): For thickness listed (1/C)
AIR SPACES[b]	See Table 1—Section B					
	Position / Heat Flow / Thickness					
	Horizontal — Up (winter) — ¾–4 in.	—	—	1.18	—	0.85
	Horizontal — Up (summer) — ¾–4 in.	—	—	1.28	—	0.78
	Horizontal — Down (winter) — ¾ in.	—	—	0.98	—	1.02
	Horizontal — Down (winter) — 1½ in.	—	—	0.87	—	1.15
	Horizontal — Down (winter) — 4 in.	—	—	0.81	—	1.23
	Horizontal — Down (winter) — 8 in.	—	—	0.80	—	1.25
	Horizontal — Down (summer) — ¾ in.	—	—	1.18	—	0.85
	Horizontal — Down (summer) — 1½ in.	—	—	1.07	—	0.93
	Horizontal — Down (summer) — 4 in.	—	—	1.01	—	0.99
	Sloping, 45° — Up (winter) — ¾–4 in.	—	—	1.11	—	0.90
	Sloping, 45° — Down (summer) — ¾–4 in.	—	—	1.12	—	0.89
	Vertical — Horiz. (winter) — ¾–4 in.	—	—	1.03	—	0.97
	Vertical — Horiz. (summer) — ¾–4 in.	—	—	1.16	—	0.86
AIR SURFACES[c]	See Table 1—Section A					
STILL AIR	Position / Heat Flow					
	Horizontal — Up	—	—	1.63	—	0.61
	Up Sloping (45°) — Up	—	—	1.60	—	0.62
	Vertical — Horizontal	—	—	1.46	—	0.68
	Sloping (45°) — Down	—	—	1.32	—	0.76
	Horizontal — Down	—	—	1.08	—	0.92
15 MPH WIND	Any position—any direction (for winter)	—	—	6.00	—	0.17
7½ MPH WIND	Any position—any direction (for summer)	—	—	4.00	—	0.25
BUILDING BOARD[d]	Asbestos-cement board	120	4.0	—	0.25	—
BOARDS, PANELS, SHEATHING, ETC.	Asbestos-cement board — ⅛ in.			33.		0.03
	Gypsum or plaster board — ⅜ in.	50	—	3.10	—	0.32
	Gypsum or plaster board — ½ in.	50	—	2.25	—	0.45
	Plywood	34	0.80	—	1.25	—
	Plywood — ¼ in.	34	—	3.20	—	0.31
	Plywood — ⅜ in.	34	—	2.12	—	0.47
	Plywood — ½ in.	34	—	1.60	—	0.63
	Plywood or wood panels — ¾ in.	—	—	1.07	—	0.94
	Wood fiber board, laminated or homogeneous	26	0.42	—	2.38	—
		31	0.50	—	2.00	—
	Wood fiber—hardboard type	65	1.40	—	0.72	—
	Wood fiber—hardboard type — ¼ in.	65	—	5.60	—	0.18
	Wood—fir or pine sheathing — 25/32 in.	—	—	1.02	—	0.98
	Wood—fir or pine — 1 in.	—	—	0.49	—	2.03
BUILDING PAPER	Vapor—permeable felt	—	—	16.70	—	0.06
	Vapor—seal, 2 layers of mopped 15 lb felt	—	—	8.35	—	0.12
	Vapor—seal, plastic film	—	—	—	—	Negl
FLOORING MATERIALS	Asphalt tile — ⅛ in.	120	—	24.80	—	0.04
	Carpet and fibrous pad	—	—	0.48	—	2.08
	Carpet and rubber pad	—	—	0.81	—	1.23
	Ceramic tile — 1 in.	—	—	12.50	—	0.08
	Cork tile	25	0.45	—	2.22	—
	Cork tile — ⅛ in.	—	—	3.60	—	0.28
	Felt, flooring	—	—	16.70	—	0.06
	Floor tile or linoleum—av. value — ⅛ in.	—	—	20.00	—	0.05
	Linoleum — ⅛ in.	80	—	12.00	—	0.08
	Plywood subfloor — ⅜ in.	—	—	1.28	—	0.78
	Rubber or plastic tile — ⅛ in.	110	—	42.40	—	0.02
	Terrazzo — 1 in.	—	—	12.50	—	0.08
	Wood subfloor — 25/32 in.	—	—	1.02	—	0.98
	Wood, hardwood finish — ¾ in.	—	—	1.47	—	0.68
INSULATING MATERIALS	Cotton fiber[e]	0.8–2.0	0.26	—	3.85	—
BLANKET AND BATT	Mineral wool, fibrous form, processed from rock, slag, or glass[e]	1.5–4.0	0.27	—	3.70	—
	Wood fiber[e]	3.2–3.6	0.25	—	4.00	—
	Wood fiber, multilayer, stitched expanding[e]	1.5–2.0	0.27	—	3.70	—

Table 4 .... Conductivities (*k*), Conductances (C), and Resistances (*R*) of Building and Insulating Materials (Continued)
(*Design Values*)[a]

Material	Description	Density (Lb per Cu Ft)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance (R) Per inch thickness $(\frac{1}{k})$	Resistance (R) For thickness listed $(\frac{1}{C})$
BOARD	Glass fiber	9.5	0.25	—	4.00	—
	Wood or cane fiber					
	Acoustical tile[f] ½ in.	—	—	0.84	—	1.19
	Acoustical tile[f] ¾ in.	—	—	0.56	—	1.78
	Interior finish (plank, tile, lath)	15.0	0.35	—	2.86	—
	Interior finish (plank, tile, lath) ½ in.	15.0	—	0.70	—	1.43
	Roof deck slab					
	Approx. 1½ in.	—	—	0.24	—	4.17
	Approx. 2 in.	—	—	0.18	—	5.56
	Approx. 3 in.	—	—	0.12	—	8.33
	Sheathing (impreg. or coated)	20.0	0.38	—	2.63	—
	Sheathing (impreg. or coated) ½ in.	20.0	—	0.76	—	1.32
	Sheathing (impreg. or coated) 25/32 in.	20.0	—	0.49	—	2.06
BOARD AND SLABS	Cellular glass	9.0	0.40	—	2.50	—
	Corkboard (without added binder)	6.5–8.0	0.27	—	3.70	—
	Hog hair (with asphalt binder)	8.5	0.33	—	3.00	—
	Plastic (foamed)	1.62	0.29	—	3.45	—
	Wood shredded (cemented in preformed slabs)	22.0	0.55	—	1.82	—
LOOSE FILL	Macerated paper or pulp products	2.5–3.5	0.28	—	3.57	—
	Mineral wool (glass, slag, or rock)	2.0–6.0	0.30	—	3.33	—
	Sawdust or shavings	8.0–15.0	0.45	—	2.22	—
	Vermiculite (expanded)	7.0	0.48	—	2.08	—
	Wood fiber: redwood, hemlock, or fir	2.0–3.5	0.30	—	3.33	—
ROOF INSULATION	All types[e]					
	Preformed, for use above deck					
	Approx. ½ in.	—	—	0.72	—	1.39
	Approx. 1 in.	—	—	0.36	—	2.78
	Approx. 1½ in.	—	—	0.24	—	4.17
	Approx. 2 in.	—	—	0.19	—	5.26
	Approx. 2½ in.	—	—	0.15	—	6.67
	Approx. 3 in.	—	—	0.12	—	8.33
MASONRY MATERIALS CONCRETES	Cement mortar	116	5.0	—	0.20	—
	Gypsum-fiber concrete 87½% gypsum, 12½% wood chips	51	1.66	—	0.60	—
	Lightweight aggregates including expanded shale, clay or slate; expanded slags; cinders; pumice; perlite; vermiculite; also cellular concretes	120	5.2	—	0.19	—
		100	3.6	—	0.28	—
		80	2.5	—	0.40	—
		60	1.7	—	0.59	—
		40	1.15	—	0.86	—
		30	0.90	—	1.11	—
		20	0.70	—	1.43	—
	Sand and gravel or stone aggregate (oven dried)	140	9.0	—	0.11	—
	Sand and gravel or stone aggregate (not dried)	140	12.0	—	0.08	—
	Stucco	116	5.0	—	0.20	—
MASONRY UNITS	Brick, common	120	5.0	—	0.20	—
	Brick, face	130	9.0	—	0.11	—
	Clay tile, hollow:					
	1 cell deep 3 in.	—	—	1.25	—	0.80
	1 cell deep 4 in.	—	—	0.90	—	1.11
	2 cells deep 6 in.	—	—	0.66	—	1.52
	2 cells deep 8 in.	—	—	0.54	—	1.85
	2 cells deep 10 in.	—	—	0.45	—	2.22
	3 cells deep 12 in.	—	—	0.40	—	2.50
	Concrete blocks, three oval core:					
	Sand and gravel aggregate 4 in.	—	—	1.40	—	0.71
	8 in.	—	—	0.90	—	1.11
	12 in.	—	—	0.78	—	1.28
	Cinder aggregate 3 in.	—	—	1.16	—	0.86
	4 in.	—	—	0.90	—	1.11
	8 in.	—	—	0.58	—	1.72
	12 in.	—	—	0.53	—	1.89
	Gypsum partition tile:					
	3 x 12 x 30 in. solid	—	—	0.79	—	1.26
	3 x 12 x 30 in. 4-cell	—	—	0.74	—	1.35
	4 x 12 x 30 in. 3-cell	—	—	0.60	—	1.67

Table 4 .... Conductivities (*k*), Conductances (C), and Resistances (*R*) of Building and Insulating Materials (*Concluded*)
(*Design Values*)[a]

Material	Description	Density (Lb per Cu Ft)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance (R) Per inch thickness $(\frac{1}{k})$	Resistance (R) For thickness listed $(\frac{1}{C})$
MASONRY UNITS (*Continued*)	Lightweight aggregate (expanded shale, clay, slate or slag; pumice) 3 in.	—	—	0.79	—	1.27
	4 in.	—	—	0.67	—	1.50
	8 in.	—	—	0.50	—	2.00
	12 in.	—	—	0.44	—	2.27
	Stone, lime or sand	—	12.50	—	0.08	—
METALS	(See Chapter 5, Table 1)					
PLASTERING MATERIALS	Cement plaster, sand aggregate	116	5.0	—	0.20	—
	Sand aggregate ½ in.	—	—	10.00	—	0.10
	Sand aggregate ¾ in.	—	—	6.66	—	0.15
	Gypsum plaster:					
	Lightweight aggregate ½ in.	45	—	3.12	—	0.32
	Lightweight aggregate ⅝ in.	45	—	2.67	—	0.39
	Lightweight agg. on metal lath ¾ in.	—	—	2.13	—	0.47
	Perlite aggregate	45	1.5	—	0.67	—
	Sand aggregate	105	5.6	—	0.18	—
	Sand aggregate ½ in.	105	—	11.10	—	0.09
	Sand aggregate ⅝ in.	105	—	9.10	—	0.11
	Sand aggregate on metal lath ¾ in.	—	—	7.70	—	0.13
	Sand aggregate on wood lath	—	—	2.50	—	0.40
	Vermiculate aggregate	45	1.7	—	0.59	—
ROOFING	Asbestos-cement shingles	120	—	4.76	—	0.21
	Asphalt roll roofing	70	—	6.50	—	0.15
	Asphalt shingles	70	—	2.27	—	0.44
	Built-up roofing ⅜ in.	70	—	3.00	—	0.33
	Slate ½ in.	—	—	20.00	—	0.05
	Sheet metal	—	400+	—	Negl	
	Wood shingles	—	—	1.06	—	0.94
SIDING MATERIALS (ON FLAT SURFACE)	Shingles					
	Wood, 16-in. 7½-in. exposure	—	—	1.15	—	0.87
	Wood, double, 16-in., 12-in. exposure	—	—	0.84	—	1.19
	Wood, plus insul. backer board 5/16 in.	—	—	0.71	—	1.40
	Siding					
	Asbestos-cement, ¼ in., lapped	—	—	4.76	—	0.21
	Asphalt roll siding	—	—	6.50	—	0.15
	Asphalt insulating siding (½ in. bd.)	—	—	0.69	—	1.45
	Wood, drop, 1 x 8 in.	—	—	1.27	—	0.79
	Wood, bevel, ½ x 8 in., lapped	—	—	1.23	—	0.81
	Wood, bevel, ¾ x 10 in., lapped	—	—	0.95	—	1.05
	Wood, plywood, ⅜ in., lapped	—	—	1.69	—	0.59
	Structural glass	—	—	10.00	—	0.10
WOODS	Maple, oak, and similar hardwoods	45	1.10	—	0.91	—
	Fir, pine, and similar softwoods	32	0.80	—	1.25	—

[a] Representative values for dry materials at 75 F mean temperature, selected by the ASHAE Technical Advisory Committee on Insulation. They are intended as design (not specification) values for materials of building construction in normal use. For conductivity of a particular product, the user may obtain the value supplied by the manufacturer or secure the results of unbiased tests.

[b] Air space resistance values shown here are for spaces faced both sides with ordinary *nonreflective* materials ($\epsilon = 0.90$ and $E = 0.82$) and are based on following conditions: Winter—60 F mean temperature and 20 deg temperature difference, Summer—90 F mean temperature and 10 deg temperature difference except for horizontal air space with heat flow downward which is based on 20 deg temperature difference.

[c] Surface resistance values shown here are for ordinary *nonreflective* materials ($\epsilon = 0.90$).

[d] See also Insulating Materials, Board.

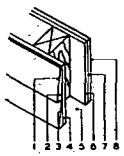
[e] Includes paper backing and facing if any. In cases where the insulation forms a boundary (highly reflective or otherwise) of an air space, refer to Table 3, Sections B and C, to obtain the insulating value of the air space for the appropriate effective emissivity and temperature conditions of the space.

[f] Insulating values of acoustical tile vary depending on density of the board and on the type, size, and depth of the perforations. An average conductivity *k* value is 0.42.

[g] The U. S. Department of Commerce, *Simplified Practice Recommendation for Thermal Conductance Factors for Preformed Above-Deck Roof Insulation*, No. R 257-55, recognizes the specification of roof insulation on the basis of the *C* values shown. Roof insulation is made in thicknesses to meet these values. Therefore, thickness supplied by different manufacturers may vary depending on the conductivity *k* value of the particular material.

Table 5 .... Coefficients of Transmission (U) of Frame Walls[a]

These coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph



Example—Wall D 4

Resistances used are given below in this table or in Table 3 or 4

Construction	Resistance (*R*)
1. Outside surface (15 mph wind)	0.17
2. Siding, wood, ½ in. x 8 in. lapped (av. *R*)	0.85
3. Building paper	0.06
4. Wood sheathing ($^{25}/_{32}$ in.)	0.93
5. Air space[b]	0.97
6. Gypsum lath (⅜ in.)	0.32
7. Plaster (sand agg.) (½ in)	0.09
8. Inside surface (still air)	0.68
Total resistance	4.12
$U = 1/R = 1/4.12 =$	0.24

See value 0.24 in boldface type in table below.

Example of Substitution

Resistances used are given below in this table or in Table 3 or 4

Replace items 3 and 4 with insul. bd. sheathing ($^{25}/_{32}$ in.) and items 6 and 7 with gypsum wall board (⅜ in.)		
Total resistance		4.12
Deduct 3. Building Paper	0.06	
4. Wood sheathing ($^{25}/_{32}$ in.)	0.93	
6. Gypsum lath (⅜ in.)	0.32	
7. Plaster (sand agg.) (½ in)	0.09	1.40
Difference		2.67
Add 4. Insul. bd. sheathing ($^{25}/_{32}$ in.)	2.06	
6. 7. Gypsum bd. (⅜ in.)	0.45	2.51
Total resistance		5.18
$U = 1/R = 1/5.18 =$		0.19

To Adjust U Values for Construction with Added Insulation between Framing Members, See Table 16.

Exterior[c]	R	Av. R	Interior Finish	R	Type of Sheathing[c]: None, Building Paper	Gypsum Board ½ in.	Plywood ⅜16 in.	Wood, $^{25}/_{32}$ in. and Building Paper	Insulation Board Sheathing: ½ in.	Insulation Board Sheathing: $^{25}/_{32}$ in.	Number
Resistance →					0.06	0.45	0.39	1.04	1.32	2.06	
Material			Material		U	U	U	U	U	U	
					A	B	C	D	E	F	
			None	—	0.57	0.47	0.48	0.36	0.33	0.27	1
			Gypsum bd. (⅜ in.)	0.32	0.33	0.29	0.30	0.25	0.23	0.20	2
			Gypsum lath (⅜ in.) and ½ in. plas. (lt. wt. agg.)	0.64	0.30	0.27	0.27	0.23	0.22	0.19	3
Wood siding			Gypsum lath (⅜ in.) and ½ in. plas. (sand agg.)	0.41	0.32	0.28	0.29	0.24	0.23	0.19	4
Drop—(1 in. x 8 in.)	0.79		Metal lath and ¾ in. plas. (lt. wt. agg.)	0.47	0.31	0.28	0.28	0.24	0.22	0.19	5
Bevel (½ in. x 8 in.)	0.81	0.85[d]	Metal lath and ¾ in. plas. (sand agg.)	0.15	0.35	0.31	0.31	0.26	0.24	0.21	6
Wood shingles 7½ in. exposure	0.87		Insul. bd. (½ in.)	1.45	0.24	0.22	0.22	0.19	0.18	0.16	7
Wood panels (¾ in.)	0.94		Insul. bd. lath (½ in.) and ½ in. plas. (sand agg.)	1.58	0.24	0.22	0.22	0.19	0.18	0.16	8
			Plywood (¼ in.)	0.31	0.33	0.29	0.30	0.25	0.23	0.20	9
			Wood panels (¾ in.)	0.94	0.27	0.25	0.25	0.22	0.20	0.18	10
			Wood lath and ½ in. plas. (sand agg.)	0.40	0.32	0.28	0.29	0.24	0.23	0.19	11
			None	—	0.73	0.56	0.58	0.42	0.38	0.30	12
			Gypsum bd. (⅜ in.)	0.32	0.37	0.33	0.33	0.27	0.25	0.21	13
			Gypsum lath (⅜ in.) and ½ in. plas. (lt. wt. agg.)	0.64	0.33	0.30	0.30	0.25	0.24	0.20	14
			Gypsum lath (⅜ in.) and ½ in. plas. (sand agg.)	0.41	0.36	0.32	0.32	0.27	0.25	0.21	15
Face-brick veneer[f]	0.44		Metal lath and ¾ in. plas. (lt. wt. agg.)	0.47	0.35	0.31	0.32	0.26	0.25	0.21	16
Plywood (⅜ in.)	0.47	0.45[d]	Metal lath and ¾ in. plas. (sand agg.)	0.15	0.40	0.35	0.36	0.29	0.27	0.22	17
			Insul. bd. (½ in.)	1.45	0.26	0.24	0.24	0.21	0.20	0.17	18
			Insul. bd. lath (½ in.) and ½ in. plas. (sand agg.)	1.58	0.26	0.23	0.24	0.21	0.19	0.17	19
			Plywood (¼ in.)	0.31	0.33	0.33	0.33	0.27	0.26	0.21	20
			Wood panels (¾ in.)	0.94	0.30	0.27	0.28	0.23	0.22	0.19	21
			Wood lath and ½ in. plas. (sand agg.)	0.40	0.36	0.32	0.32	0.27	0.25	0.21	22

Table 5 .... Coefficients of Transmission (U) of Frame Walls[a] (Concluded)

Exterior[c]	R	Av. R	Interior Finish	R	Type of Sheathing[c]: None, Building Paper	Gypsum Board ½ in.	Plywood ⅜16 in.	Wood, $^{25}/_{32}$ in. and Building Paper	Insulation Board Sheathing: ½ in.	Insulation Board Sheathing: $^{25}/_{32}$ in.	Number
Resistance →					0.06	0.45	0.39	1.04	1.32	2.06	
Material			Material		U	U	U	U	U	U	
					A	B	C	D	E	F	
			None	—	0.43	0.37	0.38	0.30	0.28	0.23	23
			Gypsum bd. (⅜ in.)	0.32	0.28	0.25	0.25	0.22	0.20	0.18	24
			Gypsum lath (⅜ in.) and ½ in. plas. (lt. wt. agg.)	0.64	0.25	0.23	0.23	0.20	0.19	0.17	25
			Gypsum lath (⅜ in.) and ½ in. plas. (sand agg.)	0.41	0.27	0.24	0.25	0.21	0.20	0.18	26
Wood shingles over insul.: backer bd. (⅜16 in.)	1.40	1.48[d]	Metal lath and ¾ in. plas. (lt. wt. agg.)	0.47	0.27	0.24	0.24	0.21	0.20	0.17	27
Asphalt insul. siding	1.46		Metal lath and ¾ in. plas. (sand agg.)	0.15	0.29	0.26	0.27	0.23	0.21	0.18	28
			Insul. bd. (½ in.)	1.45	0.21	0.20	0.20	0.18	0.17	0.15	29
			Insul. bd. lath (½ in.) and ½ in. plas. (sand agg.)	1.58	0.21	0.19	0.19	0.17	0.16	0.15	30
			Plywood (¼ in.)	0.31	0.28	0.25	0.25	0.22	0.20	0.18	31
			Wood panels (¾ in.)	0.94	0.24	0.22	0.22	0.19	0.18	0.16	32
			Wood lath and ½ in. plas. (sand agg.)	0.40	0.27	0.24	0.25	0.21	0.20	0.18	33
			None	—	0.91	0.67	0.70	0.48	0.42	0.32	34
			Gypsum bd. (⅜ in.)	0.32	0.42	0.36	0.37	0.30	0.27	0.23	35
			Gypsum lath (⅜ in.) and ½ in. plas. (lt. wt. agg.)	0.64	0.37	0.32	0.33	0.27	0.25	0.21	36
			Gypsum lath (⅜ in.) and ½ in. plas. (sand agg.)	0.41	0.40	0.35	0.36	0.29	0.27	0.22	37
Asbestos-cement siding	0.21		Metal lath and ¾ in. plas. (lt. wt. agg.)	0.47	0.39	0.34	0.35	0.28	0.26	0.22	38
Stucco[e] 1 in.	0.20	0.19[d]	Metal lath and ¾ in. plas. (sand agg.)	0.15	0.45	0.39	0.40	0.31	0.29	0.24	39
Asphalt roll siding	0.15		Insul. bd. (½ in.)	1.45	0.29	0.26	0.26	0.22	0.21	0.18	40
			Insul. bd. lath (½ in.) and ½ in. plas. (sand agg.)	1.58	0.28	0.25	0.26	0.22	0.21	0.18	41
			Plywood (¼ in.)	0.31	0.42	0.36	0.37	0.30	0.27	0.23	42
			Wood panels (¾ in.)	0.94	0.33	0.29	0.30	0.25	0.23	0.20	43
			Wood lath and ½ in. plas. (sand agg.)	0.40	0.40	0.35	0.36	0.29	0.27	0.22	44

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] To adjust *U* values for the effect of added insulation between framing members, see Table 16.
[c] Note that although several types of exterior finish may be grouped because they have approximately the same thermal resistance value, it is not implied that all types may be suitable for application over all types of sheathing listed.
[d] Average resistance of items listed. This average was used in computation of *U* values shown.
[e] Building paper is not included except where noted.
[f] Small air space between building paper and brick veneer neglected.
[g] Where stucco is applied over insulating board or gypsum sheathing, building paper is generally required, but the change in *U* value is negligible.

Solution: In Fig. 5 the following paths of heat flow from plane *A* to plane *F* will be noted:

1. From *A* to *B*: One path through 3 in. of concrete.
2. From *B* to *C*: Two paths, (*a*) through 1 in. of tie rod, and (*b*) through 1 in. of concrete.
3. From *C* to *D*: Two paths, (*a*) through 2½ in. of tie rod, and (*b*) through 2½ in. of insulation.
4. From *D* to *E*: Two paths, (*a*) through 1 in. of tie rod, and (*b*) through 1 in. of concrete.
5. From *E* to *F*: One path through 3 in. of concrete.

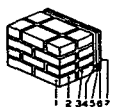
It will be noted that items 2 and 4 are paths of similar flow, and could be treated as one. If equilibrium or steady-state heat transfer is assumed, there will exist a temperature difference between the metal tie rod and the concrete, and also between the metal tie rod and the insulating material. The rate of heat transfer between these materials is dependent upon their conductivity values and the temperature difference. As the conductivity of the metal tie rods is considerably higher than that of the concrete or insulating material, it cannot be assumed that the same rate of heat transfer takes place for all parallel paths. Likewise, an appreciable error would be made by assuming that no heat transfer takes place between the metal tie rod and the surrounding materials. Although the pattern of the isotherms is unknown, the following method of calculation does partially take into account the heat flow between the metal tie rods and its bounding materials.

Parallel Flow. The conductances through the areas of parallel heat flow may be determined as follows:

1. The area of each ¼-in. diameter tie rod is 0.00034 sq ft, and as the tie rods are spaced 9 in. vertically, and 12 in. hori-

Table 6 .... Coefficients of Transmission (*U*) of Solid Masonry Walls[a]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph



Example—Wall G 2

Resistances used are given below in this table or in Table 3 or 4

Construction	Resistance (*R*)
1. Outside surface (15 mph wind)	*0.17*
2. Face brick (4 in.)	*0.44*
3. Common brick (4 in.)	*0.80*
4. Air space[b]	*0.97*
5. Gypsum lath (⅜ in.)	*0.32*
6. Plas. (sand agg.) (½ in.)	*0.09*
7. Inside surface (still air)	*0.68*
Total resistance	*3.47*
$U = 1/R = 1/3.47 =$	0.29

See value 0.29 in boldface type in table below.

Example of Substitution

Resistances used are given below in this table or in Table 3 or 4

Assume plain wall—no furring or plaster.		
Total resistance		*3.47*
Deduct 4. Air space	*0.97*	
5. Gypsum lath (⅜ in.)	*0.32*	
6. Plas. (sand agg.) (½ in.)	*0.09*	*1.38*
Total resistance		*2.09*
$U = 1/R = 1/2.09 =$		0.48

To Adjust U Values for Construction with Added Insulation between Furring Strips, See Table 16

Exterior Construction[c]		*Interior Finish*												
		None	Plas. ⅜ in. on Wall		Metal Lath and ¾ in. Plas. on Furring		Gypsum Lath (⅜ in.) and ½ in. Plas. on Furring			Insul. Bd. Lath (½ in.) and ½ in. Plas. on Furring			Wood Lath and ½ in. Plas.	Number
			(Sand agg.)	(Lt. wt. agg.)	(Sand agg.)	(Lt. wt. agg.)	No plas.	(Sand agg.)	(Lt. wt. agg.)	No plas.	(Sand agg.)	(Lt. wt. agg.)	(Sand agg.)	
Resistance →			*0.11*	*0.39*	*0.13*	*0.47*	*0.32*	*0.41*	*0.64*	*1.43*	*1.52*	*0.40*		
Material	***R***	*U*	*U*	*U*	*U*	*U*	*U*	*U*	*U*	*U*	*U*	*U*		
		A	B	C	D	E	F	G	H	I	J	K		
Brick (face and common)[d]														
(6 in.)	*0.61*	0.68	0.64	0.54	0.39	0.34	0.36	0.35	0.33	0.26	0.25	0.35		1
(8 in.)	*1.24*	0.48	0.45	0.41	0.31	0.28	0.30	0.29	0.27	0.22	0.22	0.29		2
(12 in.)	*2.04*	0.35	0.33	0.30	0.25	0.23	0.24	0.23	0.22	0.19	0.19	0.23		3
(16 in.)	*2.84*	0.27	0.26	0.25	0.21	0.19	0.20	0.20	0.19	0.16	0.16	0.20		4
Brick (common only)														
(8 in.)	*1.60*	0.41	0.39	0.35	0.28	0.26	0.27	0.26	0.25	0.21	0.20	0.26		5
(12 in.)	*2.40*	0.31	0.30	0.27	0.23	0.21	0.22	0.22	0.21	0.18	0.17	0.22		6
(16 in.)	*3.20*	0.25	0.24	0.23	0.19	0.18	0.19	0.18	0.18	0.16	0.15	0.18		7
Stone (lime and sand)														
(8 in.)	*0.64*	0.67	0.63	0.53	0.39	0.34	0.36	0.35	0.32	0.26	0.25	0.35		8
(12 in.)	*0.96*	0.55	0.52	0.45	0.34	0.31	0.32	0.31	0.29	0.24	0.23	0.31		9
(16 in.)	*1.28*	0.47	0.45	0.40	0.31	0.28	0.29	0.28	0.27	0.22	0.22	0.29		10
(24) in.	*1.92*	0.36	0.35	0.32	0.26	0.24	0.25	0.24	0.23	0.19	0.19	0.24		11
Hollow clay tile														
(8 in.)	*1.85*	0.36	0.36	0.32	0.26	0.24	0.25	0.25	0.23	0.20	0.19	0.25		12
(10 in.)	*2.22*	0.33	0.31	0.29	0.24	0.22	0.23	0.22	0.21	0.18	0.18	0.23		13
(12 in.)	*2.50*	0.30	0.29	0.27	0.22	0.21	0.22	0.21	0.20	0.17	0.17	0.21		14
Poured concrete														
30 lb per cut ft														
(4 in.)	*4.44*	0.19	0.19	0.18	0.16	0.15	0.15	0.15	0.14	0.13	0.13	0.15		15
(6 in.)	*6.66*	0.13	0.13	0.13	0.12	0.11	0.11	0.11	0.11	0.10	0.10	0.11		16
(8 in.)	*8.88*	0.10	0.10	0.10	0.09	0.09	0.09	0.09	0.09	0.08	0.08	0.09		17
(10 in.)	*11.10*	0.08	0.08	0.08	0.08	0.07	0.08	0.08	0.07	0.07	0.07	0.08		18
80 lb per cut ft														
(6 in.)	*2.40*	0.31	0.30	0.27	0.23	0.21	0.22	0.22	0.21	0.18	0.17	0.22		19
(8 in.)	*3.20*	0.25	0.24	0.23	0.19	0.18	0.19	0.18	0.18	0.16	0.15	0.18		20
(10 in.)	*4.00*	0.21	0.20	0.19	0.17	0.16	0.16	0.16	0.15	0.14	0.14	0.16		21
(12 in.)	*4.80*	0.18	0.17	0.17	0.15	0.14	0.14	0.14	0.14	0.12	0.12	0.14		22
140 lb per cut ft														
(6 in.)	*0.48*	0.75	0.69	0.58	0.41	0.36	0.38	0.37	0.34	0.27	0.26	0.37		23
(8 in.)	*0.64*	0.67	0.63	0.53	0.39	0.34	0.36	0.35	0.32	0.26	0.25	0.35		24
(10 in.)	*0.80*	0.61	0.57	0.49	0.36	0.32	0.34	0.33	0.31	0.25	0.24	0.33		25
(12 in.)	*0.96*	0.55	0.52	0.45	0.34	0.31	0.32	0.31	0.29	0.24	0.23	0.31		26

Table 6 .... Coefficients of Transmission (*U*) of Solid Masonry Walls (*Concluded*)

Exterior Construction[c]		*Interior Finish*												
		None	Plas. ⅜ in. on Wall		Metal Lath and ¾ in. Plas. on Furring		Gypsum Lath (⅜ in.) and ½ in. Plas. on Furring			Insul. Bd. Lath (½ in.) and ½ in. Plas. on Furring			Wood Lath and ½ in. Plas.	Number
			(Sand agg.)	(Lt. wt. agg.)	(Sand agg.)	(Lt. wt. agg.)	No plas.	(Sand agg.)	(Lt. wt. agg.)	No plas.	(Sand agg.)	(Lt. wt. agg.)	(Sand agg.)	
Resistance →			*0.11*	*0.39*	*0.13*	*0.47*	*0.32*	*0.41*	*0.64*	*1.43*	*1.52*	*0.40*		
Material	***R***	*U*	*U*	*U*	*U*	*U*	*U*	*U*	*U*	*U*	*U*	*U*		
		A	B	C	D	E	F	G	H	I	J	K		
Concrete block (gravel agg.)														
(8 in.)	*1.11*	0.52	0.48	0.43	0.33	0.29	0.31	0.30	0.28	0.23	0.22	0.30		27
(12 in.)	*1.28*	0.47	0.45	0.40	0.31	0.28	0.29	0.28	0.27	0.22	0.22	0.29		28
(Cinder agg.)														
(8 in.)	*1.72*	0.39	0.37	0.34	0.27	0.25	0.26	0.25	0.24	0.20	0.20	0.25		29
(12 in.)	*1.89*	0.36	0.35	0.32	0.26	0.24	0.25	0.24	0.23	0.19	0.19	0.24		30
(Lt. wt. agg.)														
(8 in.)	*2.00*	0.35	0.34	0.31	0.26	0.23	0.24	0.24	0.22	0.19	0.19	0.24		31
(12 in.)	*2.27*	0.32	0.31	0.28	0.24	0.22	0.23	0.22	0.21	0.18	0.18	0.22		32

[a] See text section Calculating Overall Coefficients for basis of calculations.

[b] To adjust *U* values for the effect of added insulation between framing members, see Table 16.

[c] If stucco or structural glass is applied to the exterior, the additional resistance value of 0.10 would have a negligible effect on the *U* value.

[d] Brick, 8 in. (7⅝ in. actual) is assumed to have no backing. Walls 8, 12 and 16 in. have 4 in. of face brick and balance of common brick.

zontally, there will be $0.00034 \times \frac{1}{3} = 0.00045$ sq ft of tie rod to each square foot of wall area. Then from plane B to plane C, the conductance C_1 is

$$C_1 = \frac{0.00045}{1.0} \times \frac{400}{1.0} + \frac{0.99955}{1.0} \times \frac{12}{1.0}$$

$$= 0.180 + 11.994 = 12.174$$

2. For tie rod and insulation from plane C to plane D the conductance C_2 is

$$C_2 = \frac{0.00045}{1.0} \times \frac{400}{2.5} + \frac{0.99955}{1.0} \times \frac{0.30}{2.5}$$

$$= 0.072 + 0.120 = 0.192$$

3. For tie rod and concrete from plane D to plane E the conductance C_3 is

$$C_3 = \frac{0.00045}{1.0} \times \frac{400}{1.0} + \frac{0.99955}{1.0} \times \frac{12}{1.0}$$

$$= 0.180 + 11.994 = 12.174$$

Series Flow. After the conductance values have been determined, the total resistance and U value can be determined as follows;

$$R_T = \frac{1}{f_i} + \frac{z_1}{k_1} + \frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3} + \frac{z_2}{k_2} + \frac{1}{f_o}$$

$$R_T = \frac{1}{1.65} + \frac{3.0}{12.0} + \frac{1}{12.174} + \frac{1}{0.192} + \frac{1}{12.174} + \frac{3.0}{12.0} + \frac{1}{6.0}$$

$$R_T = 0.606 + 0.250 + 0.0822 + 5.208$$

$$+ 0.0822 + 0.250 + 0.167 = 6.727$$

$$U = \frac{1}{R_T} = \frac{1}{6.727} = 0.149 \text{ Btu per (hr) (sq ft) (F deg).}$$

The hot box test value, from University of Minnesota, for this wall, corrected for a 15 mph wind velocity, was $U = 0.150$ Btu per (hr) (sq ft) (F deg). The error between the calculated and test values would be

$$\frac{0.150 - 0.149}{0.150} \times 100 = 0.67 \text{ percent.}$$

If the effect of the tie rods were omitted from the calculations, the overall U value would be 0.103. Although the percentage of area occupied by the tie rods per square foot of wall area is $0.00045/1.0 \times 100 = 0.045$ percent, the error between the calculated and test values would be

$$\frac{0.150 - 0.103}{0.150} \times 100 = 31 \text{ percent.}$$

Values Used in Calculation of *U* Value Tables

Tables 5 through 15 are based on values given in Table 4. The following conditions have been used:

Equilibrium or steady-state heat transfer, eliminating effects of heat capacity.

Surrounding surfaces at ambient air temperatures.

Exterior wind velocity of 15 mph for winter and 7.5 mph for summer.

Surface emissivity of ordinary building materials $\epsilon = 0.90$.

Stud space in frame construction not insulated. (See Table 16 for method of correcting for added insulation.)

In construction involving air spaces the U values shown are calculated for areas between framing. See Fig. 6 in section Correction for Framing if an allowance is to be made for this effect.

Air space resistance values used are those shown in Table 4 under Air Spaces.

Air spaces are ¾ in. or more in width.

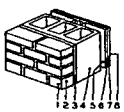
Variations of conductivity with mean temperature is neglected.

Corrections for framing to be made on basis of parallel heat

(*Continued on p. 108.*)

Table 7 .... Coefficients of Transmission (U) of Masonry Walls[a]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph



Example—Wall G 1

Resistances used are given below in this table or in Table 3 or 4

Construction	Resistance (R)
1. Outside surface (15 mph wind)	0.17
2. Face brick (4 in.) (av. R.)	0.39
3. Cement mortar (½ in)	0.10
4. Concrete block (cinder agg.) (4 in.)	1.11
5. Air space[b]	0.97
6. Gypsum lath (⅜ in.)	0.32
7. Plas. (sand agg.) (½ in)	0.09
8. Inside surface (still air)	0.68
Total resistance	3.85
$U = 1/R = 1/3.85 =$	0.26

See value 0.26 in boldface type in table below.

Example of Substitution

Resistances used are given below in this table or in Table 3 or 4

Replace items 6 and 7 with wood panels (¾ in.) and vapor barrier applied over furring strips		
Total resistance		3.85
Deduct 6. Gypsum lath (⅜ in.)	0.32	
7. Plas. (sand agg.) (½ in)	0.09	0.41
Difference		3.44
Add 6. Vapor barrier	0.06	
7. Wood panel (¾ in.)	0.94	1.00
Total resistance		4.44
$U = 1/R = 1/4.44 =$		0.23

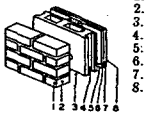
To Adjust U Values for Construction with Added Insulation between Furring Strips, See Table 16

Exterior Facing			Backing		Interior Finish: None	Plas. ⅝ in. on Wall (Sand agg.)	Plas. ⅝ in. on Wall (Lt. wt. agg.)	Metal Lath and ¾ in. Plas. on Furring (Sand agg.)	Metal Lath and ¾ in. Plas. on Furring (Lt. wt. agg.)	Gypsum Lath (⅜ in.) and ½ in. Plas. on Furring (No plas.)	Gypsum Lath (⅜ in.) and ½ in. Plas. on Furring (Sand agg.)	Gypsum Lath (⅜ in.) and ½ in. Plas. on Furring (Lt. wt. agg.)	Insul. Bd. Lath (½ in.) and ½ in. Plas. on Furring (No plas.)	Insul. Bd. Lath (½ in.) and ½ in. Plas. on Furring (Sand agg.)	Wood Lath and ½ in. Plas. (Sand agg.)	Number
			Resistance →		0	0.11	0.39	0.13	0.47	0.32	0.41	0.64	1.43	1.52	0.40	
					U	U	U	U	U	U	U	U	U	U	U	
Material	R	Av. R	Material	R	A	B	C	D	E	F	G	H	I	J	K	
			Concrete block (Cinder agg.)													
			(4 in.)	1.11	0.41	0.39	0.35	0.28	0.26	0.27	0.26	0.25	0.21	0.20	0.26	1
			(8 in.)	1.72	0.33	0.32	0.29	0.24	0.22	0.23	0.23	0.21	0.18	0.18	0.23	2
			(12 in.)	1.89	0.31	0.30	0.28	0.23	0.21	0.22	0.22	0.21	0.18	0.17	0.22	3
			(Lt. wt. agg.)													
			(4 in.)	1.50	0.35	0.34	0.31	0.25	0.23	0.24	0.24	0.22	0.19	0.19	0.24	4
Face brick			(8 in.)	2.00	0.30	0.29	0.27	0.23	0.21	0.22	0.21	0.20	0.17	0.17	0.21	5
4 in.	0.44		(12 in.)	2.27	0.28	0.27	0.25	0.21	0.20	0.20	0.20	0.19	0.17	0.16	0.20	6
Stone			(Sand agg.)													
4 in.	0.32		(4 in.)	0.71	0.49	0.46	0.41	0.32	0.29	0.30	0.29	0.27	0.22	0.22	0.29	7
Precast concrete (sand agg.)		0.39	(8 in.)	1.11	0.41	0.39	0.35	0.28	0.26	0.27	0.26	0.25	0.21	0.20	0.26	8
4 in.	0.32		(12 in.)	1.28	0.38	0.37	0.33	0.27	0.25	0.26	0.25	0.24	0.20	0.20	0.25	9
6 in.	0.48		Hollow clay tile													
			(4 in.)	1.11	0.41	0.39	0.35	0.28	0.26	0.27	0.26	0.25	0.21	0.20	0.26	10
			(8 in.)	1.85	0.31	0.30	0.28	0.23	0.22	0.22	0.22	0.21	0.18	0.18	0.22	11
			(12 in.)	2.50	0.26	0.25	0.24	0.20	0.19	0.19	0.19	0.18	0.16	0.16	0.19	12
			Concrete (Sand agg.)													
			(4 in.)	0.32	0.60	0.56	0.49	0.36	0.32	0.34	0.33	0.31	0.25	0.24	0.33	13
			(6 in.)	0.48	0.55	0.52	0.45	0.34	0.31	0.32	0.31	0.29	0.24	0.23	0.31	14
			(8 in.)	0.64	0.51	0.48	0.42	0.32	0.29	0.31	0.30	0.28	0.23	0.22	0.30	15
			Concrete block (Cinder agg.)													
			(4 in.)	1.11	0.36	0.35	0.32	0.26	0.24	0.25	0.24	0.23	0.19	0.19	0.24	16
			(8 in.)	1.72	0.29	0.29	0.26	0.22	0.21	0.21	0.21	0.20	0.17	0.17	0.21	17
			(12 in.)	1.89	0.28	0.27	0.25	0.21	0.20	0.21	0.20	0.19	0.17	0.17	0.20	18
			(Lt. wt. agg.)													
			(4 in.)	1.50	0.32	0.30	0.28	0.23	0.22	0.22	0.22	0.21	0.18	0.18	0.22	19
			(8 in.)	2.00	0.27	0.26	0.25	0.21	0.20	0.20	0.20	0.19	0.16	0.16	0.20	20
			(12 in.)	2.27	0.25	0.25	0.23	0.20	0.19	0.19	0.19	0.18	0.16	0.16	0.19	21
Common brick			(Sand agg.)													
4 in.	0.80		(4 in.)	0.71	0.42	0.40	0.36	0.29	0.26	0.27	0.27	0.25	0.21	0.21	0.27	22
Precast concrete (sand agg.)		0.72	(8 in.)	1.11	0.36	0.35	0.32	0.26	0.24	0.25	0.24	0.23	0.19	0.19	0.24	23
8 in.	0.64		(12 in.)	1.28	0.34	0.33	0.30	0.25	0.23	0.24	0.23	0.22	0.19	0.18	0.23	24
			Hollow clay tile													
			(4 in.)	1.11	0.36	0.35	0.32	0.26	0.24	0.25	0.24	0.23	0.19	0.19	0.24	25
			(8 in.)	1.85	0.28	0.28	0.26	0.22	0.20	0.21	0.20	0.19	0.17	0.17	0.20	26
			(12 in.)	2.50	0.24	0.23	0.22	0.19	0.18	0.18	0.18	0.17	0.15	0.15	0.18	27
			Concrete (Sand agg.)													
			(4 in.)	0.32	0.50	0.48	0.42	0.32	0.29	0.30	0.30	0.28	0.23	0.22	0.30	28
			(6 in.)	0.48	0.47	0.44	0.39	0.31	0.28	0.29	0.28	0.27	0.22	0.22	0.28	29
			(8 in.)	0.64	0.43	0.41	0.37	0.29	0.27	0.28	0.27	0.26	0.21	0.21	0.27	30

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] To adjust U values for the effect of added insulation between framing members, see Table 16.

Table 8 .... Coefficients of Transmission (U) of Masonry Cavity Walls[a]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph



Example—Wall H 6

Resistances used are given below in this table or in Table 3 or 4

Construction	Resistance (R)
1. Outside surface (15 mph wind)	0.17
2. Common brick (4 in.) (av. R.)	0.76
3. Air space[b]	0.97
4. Concrete block (gravel agg.) (4 in.)	0.71
5. Air space[b]	0.97
6. Gypsum lath (⅜ in.)	0.32
7. Plas. (lt. wt. agg.) (½ in.)	0.52
8. Inside surface (still air)	0.68
Total resistance	4.90
$U = 1/R = 1/4.90 =$	0.20

See value 0.20 in boldface type in table below.

Example of Substitution

Resistances used are given below in this table or in Table 3 or 4

Replace item 4 with 8 in. concrete block and items 6 and 7 with ⅝ in. plas. (sand agg.) applied directly to concrete block.		
Total resistance		4.90
Deduct 4. Concrete block (gravel agg.) 4 in.	0.71	
5. Air space	0.97	
6. Gypsum lath (⅜ in.)	0.32	
7. Plas. (lt. wt. agg.) (½ in.)	0.32	2.32
Difference		2.58
Add 4. Concrete block (gravel agg.) 8 in.	1.11	
7. Plas. (sand agg.) (⅝ in.)	0.11	1.22
Total resistance =		3.80
$U = 1/R = 1/3.80$		0.26

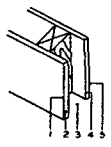
To Adjust U Values for Construction with Added Insulation[c] between Inner and Outer Tiers or between Furring Strips, See Table 16

Exterior Construction			Inner Section		Interior Finish: None	Plas. ⅝ in. on Wall (Sand agg.)	Plas. ⅝ in. on Wall (Lt. wt. agg.)	Metal Lath and ¾ in. Plas. on Furring (Sand agg.)	Metal Lath and ¾ in. Plas. on Furring (Lt. wt. agg.)	Gypsum Lath (⅜ in.) and ½ in. Plas. on Furring (No plas.)	Gypsum Lath (⅜ in.) and ½ in. Plas. on Furring (Sand agg.)	Gypsum Lath (⅜ in.) and ½ in. Plas. on Furring (Lt. wt. agg.)	Insul. Bd. Lath and ½ in. Plas. on Furring (No plas.)	Insul. Bd. Lath and ½ in. Plas. on Furring (Sand agg.)	Wood Lath and ½ in. Plas. (Sand agg.)	Number
			Resistance →		0	0.11	0.39	0.13	0.47	0.32	0.41	0.64	1.43	1.52	0.40	
					U	U	U	U	U	U	U	U	U	U	U	
Material	R	Av. R	Material	R	A	B	C	D	E	F	G	H	I	J	K	
			Concrete block (4 in.)													
			(Gravel agg.)	0.71	0.34	0.32	0.30	0.25	0.23	0.23	0.23	0.22	0.19	0.18	0.23	1
			(Cinder agg.)	1.11	0.30	0.29	0.27	0.22	0.21	0.21	0.21	0.20	0.17	0.17	0.21	2
Face brick (4 in.)	0.44		(Lt. wt. agg.)	1.60	0.27	0.26	0.24	0.21	0.19	0.20	0.19	0.19	0.16	0.16	0.19	3
			Common brick (4 in.)	0.80	0.33	0.32	0.29	0.24	0.22	0.23	0.23	0.21	0.18	0.18	0.23	4
			Clay tile (4 in.)	1.11	0.30	0.29	0.27	0.22	0.21	0.21	0.21	0.20	0.17	0.17	0.21	5
			Concrete block (4 in.)													
Common brick (4 in.)	0.80		(Gravel agg.)	0.71	0.30	0.29	0.27	0.23	0.21	0.22	0.21	0.20	0.18	0.17	0.21	6
Concrete block (gravel agg.)		0.76	(Cinder agg.)	1.11	0.27	0.26	0.25	0.21	0.19	0.20	0.20	0.19	0.16	0.16	0.20	7
(4 in.)	0.71		(Lt. wt. agg.)	1.60	0.25	0.24	0.22	0.19	0.18	0.19	0.18	0.18	0.15	0.15	0.18	8
			Common brick (4 in.)	0.80	0.30	0.29	0.27	0.22	0.21	0.21	0.21	0.20	0.17	0.17	0.21	9
			Clay tile (4 in.)	1.11	0.27	0.26	0.25	0.21	0.19	0.20	0.20	0.19	0.16	0.16	0.20	10
			Concrete block (4 in.)													
			(Gravel agg.)	0.71	0.27	0.27	0.25	0.21	0.20	0.20	0.20	0.19	0.17	0.16	0.20	11
Concrete block (cinder agg.)			(Cinder agg.)	1.11	0.25	0.24	0.23	0.19	0.18	0.19	0.18	0.18	0.16	0.15	0.18	12
(4 in.)	1.11		(Lt. wt. agg.)	1.60	0.23	0.22	0.21	0.18	0.17	0.17	0.17	0.17	0.15	0.14	0.17	13
			Common brick (4 in.)	0.80	0.27	0.26	0.24	0.21	0.19	0.20	0.20	0.19	0.16	0.16	0.20	14
			Clay tile (4 in.)	1.11	0.25	0.24	0.23	0.19	0.18	0.19	0.18	0.18	0.16	0.15	0.18	15

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] To adjust U values for the effect of added insulation between framing members, see Table 16.
[c] If insulation is to be used in the cavity it should be a water resistant type.

Table 9 .... Coefficients of Transmission (*U*) of Frame Partitions or Interior Walls[a]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides



Example—Wall B 1		*Example of Substitution*	
Resistances used are given below in this table or in Table 3 or 4		*Resistances used are given below in this table or in Table 3 or 4*	
Construction	Resistance (*R*)	Replace item 2 with wood fiber hardboard (¼ in.).	
1. Surface (still air)	*0.68*	Total resistance	*2.97*
2. Gypsum bd. (⅜ in.)	*0.32*	Deduct 2. Gypsum wall board (⅜ in.)	*0.32*
3. Air space[b]	*0.97*		
4. Gypsum wall board (⅜ in.)	*0.32*	Difference	*2.65*
5. Surface (still air)	*0.68*	Add 2. Hardboard (¼ in.)	*0.18*
Total resistance	*2.97*	Total resistance	*2.83*
$U = 1/R = 1/2.97$	0.34	$U = 1/R = 1/2.83 =$	0.35
See value 0.34 in boldface type in table below.			

To Adjust U Values for Construction with Added Insulation between Members, See Table 16

Type of Interior Finish		*Single Partition (Finish on Only One Side of Studs)*	*Double Partition (Finish on Both Sides of Studs)*	
Material	*R*	*U*	*U*	Number
		A	B	
Gypsum bd. (⅜ in.)	*0.32*	0.60	0.34	1
Gypsum lath (⅜ in.) and ½ in. plas. (lt. wt. agg.)	*0.64*	0.50	0.28	2
Gypsum lath (⅜ in.) and ½ in. plas. (sand agg.)	*0.41*	0.56	0.32	3
Metal lath and ¾ in. plas. (lt. wt. agg.)	*0.47*	0.55	0.31	4
Metal lath and ¾ in. plas. (sand agg.)	*0.13*	0.67	0.39	5
Insul. bd. (½ in.)	*1.43*	0.36	0.19	6
Insul. bd. lath (½ in.) and ½ plas. (sand agg.)	*1.52*	0.35	0.19	7
Plywood: (¼ in.)	*0.31*	0.60	0.34	8
(⅜ in.)	*0.47*	0.55	0.31	9
(½ in.)	*0.63*	0.50	0.28	10
Wood panels (¾ in.)	*0.94*	0.43	0.24	11
Wood-lath and ½ in. plas. (sand agg.)	*0.40*	0.57	0.32	12
Sheet-metal panels	*0*	0.74	0.43	13
Glass and glass blocks		See Table 21		

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] To adjust *U* values for the effect of added insulation between framing members, see Table 16.

flow through 2 x 4 in. (nominal) studs, 16 in. on centers, the framing covering 15 percent of wall area, as shown in Fig. 6.

Thermal resistance for gypsum board and insulating board apply equally to plain material and to those which may be decorated at the factory or on the job.

In order to condense the tables an average resistance value (Av.*R*) has been used in some tables for types of materials having approximately the same thermal resistance values. The difference between the average value and the exact value for any given material usually causes no significant change in the resulting *U* value.

Actual thicknesses of lumber assumed to be as follows:

Nominal	*Actual*	*Nominal*	*Actual*
1 in. (S-2-S)	²⅝₃₂ in.	3 in. (S-2-S)	2⅝ in.
1½ in. (S-2-S)	1⁵⁄₁₆ in.	4 in. (S-2-S)	3⅝ in.
2 in. (S-2-S)	1⅝ in.	Finish flooring,	
2½ in. (S-2-S)	2⅛ in.	(maple or oak)	¾ in.

It should be noted that the effects of poor workmanship in construction and installation have an increasingly greater percentage effect on heat transmission as the coefficient becomes numerically smaller. Failure to meet design estimates may be caused by lack of proper attention to exact compliance with specifications. A factor of safety may be employed as a precaution when it is judged desirable.

Insulating Materials

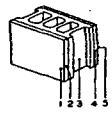
In order to determine the benefit derived from the addition of insulating materials to a given construction, the overall coefficient of heat transmission U_i of the insulated construction may be compared with the corresponding coefficient *U* without insulation. Table 16 (Part A to Part E) and Fig. 6 may be used for determining the coefficients of transmission of constructions with any combination of fibrous, bulk, or mass insulation and air spaces of various effective emissivities.

Example 2: Find the coefficient of transmission of a frame wall consisting of ½ x 8-in. bevel lap wood siding, building pa-

(Continued on p. 111.)

Table 10 .... Coefficients of Transmission (*U*) of Masonry Partitions[a]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides



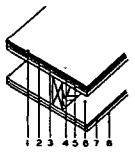
Example—Wall C 2		*Example of Substitution*	
Resistances used are given below in this table or in Table 3 or 4		*Resistances used are given below in this table or in Table 3 or 4*	
Construction	Resistance (*R*)	Replace item 3 with gypsum tile (4 in.)	
1. Inside surface (still air)	*0.68*	Total resistance	*3.25*
2. Plas. (lt. wt. agg.) ⅝ in.	*0.39*	Deduct 3. Cement block (cinder agg.) (4 in.)	*1.11*
3. Cement block (cinder agg.) (4 in.)	*1.11*		
4. Plas. (lt. wt. agg.) ⅝ in.	*0.39*	Difference	*2.14*
5. Inside surface (still air)	*0.68*	Add 3. Gypsum tile (4 in.)	*1.67*
Total resistance	*3.25*	Total resistance	*3.81*
$U = 1/R = 1/3.25 =$	0.31	$U = 1/R = 1/3.81 =$	0.26
See value 0.31 in boldface type in table below.			

Type of Partition		*Surface Finish*					
		None	Plas. (lt. wt. agg.) ⅝ in.		Plas. (sand agg.) ⅝ in.		
	Resistance ↓		One side 0.39	Two sides 0.78	One side 0.11	Two sides 0.22	Number
Material	*R*	*U*	*U*	*U*	*U*	*U*	
		A	B	C	D	E	
Hollow concrete block (Cinder agg.)							
(3 in.)	*0.86*	0.45	0.38	0.33	0.43	0.41	1
(4 in.)	*1.11*	0.40	0.35	0.31	0.39	0.37	2
(8 in.)	*1.72*	0.32	0.29	0.26	0.31	0.30	3
(12 in.)	*1.89*	0.31	0.27	0.25	0.30	0.29	4
(Lt. wt. agg.)							
(3 in.)	*1.27*	0.38	0.33	0.30	0.36	0.35	5
(4 in.)	*1.60*	0.35	0.31	0.27	0.34	0.32	6
(8 in.)	*2.00*	0.30	0.27	0.24	0.29	0.28	7
(12 in.)	*2.27*	0.28	0.25	0.23	0.27	0.26	8
(Gravel agg.)							
(8 in.)	*1.11*	0.40	0.35	0.31	0.39	0.37	9
(12 in.)	*1.28*	0.38	0.33	0.29	0.36	0.35	10
Hollow clay tile							
(3 in.)	*0.80*	0.46	0.39	0.34	0.44	0.42	11
(4 in.)	*1.11*	0.40	0.35	0.31	0.39	0.37	12
(6 in.)	*1.52*	0.35	0.31	0.27	0.33	0.32	13
(8 in.)	*1.85*	0.31	0.28	0.25	0.30	0.29	14
Hollow gypsum tile							
(3 in.)	*1.36*	0.37	0.32	0.29	0.35	0.34	15
(4 in.)	*1.67*	0.33	0.29	0.26	0.32	0.31	16
Solid plaster walls							
Gypsum lath (½ in.) and plas. ¾ in. each side							
(Lt. wt. agg.)	*1.39*	0.36	—	—	—	—	17
(Sand agg.)	*0.71*	0.48	—	—	—	—	18
1 in. each side							
(Lt. wt. agg.)	*1.73*	0.32	—	—	—	—	19
(Sand agg.)	*0.81*	0.46	—	—	—	—	20
Metal lath and plas.[b] 2 in. total thickness							
(Lt. wt. agg.)	*1.23*	0.38	—	—	—	—	21
(Sand agg.)	*0.56*	0.58	—	—	—	—	22
2½ in. total thickness							
(Lt. wt. agg.)	*1.60*	0.34	—	—	—	—	23
(Sand agg.)	*0.45*	0.55	—	—	—	—	24
Glass and glass blocks		See Table 21					

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] Metal core and supports disregarded. Plaster troweled smooth both sides.

Table 11 .... Coefficients of Transmission (*U*) of Frame Construction Ceilings and Floors[a]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference between the air on the two sides) and are based on still air (no wind) conditions on both sides



Example—Floor K 5

Resistances used are given below in this table or in Table 3 or 4

Heated room above unheated space

Construction (Heat flow down)	Resistance (*R*)
1. Top surface (still air)	0.92
2. Linoleum or tile (av. *R*)	0.05
3. Felt	0.06
4. Plywood (⅜ in.)	0.78
5. Wood subfloor ($^{25}/_{32}$ in.)	0.98
6. Air space[b] (7½ in.)	1.25
7. Metal lath and ¾ in. plas. (lt. wt. agg.)	0.47
8. Bottom surface (still air)	0.92
Total resistance	5.45
$U = 1/R = 1/5.45 =$	0.18

See value 0.18 in boldface type in table below.

Example of Substitution

Resistances used are given below in this table or in Table 3 or 4

Assume heated room is below unheated space so heat flow is up		
Total resistance		5.45
Deduct 1. Top surface (heat flow down)	0.92	
6. Air space (heat flow down)	1.25	
8. Bottom surface (heat flow down)	0.92	3.09
Difference		2.36
Add 1. Top surface (heat flow up)	0.61	
6. Air space (heat flow up)	0.85	
8. Bottom surface (heat flow up)	0.61	2.07
Total resistance		4.43
$U = 1/R = 1/4.43 =$		0.23

To Adjust U Values for Construction with Added Insulation between Framing Members, See Table 16

Direction of Heat →		*Heat Flow Upward (Winter Conditions)*						*Heat Flow Downward (Summer Conditions)*						
		Type of Floor												
Type of Ceiling		None	Wood subfloor ($^{25}/_{32}$ in.)	Wood subfloor ($^{25}/_{32}$ in.), felt, and— Cement (1½ in.) and ceramic tile (1½ in.)	Wood subfloor ($^{25}/_{32}$ in.), felt, and— Hardwood floor (⅝ in.)	Wood subfloor ($^{25}/_{32}$ in.), felt, and— Plywood (⅜ in.) and floor tile or linoleum (1⅛ in.)	Wood subfloor ($^{25}/_{32}$ in.), felt, and— Insul. bd. (½ in.) and hard bd. (¼ in.) and floor tile or linoleum (1⅛ in.)	None	Wood subfloor ($^{25}/_{32}$ in.)	Wood subfloor ($^{25}/_{32}$ in.), felt, and— Cement (1½ in.) and ceramic tile (1½ in.)	Wood subfloor ($^{25}/_{32}$ in.), felt, and— Hardwood floor (¾ in.)	Wood subfloor ($^{25}/_{32}$ in.), felt, and— Plywood (⅜ in.) and floor tile or linoleum (1⅛ in.)	Wood subfloor ($^{25}/_{32}$ in.), felt, and— Insul. bd. (½ in.) and hard bd. (¼ in.) and floor tile or linoleum (1⅛ in.)	Number
Resistance →		—	0.98	1.38	1.72	1.87	2.26	—	0.98	1.38	1.72	1.87	2.26	
Material	*R*	*U* A	*U* B	*U* C	*U* D	*U* E	*U* F	*U* G	*U* H	*U* I	*U* J	*U* K	*U* L	
None	—	—	0.45	0.38	0.34	0.31	0.28	—	0.35	0.31	0.28	0.26	0.24	1
Gypsum bd. (⅜ in.)	0.32	0.65	0.30	0.27	0.24	0.23	0.21	0.46	0.23	0.21	0.20	0.18	0.17	2
Gypsum lath (⅜ in.) and ½ in. plas. (lt. wt. agg.)	0.64	0.54	0.27	0.24	0.23	0.21	0.20	0.40	0.21	0.20	0.18	0.17	0.16	3
Gypsum lath (⅜ in.) and ½ in. plas. (sand agg.)	0.41	0.61	0.29	0.26	0.24	0.22	0.21	0.44	0.22	0.21	0.19	0.18	0.17	4
Metal lath and ¾ in. plas. (lt. wt. agg.)	0.47	0.59	0.28	0.26	0.23	0.22	0.20	0.43	0.22	0.20	0.19	0.18	0.17	5
Metal lath and ¾ in. plas. (sand agg.)	0.13	0.74	0.31	0.28	0.26	0.24	0.22	0.51	0.24	0.22	0.20	0.19	0.18	6
Insul. bd. (½ in.)	1.43	0.38	0.22	0.20	0.19	0.18	0.17	0.31	0.18	0.17	0.16	0.15	0.15	7
Insul. bd. lath (½ in.) and ½ in. plas. (sand agg.)	1.52	0.36	0.22	0.20	0.19	0.18	0.17	0.30	0.18	0.17	0.16	0.15	0.14	8
Acoustical tile														
(½ in.) on gypsum bd. (⅜ in.)	1.61[d]	0.37	0.22	0.20	0.19	0.18	0.17	0.30	0.18	0.17	0.16	0.15	0.14	9
(½ in.) on furring	1.19	0.41	0.24	0.22	0.20	0.19	0.18	0.33	0.19	0.18	0.17	0.16	0.15	10
(¾ in.) on gypsum bd. (⅜ in.)	2.10[d]	0.30	0.19	0.18	0.17	0.16	0.15	0.25	0.16	0.15	0.15	0.14	0.13	11
(¾ in.) on furring	1.78	0.33	0.21	0.19	0.18	0.17	0.16	0.28	0.17	0.16	0.15	0.15	0.14	12
Wood lath and ½ in. plas. (sand agg.)	0.40	0.62	0.29	0.26	0.24	0.22	0.21	0.45	0.22	0.21	0.19	0.18	0.17	13

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] To adjust *U* values for the effect of added insulation between framing members, see Table 16.
[c] Includes asphalt, rubber, and plastic tile (⅛ in.), ceramic tile, or terrazzo (1 in.).
[d] Includes thermal resistance of ⅜ in. gypsum wall board.

per, $^{25}/_{32}$-in. wood sheathing, studs, gypsum lath, and sand aggregate plaster, with 2-in. fibrous insulation between studs.

Solution: According to the example calculation in Table 5 a wall of this construction with no insulation between studs has a coefficient *U* of 0.24. Referring to Table 16, Part A, it will be found that a wall of this value with 2-in. fibrous insulation between the studs has a coefficient *U* of 0.087.

Attention is called to the necessity of applying the insulating material in accordance with the manufacturer's specification. The engineer must evaluate carefully the economic considerations involved in the selection of an insulating material as adapted to various building constructions. Lack of proper evaluation, or improper installation may lead to unsatisfactory results. Special attention must be given to vapor barriers as outlined in Chapter 10. Moisture from condensation or other sources materially reduces the heat-flow resistance of insulation.

INSULATED CONSTRUCTIONS—HOW TO USE TABLE 16

In Tables 5 through 15, *U* values are given for many common types of building wall, floor, and ceiling constructions. For such of these constructions as contain an air space, the tabulated *U* value is based on the assumption that the air space is empty, and that its surfaces are of ordinary building materials of low thermal reflectivity, such as wood, masonry, plaster, or paper. Considerable benefit in reducing the heat transmission coefficient of a construction can be effected by the application of thermal insulating materials in the air space.

Table 16 provides a means of determining, without calculation, the *U* value of the between-framing area of such constructions with the added insulation installed in the air space. Column 1 of Table 16 refers to the *U* values of uninsulated constructions as taken from Tables 5 through 15. Columns 2 to 14 of the table give corresponding coefficients U_i for the constructions with various insulating applications in the between-framing air space, as indicated by the column headings. Table 16 is in 5 parts, (A, B, C, D, and E) corresponding to the type of building element and the direction of heat flow. Each part is based on temperature conditions considered generally appropriate for the case.

Any and all *U* values are based on a series of assumptions as to nominal characteristics. Common variations in conditions, materials, workmanship, etc., can introduce much greater variations in *U* values than the variations resulting from the assumed mean temperatures and temperature differences described. From this it is also clear that the use of more than two significant figures in stating a *U* value is assuming more precision than can possibly exist. Three significant figures are used in Table 16 merely as a means of reducing cumulative errors when the table is used several times to obtain a single result. It should ont be assumed that the figures are accurate, overall, to three significant figures. Also, a result taken from Table 16 should always be rounded off to two significant figures.

To use Table 16:

1. Find in Column 1 of the appropriate table the value for *U* obtained from Tables 5 through 15 for the construction without added insulation.

2. If there is a column which exactly corresponds to the condition for which you desire the *U* value, read the answer in this column, opposite the Column 1 value. Interpolate if necessary.

3. If there is no column which fully corresponds to the condition for which you desire the *U* value, then obtain the answer by means of two or more steps—each time using the value obtained in one step to reenter the same table through Column 1. In this way *U* values may be obtained for combinations of insulation in the framing space.

For combinations of fibrous insulations and air spaces, take account of the fibrous component first and the air space second as illustrated in Example 6. This is necessary to assure the approximately correct temperature difference across the air space.

Examples of the Use of Table 16

Example 3: Find the coefficient of transmission U_i of the frame wall shown as an example at the top of Table 5 when (*a*) a 2-inch blanket fibrous insulation is added between the studs and in contact with the gypsum lath and when (*b*) aluminum-foil-backed gypsum lath is added.

Solution: According to Table 5, the *U* value for this construction with no insulation in the air space is 0.24. Referring to Table 16, Part A, for walls, it is found that corresponding to the value 0.24 in Column 1, the coefficient for conditions (*a*) and (*b*) are:

(*a*) In Column 4, find U_i = 0.087. Use 0.09.

(*b*) From Table 3, Section C, the effective emissivity E of aluminum foil is 0.05.

In Column 8 find U_i = 0.173. Use 0.17.

Example 4: Consider the floor-ceiling construction shown in the example at the top of Table 11, insulated with a sheet of aluminum foil, or paper faced on both sides with foil, (effective emissivity E of air space = 0.05) placed between the joists and dividing the air space into two equal spaces. From Table 11, the *U* value for the uninsulated construction is 0.18 for heat flow down for summer, and 0.22 for heat flow up for winter. Determine the coefficient U_i for:

(*a*) Heat flowing downward from uncooled room above to cooled space below (a summer condition) and,

(*b*) Heat flowing upward from heated room below to unheated space above (a winter condition).

Solution:

(*a*) Use Table 16, Part D, and corresponding to *U* = 0.18 in Column 1 find, in Column 11, U_i = 0.049. Use 0.05.

(*b*) Use Table 16, Part C, and corresponding to *U* = 0.22 in Column 1 find, in Column 11, U_i = 0.12.

Special Uses of Table 16

Values of U_i for insulating applications or combinations other than those indicated by the headings of Columns 2 to 14 of Table 16 can be ascertained if the table is used appropriately. For instance, going horizontally in the table from Column 2 to Column 3 is equivalent to adding ¼ in. of fibrous insulation to the construction. Similarly, going from Column 2 to Column 4 adds 1½ in. of fibrous insulation to the construction. In the same way, going horizontally from Column 8 to Column 11 is equivalent to adding to a construction the insulating value of one additional highly reflective (E = 0.05) air space, and going from Column 6 to Column 12 in effect adds two non-reflective (E = 0.82) air spaces to the construction, etc.

Examples 5 and 6 show the combinational use of Table 16

Example 5: Determine the coefficient U_i for the wall of *Example 3*, with 1½ in. of fibrous insulation added on one side of the air space.

Solution: U_i can be determined in several ways, by using Table 16, Part A, for example;

(*a*) By going from Column 1 to Column 2 three times, for a total of 1½ inches of insulation, as follows; enter Column 1 at 0.24 and find U_i = 0.166 in Column 2; enter Column 1 at 0.166 and find U_i = 0.126 in Column 2; enter Column 1 at 0.126 and find U_i = 0.102 in Column 2. Therefore, for a total of 1½ in. of insulation, use 0.10.

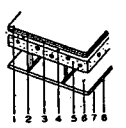
(*b*) More simply, by going from 0.24 in Column 2 to 0.103 in Column 4. Use 0.10.

Example 6: Considering again the wall of *Example 3*, assume that one-inch blanket insulation is to be installed in mid-space,

(*Continued on p. 113.*)

Table 12A .... Coefficients of Transmission (*U*) of Concrete Floor-Ceiling Constructions[a] (Winter Conditions, Upward Flow)

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides



Example—Floor 3 4

Resistances used are given below in this table or in Table 3 or 4

Heated room below unheated space Construction (heat flow up)	Resistance (*R*)
1. Top surface (still air)	0.61
2. Asphalt tile and felt	0.11
3. Plywood (⅜ in.)	0.78
4. Air space[a]	0.85
5. Concrete slab 4 in. (av. *R*)	0.40
6. Air space[b] (8 in.)	0.85
7. Metal lath and ½ plas. (sand agg.)	0.13
8. Bottom surface (still air)	0.61
Total resistance	4.54
$U = 1/R = 1/4.54 =$	0.23

See value 0.23 in bold face type in table below.

Example of Substitution

Resistances used are given below in this table or in Table 3 or 4

Replace items 2, 3, and 4 with hard wood block (1 13/16 in.) on slab		
Total resistance		4.54
Deduct 2. Asphalt tile and felt	0.11	
3. Plywood (⅜ in.)	0.78	
4. Air space	0.85	1.74
Difference		2.80
Add 2. Wood block (1 13/16 in.)		0.74
Total resistance		3.54
$U = 1/R = 1/3.54 =$		0.30

To Adjust U Values for Construction with Added Insulation in Air Space above Suspended Ceiling, See Table 16

Type of Deck	R	Av. R	Type of Finish Floor	Av. R	Type of Ceiling: None	Ceiling Applied Directly to Slab: Plas. (Lt. wt. agg.) ½ in.	Ceiling Applied Directly to Slab: Plas. (Sand agg.) ½ in.	Ceiling Applied Directly to Slab: Acoustical tile—glued ½ in.	Ceiling Applied Directly to Slab: Acoustical tile—glued ¾ in.	Suspended Ceiling: Gypsum bd. (½ in.) and Plas. No plas.	Suspended Ceiling: Gypsum bd. (½ in.) and Plas. (Lt. wt. agg.) ½ in.	Suspended Ceiling: Gypsum bd. (½ in.) and Plas. (Sand agg.) ¾ in.	Suspended Ceiling: Metal lath and plas. (Lt. wt. agg.) ½ in.	Suspended Ceiling: Metal lath and plas. (Sand agg.) ¾ in.	Suspended Ceiling: Acoustical tile: On furring or channels ¼ in.	Suspended Ceiling: Acoustical tile: On furring or channels ¾ in.	Suspended Ceiling: Acoustical tile: On gypsum bd. (⅜ in.) ¼ in.	Suspended Ceiling: Acoustical tile: On gypsum bd. (⅜ in.) ¾ in.	Number
			Resistance →		—	0.08	0.02	1.19	1.78	0.32	0.64	0.41	0.47	0.13	1.19	1.78	1.51	2.10	
Material					U	U	U	U	U	U	U	U	U	U	U	U	U	U	
					A	B	C	D	E	F	G	H	I	J	K	L	M	N	
Concrete[d] (sand agg.) (4 in.) (6 in.)	0.32 0.48	0.40	None	—	0.62	0.59	0.61	0.36	0.29	0.36	0.32	0.35	0.34	0.38	0.27	0.24	0.25	0.22	1
			Floor tile[e] or linoleum (⅛ in.)	0.05	0.60	0.57	0.59	0.35	0.29	0.35	0.32	0.34	0.33	0.38	0.27	0.23	0.25	0.22	2
			Wood block (1 13/16 in.) on slab	0.74	0.42	0.41	0.42	0.28	0.24	0.28	0.26	0.28	0.27	0.30	0.23	0.20	0.21	0.19	3
			Floor on sleepers Plywood subfloor (⅝ in.), felt and floor tile[e] or linoleum (⅛ in.)	0.89	0.30	0.29	0.30	0.22	0.19	0.22	0.21	0.22	0.21	0.23	0.19	0.17	0.17	0.16	4
			Wood subfloor (2 5/32 in.), felt and hardwood (¾ in.)	1.78	0.24	0.24	0.24	0.19	0.17	0.19	0.18	0.18	0.18	0.19	0.16	0.15	0.15	0.14	5
Concrete[d] (sand agg.) (8 in.) (10 in.)	0.64 0.80	0.72	None	—	0.52	0.50	0.51	0.32	0.27	0.32	0.29	0.31	0.31	0.34	0.25	0.22	0.23	0.20	6
			Floor tile[e] or linoleum (⅛ in.)	0.05	0.50	0.48	0.50	0.31	0.26	0.32	0.29	0.31	0.30	0.34	0.25	0.22	0.23	0.20	7
			Wood block (1 13/16 in.) on slab	0.74	0.37	0.36	0.37	0.26	0.22	0.26	0.24	0.25	0.25	0.27	0.21	0.19	0.20	0.18	8
			Floor on sleepers Plywood subfloor (⅝ in.), felt and floor tile[e] or linoleum (⅛ in.)	0.89	0.27	0.27	0.27	0.21	0.18	0.21	0.19	0.20	0.20	0.21	0.17	0.16	0.17	0.15	9
			Wood subfloor (2 5/32 in.), felt and hardwood (¾ in.)	1.78	0.22	0.22	0.22	0.18	0.16	0.18	0.17	0.17	0.17	0.18	0.16	0.14	0.15	0.13	10

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] To adjust *U* values for the effect of added insulation between framing members, see Table 16.
[c] Table 16 can be used only if sleeper space is non-reflective. The sleeper air space is not to be counted in using Table 16.
[d] Concrete is assumed to have a thermal conductivity *k* of 12.0 as given in Table 4.
[e] Includes asphalt, rubber, and plastic tile (⅛ in.), ceramic tile on terrazzo (1 in.).

Table 12B .... Coefficients of Transmission (*U*) of Concrete Floor-Ceiling Constructions[a] (Summer Conditions, Downward Flow)

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides

Type of Deck	R	Av. R	Type of Finish Floor	Av. R	Type of Ceiling: None	Ceiling Applied Directly to Slab: Plas. (Lt. wt. agg.) ⅛ in.	Ceiling Applied Directly to Slab: Plas. (Sand agg.) ⅛ in.	Ceiling Applied Directly to Slab: Acoustical tile—glued ½ in.	Ceiling Applied Directly to Slab: Acoustical tile—glued ¾ in.	Suspended Ceiling: Gypsum bd. (⅜ in.) and Plas. No plas.	Suspended Ceiling: Gypsum bd. (⅜ in.) and Plas. (Lt. wt. agg.) ½ in.	Suspended Ceiling: Gypsum bd. (⅜ in.) and Plas. (Sand agg.) ½ in.	Suspended Ceiling: Metal lath and plas. (Lt. wt. agg.) ¾ in.	Suspended Ceiling: Metal lath and plas. (Sand agg.) ¾ in.	Suspended Ceiling: Acoustical tile: On furring or channels ½ in.	Suspended Ceiling: Acoustical tile: On furring or channels ¾ in.	Suspended Ceiling: Acoustical tile: On gypsum bd. (⅜ in.) ½ in.	Suspended Ceiling: Acoustical tile: On gypsum bd. (⅜ in.) ¾ in.	Number
			Resistance →		—	0.08	0.02	1.19	1.78	0.32	0.64	0.41	0.47	0.13	1.19	1.78	1.51	2.10	
Material					U	U	U	U	U	U	U	U	U	U	U	U	U	U	
					O	P	Q	R	S	T	U	V	W	X	Y	Z	Z′	Z″	
Concrete[d] (sand agg.) (4 in.) (6 in.)	0.32 0.48	0.40	None	—	0.45	0.43	0.44	0.29	0.25	0.26	0.24	0.26	0.25	0.28	0.21	0.19	0.20	0.18	1
			Floor tile[e] or linoleum (⅛ in.)	0.05	0.44	0.42	0.43	0.29	0.25	0.26	0.24	0.25	0.25	0.27	0.21	0.19	0.20	0.18	2
			Wood block (1 13/16 in.) on slab	0.74	0.34	0.33	0.33	0.24	0.21	0.22	0.21	0.22	0.21	0.23	0.19	0.17	0.17	0.16	3
			Floor on sleepers Plywood subfloor (⅝ in.), felt and floor tile[e] or linoleum (⅛ in.)	0.89	0.23	0.23	0.23	0.18	0.17	0.17	0.16	0.17	0.17	0.18	0.15	0.14	0.14	0.13	4
			Wood subfloor (2 5/32 in.), felt and hardwood (¾ in.)	1.78	0.20	0.19	0.20	0.16	0.15	0.15	0.14	0.15	0.15	0.16	0.13	0.12	0.13	0.12	5
Concrete[d] (sand agg.) (8 in.) (10 in.)	0.64 0.80	0.72	None	—	0.39	0.38	0.39	0.27	0.23	0.24	0.23	0.24	0.23	0.26	0.20	0.18	0.19	0.17	6
			Floor tile[e] or linoleum (⅛ in.)	0.05	0.38	0.37	0.38	0.26	0.23	0.24	0.22	0.24	0.23	0.25	0.20	0.18	0.19	0.17	7
			Wood block (1 13/16 in.) on slab	0.74	0.30	0.30	0.30	0.22	0.20	0.21	0.19	0.20	0.20	0.21	0.17	0.16	0.17	0.15	8
			Floor on sleepers Plywood-subfloor (⅝ in.), felt and floor tile[e] or linoleum (⅛ in.)	0.89	0.22	0.21	0.22	0.17	0.16	0.16	0.15	0.16	0.16	0.17	0.14	0.13	0.14	0.13	9
			Wood subfloor (2 5/32 in.), felt and hardwood (¾ in.)	1.78	0.19	0.18	0.18	0.15	0.14	0.14	0.14	0.14	0.14	0.15	0.13	0.12	0.12	0.11	10

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] To adjust *U* values for the effect of added insulation between framing members, see Table 16.
[c] Table 16 can be used only if sleeper space is nonreflective. The sleeper air space is not to be counted in using Table 16.
[d] Concrete is assumed to have a thermal conductivity *k* of 12.0 as given in Table 4.
[e] Includes asphalt, rubber, and plastic tile (⅛ in.), ceramic tile on terrazzo (1 in.).

leaving equal air spaces on the two sides. The coefficient U_i is desired for two cases in which

(*a*) the blanket has non-reflective surfaces on both sides, so that both air spaces are non-reflective (E = 0.82) and,

(*b*) the blanket has one highly reflective surface, so that one of the air spaces is highly reflective (E = 0.05).

Solution:

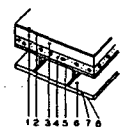
(*a*) In Table 16, Part A, when *U* without added insulation is 0.24, U_i for one inch of fibrous insulation and one non-reflective air space, is 0.127 in Column 3. To add the second non-reflective air space, enter Column 6 at 0.127, and by interpolating find U_i = 0.112 in Column 9. Use 0.11.

(*b*) U_i for one inch of fibrous insulation and one non-reflective air space is 0.127 as in (*a*). To add one E = 0.05 air space, enter Column 8 at 0.127 and by interpolating find U_i = 0.092 in Column 11. Use 0.09.

(*Continued on p. 121.*)

Table 13A .... Coefficients of Transmission (*U*) of Flat Masonry Roofs with Built-up Roofing, with and without Suspended Ceilings[a] (Winter Conditions, Upward Flow)

These Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph



Example—K 4

Resistances used are given below in this table or in Table 3 or 4

Construction (heat flow up)	Resistance (*R*)
1. Outside surface (15 mph wind)	*0.17*
2. Built-up roofing—⅜ in.	*0.33*
3. Roof insulation (none)	—
4. Concrete slab (lt. wt. agg.) (2 in.)	*2.22*
5. Corrugated metal	*0*
6. Air space[b]	*0.85*
7. Metal lath and ¾ in. plas. (lt. wt. agg.)	*0.47*
8. Inside surface (still air)	*0.61*
Total resistance	*4.65*
$U = 1/R = 1/4.65 =$	0.22

See value 0.22 in boldface type in table below.

Example of Substitution

Resistances used are given below in this table or in Table 3 or 4

Replace item 4 with 4 in. concrete slab (gravel agg.) and roof insulation (*C* = 0.36) on top of slab.	
Total resistance	*4.65*
Deduct	
4. Concrete slab (lt. wt. agg.) (2 in.)	*2.22*
Difference	*2.43*
Add 3. Roof insulation (*C* = 0.36)	*2.78*
4. Concrete slab (gravel agg.) 4 in.	*0.44*
Total resistance	*5.65*
$U = 1/R = 1/5.65 =$	0.18

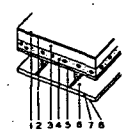
To Adjust U Values for Construction with Added Insulation in Air Space, See Table 16

Type of Deck	Type of Form		Type of Ceiling: Roof Insulation No Ceiling — C value of roof insulation: None	0.72	0.36	0.24	0.19	0.15	0.12	Suspended Ceiling[b]: Gypsum bd. (⅜ in.) and plas.: No plas.	Lt. wt. agg. ½ in.	Sand agg. ½ in.	Metal lath and plas.: Lt. wt. agg. ¾ in.	Sand agg. ¾ in.	Acoustical tile: On furring or channels: ½ in.	¾ in.	On gypsum bd. (⅜ in.): ½ in.	¾ in.	Number
	Resistance →		—	1.39	2.78	4.17	5.26	6.67	8.33	0.32	0.64	0.41	0.47	0.13	1.19	1.75	1.51	2.10	
Material	*R*	Material	*R*	*U* A	*U* B	*U* C	*U* D	*U* E	*U* F	*U* G	*U* H	*U* I	*U* J	*U* K	*U* L	*U* M	*U* N	*U* O	*U* P	
Concrete slab[d] Gravel agg.																				
(4 in.).. *0.32*		Temporary	—	0.70	0.35	0.24	0.18	0.15	0.12	0.10	0.38	0.34	0.37	0.36	0.41	0.29	0.25	0.26	0.23	1
(6 in.).. *0.48*		Temporary	—	0.63	0.34	0.23	0.17	0.15	0.12	0.10	0.36	0.32	0.35	0.34	0.39	0.27	0.24	0.25	0.22	2
(8 in.).. *0.64*		Temporary	—	0.57	0.32	0.22	0.17	0.14	0.12	0.10	0.34	0.31	0.33	0.33	0.37	0.26	0.23	0.24	0.21	3
Lt. wt. agg.[c]																				
(2 in.).. *2.22*		Corrugated metal[e]	*0*	0.30	0.21	0.16	0.13	0.12	0.10	0.09	0.22	0.21	0.22	0.22	0.23	0.19	0.17	0.18	0.16	4
		Insulation bd. (1 in.)	*2.78*	0.16	0.13	0.11	0.10	0.09	0.08	0.07	0.14	0.13	0.14	0.13	0.14	0.12	0.11	0.12	0.11	5
		Insulation bd. (1½ in.)	*4.17*	0.13	0.11	0.10	0.09	0.08	0.07	0.06	0.12	0.11	*0.11*	*0.11*	*0.12*	*0.10*	*0.10*	*0.10*	*0.10*	6
		Glass fiber bd. (1 in.)	*4.00*	0.14	0.11	0.10	0.09	0.08	0.07	0.06	0.12	0.11	0.12	0.12	0.12	0.11	0.10	0.10	0.10	7
(3 in.).. *3.33*		Corrugated metal[e]	*0*	0.23	0.17	0.14	0.12	0.10	0.09	0.08	0.18	0.17	0.18	0.17	0.18	0.15	0.14	0.15	0.14	8
		Insulation bd. (1 in.)	*2.78*	0.14	0.12	0.10	0.09	0.08	0.07	0.06	0.12	0.11	0.12	0.12	0.12	0.11	0.10	0.10	0.10	9
		Insulation bd. (1½ in.)	*4.17*	0.12	0.10	0.09	0.08	0.07	0.07	0.06	0.10	0.10	0.10	0.10	0.10	0.09	0.09	0.09	0.09	10
		Glass fiber bd. (1 in.)	*4.00*	0.12	0.10	0.09	0.08	0.07	0.07	0.06	0.10	0.10	0.10	0.10	0.11	0.10	0.09	0.09	0.09	11
(4 in.).. *4.44*		Corrugated metal[e]	*0*	0.18	0.14	0.12	0.10	0.09	0.08	0.07	0.15	0.14	0.15	0.15	0.15	0.13	0.12	0.13	0.12	12
		Insulation bd. (1 in.)	*2.78*	0.12	0.10	0.09	0.08	0.07	0.07	0.06	0.11	0.10	0.10	0.10	0.11	0.10	0.09	0.09	0.09	13
		Insulation bd. (1½ in.)	*4.17*	0.10	0.09	0.08	0.07	0.07	0.06	0.06	0.09	0.09	0.09	0.09	0.09	0.09	0.08	0.08	0.08	14
		Glass fiber bd. (1 in.)	*4.00*	0.10	0.09	0.08	0.07	0.07	0.06	0.06	0.09	0.09	0.09	0.09	0.09	0.09	0.08	0.08	0.08	15
Gypsum slab[g]																				
(2 in.).. *1.20*		Gypsum bd. (½ in.)	*0.45*	0.36	0.24	0.18	0.14	0.12	0.11	0.09	0.25	0.24	0.25	0.25	0.27	0.21	0.19	0.20	0.18	16
		Insulation bd. (1 in.)	*2.78*	0.20	0.15	0.13	0.11	0.10	0.09	0.07	0.16	0.15	0.16	0.16	0.16	0.14	0.13	0.13	0.12	17
		Insulation bd. (1½ in.)	*4.17*	0.15	0.13	0.11	0.09	0.09	0.08	0.07	0.13	0.13	0.13	0.13	0.13	0.12	0.11	0.11	0.11	18
		Asbestos-cement bd.[f] (¼ in.)	*0.06*	0.40	0.26	0.19	0.15	0.13	0.11	0.09	0.27	0.25	0.26	0.26	0.29	0.22	0.19	0.20	0.18	19
		Glass fiber bd. (1 in.)	*4.00*	0.16	0.13	0.11	0.10	0.09	0.08	0.07	0.13	0.13	0.13	0.13	0.14	0.12	0.11	0.12	0.11	20
(3 in.).. *1.80*		Gypsum bd. (½ in.)	*0.45*	0.30	0.21	0.16	0.13	0.12	0.10	0.09	0.22	0.21	0.22	0.21	0.23	0.19	0.17	0.17	0.16	21
		Insulation bd. (1 in.)	*2.78*	0.15	0.14	0.12	0.10	0.09	0.08	0.07	0.15	0.14	0.14	0.14	0.14	0.13	0.12	0.12	0.12	22
		Insulation bd. (1½ in.)	*4.17*	0.14	0.12	0.10	0.09	0.08	0.07	0.06	0.12	0.12	0.12	0.12	0.12	0.11	0.10	0.11	0.10	23
		Asbestos-cement bd. (¼ in.)	*0.06*	0.34	0.23	0.17	0.14	0.12	0.10	0.09	0.24	0.22	0.24	0.23	0.25	0.20	0.18	0.19	0.17	24
		Glass fiber bd. (1 in.)	*4.00*	0.14	0.12	0.10	0.09	0.08	0.07	0.07	0.12	0.12	0.12	0.12	0.13	0.11	0.10	0.11	0.10	25
(4 in.).. *2.40*		*Gypsum bd. (½ in.)*	*0.45*	0.25	0.19	0.15	0.12	0.11	0.09	0.08	0.19	0.18	0.19	0.19	0.20	0.17	0.15	0.16	0.14	26
		Insulation bd. (1 in.)	*2.78*	0.16	0.13	0.11	0.10	0.09	0.08	0.07	0.13	0.13	0.13	0.13	0.14	0.12	0.11	0.12	0.11	27
		Insulation bd. (1½ in.)	*4.17*	0.13	0.11	0.10	0.08	0.08	0.07	0.06	0.11	0.11	0.11	0.11	0.12	0.10	0.10	0.10	0.09	28
		Asbestos-cement bd. (¼ in.)	*0.06*	0.28	0.20	0.16	0.13	0.11	0.10	0.08	0.21	0.20	0.21	0.20	0.22	0.18	0.16	0.17	0.15	29
		Glass fiber bd. (1 in.)	*4.00*	0.13	0.11	0.10	0.09	0.08	0.07	0.06	0.12	0.11	0.11	0.11	0.12	0.10	0.10	0.10	0.10	30

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] To adjust *U* values for the effect of added insulation between framing members, see Table 16.
[c] *U* values would also apply if slab were poured on metal lath, paper-backed wire, fabric, or asbestos-cement board (¼ in.).
[d] Concrete assumed to have a thermal conductivity *k* of 12.0 and a density of 140 lb per cu ft.
[e] Concrete assumed to have a thermal conductivity *k* of 0.90 and a density of 30 lb per cu ft.
[f] Gypsum slab ¾ in. thick since this is recommended practice.
[g] Gypsum fiber concrete with 12½ percent wood chips (thermal conductivity $k = 1.66$).
[h] See Table 16 E for *U* value of roof and ceiling construction with roof insulation added to roof deck.

Table 13B .... Coefficients of Transmission (*U*) of Flat Masonry Roofs with Built-up Roofing with and without Suspended Ceiling[a] (Summer Conditions, Downward Flow)

These coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 7.5 mph



Example—Roof 1' 2

Resistances used are given below in this table or in Table 3 or 4

Construction (heat flow down)	Resistance (*R*)
1. Outside surface (7.5 mph wind)	*0.25*
2. Built-up roofing (⅜ in.)	*0.33*
3. Roof insulation (none)	*0.00*
4. Concrete slab (gravel agg.) (6 in.)	*0.48*
5. Temporary form bd.	*0.00*
6. Air space[b] (8 in.)	*1.25*
7. Gypsum lath (⅜ in.) and ½ in. plas. (lt. wt. agg.)	*0.64*
8. Inside surface (still air)	*0.92*
Total resistance	*3.87*
$U = 1/R = 1/3.87 =$	0.26

See value 0.26 in boldface type in table below.

Example of Substitution

Resistances used are given below in this table or in Table 3 or 4

Replace item 3 with roof insulation (*C* = 0.36) and remove suspended ceiling	
Total resistance	*3.87*
Deduct 6. Air space	*1.25*
7. Gypsum lath (⅜ in.) and ½ in. plas. (lt. wt. agg.)	*0.64*
Difference	*1.98*
Add 3. Roof insulation (*C* = 0.36)	*2.78*
Total resistance =	*4.76*
$U = 1/R = 1/4.76 =$	0.21

To adjust U Values for Construction with Added Insulation in Air Space, See Table 16

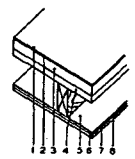
Type of Deck	Type of Form		Type of Ceiling: Roof Insulation—No Ceiling — C value of roof insulation: None	0.72	0.36	0.24	0.19	0.15	0.12	Suspended Ceiling[b]: Gypsum bd. (⅜ in.) and plas.: No plas.	Lt. wt. agg. (½ in.)	Sand agg. (½ in.)	Metal lath and plas.: Lt. wt. agg. (¾ in.)	Sand agg. (¾ in.)	Acoustical tile: On furring or channels: ½ in.	¾ in.	On gypsum bd. (⅜ in.): ½ in.	¾ in.	Number
	Resistance →		—	1.39	2.78	4.17	5.26	6.67	8.33	0.32	0.64	0.41	0.47	0.13	1.19	1.78	1.51	2.10	
Material	*R*	Material	*R*	*U* A′	*U* B′	*U* C′	*U* D′	*U* E′	*U* F′	*U* G′	*U* H′	*U* I′	*U* J′	*U* K′	*U* L′	*U* M′	*U* N′	*U* O′	*U* P′	
Concrete slab[d] (Gravel agg.)																				
(4 in.).. *0.32*		Temporary	—	0.55	0.31	0.22	0.17	0.14	0.12	0.10	0.30	0.27	0.29	0.28	0.31	0.24	0.21	0.22	0.19	1
(6 in.).. *0.48*		Temporary	—	0.51	0.30	0.21	0.16	0.14	0.12	0.10	0.28	0.26	0.28	0.27	0.30	0.23	0.20	0.21	0.19	2
(8 in.).. *0.64*		Temporary	—	0.47	0.28	0.20	0.16	0.14	0.11	0.10	0.27	0.25	0.26	0.26	0.29	0.22	0.19	0.20	0.18	3
Lt. wt. agg.[c]																				
(2 in.).. *2.22*		Corrugated metal[e]	*0*	0.27	0.20	0.15	0.13	0.11	0.10	0.08	0.19	0.18	0.19	0.18	0.20	0.16	0.15	0.15	0.14	4
		Insulation bd. (1 in.)	*2.78*	0.15	0.13	0.11	0.09	0.09	0.08	0.07	0.12	0.12	0.12	0.13	0.11	0.11	0.11	0.10	0.10	5
		Insulation bd. (1½ in.)	*4.17*	0.13	0.11	0.09	0.08	0.08	0.07	0.06	0.11	0.10	0.10	0.10	0.11	0.10	0.09	0.09	0.09	6
		Glass fiber bd. (1 in.)	*4.00*	0.13	0.11	0.10	0.08	0.08	0.07	0.06	0.11	0.10	0.10	0.11	0.11	0.10	0.09	0.10	0.09	7
(3 in.).. *3.33*		Corrugated metal[e]	*0*	0.21	0.16	0.13	0.11	0.10	0.09	0.08	0.16	0.15	0.15	0.15	0.16	0.14	0.13	0.13	0.12	8
		Insulation bd. (1 in.)	*2.78*	0.13	0.11	0.10	0.08	0.08	0.07	0.06	0.11	0.11	0.11	0.11	0.11	0.10	0.09	0.10	0.09	9
		Insulation bd. (1½ in.)	*4.17*	0.11	0.10	0.08	0.08	0.07	0.06	0.06	0.09	0.09	0.09	0.09	0.10	0.09	0.08	0.09	0.08	10
		Glass fiber bd. (1 in.)	*4.00*	0.11	0.10	0.09	0.08	0.07	0.06	0.06	0.10	0.09	0.09	0.09	0.10	0.09	0.08	0.09	0.08	11
(4 in.).. *4.44*		Corrugated metal[e]	*0*	0.17	0.14	0.11	0.10	0.09	0.08	0.07	0.13	0.13	0.13	0.13	0.14	0.12	0.11	0.12	0.11	12
		Insulation bd. (1 in.)	*2.78*	0.11	0.10	0.09	0.08	0.07	0.06	0.06	0.10	0.09	0.10	0.10	0.10	0.09	0.09	0.09	0.08	13
		Insulation bd. (1½ in.)	*4.17*	0.10	0.09	0.08	0.07	0.07	0.06	0.05	0.09	0.08	0.09	0.08	0.09	0.08	0.08	0.08	0.07	14
		Glass fiber bd. (1 in.)	*4.00*	0.10	0.09	0.08	0.07	0.07	0.06	0.05	0.09	0.08	0.09	0.09	0.09	0.08	0.08	0.08	0.08	15
Gypsum slab[g]																				
(2 in.).. *1.20*		Gypsum bd. (½ in.)	*0.45*	0.32	0.22	0.17	0.14	0.12	0.10	0.09	0.21	0.20	0.21	0.21	0.22	0.18	0.16	0.17	0.15	16
		Insulation bd. (1 in.)	*2.78*	0.18	0.15	0.12	0.10	0.09	0.08	0.07	0.14	0.14	0.14	0.14	0.15	0.13	0.12	0.12	0.11	17
		Insulation bd. (1½ in.)	*4.17*	0.15	0.12	0.10	0.09	0.08	0.07	0.07	0.12	0.12	0.12	0.12	0.12	0.11	0.10	0.10	0.10	18
		Asbestos-cement bd.[f] (¼ in.)	*0.06*	0.34	0.23	0.18	0.14	0.12	0.10	0.09	0.22	0.21	0.22	0.22	0.23	0.19	0.17	0.18	0.16	19
		Glass fiber bd. (1 in.)	*4.00*	0.15	0.12	0.11	0.09	0.08	0.07	0.07	0.12	0.12	0.12	0.12	0.12	0.11	0.10	0.11	0.10	20
(3 in.).. *1.80*		Gypsum bd. (½ in.)	*0.45*	0.27	0.19	0.15	0.13	0.11	0.10	0.08	0.19	0.18	0.19	0.18	0.20	0.16	0.15	0.15	0.14	21
		Insulation bd. (1 in.)	*2.78*	0.16	0.13	0.11	0.10	0.09	0.08	0.07	0.13	0.13	0.13	0.13	0.13	0.12	0.11	0.11	0.11	22
		Insulation bd. (1½ in.)	*4.17*	0.13	0.11	0.10	0.09	0.08	0.07	0.06	0.11	0.11	0.11	0.11	0.11	0.10	0.10	0.10	0.09	23
		Asbestos-cement bd. (¼ in.)	*0.06*	0.30	0.21	0.16	0.13	0.12	0.10	0.09	0.20	0.19	0.20	0.20	0.21	0.17	0.16	0.16	0.15	24
		Glass fiber bd. (1 in.)	*4.00*	0.14	0.12	0.10	0.09	0.08	0.07	0.06	0.11	0.11	0.11	0.11	0.12	0.10	0.10	0.10	0.09	25
(4 in.).. *2.40*		Gypsum bd. (½ in.)	*0.45*	0.23	0.17	0.14	0.12	0.10	0.09	0.08	0.17	0.16	0.17	0.17	0.18	0.15	0.14	0.14	0.13	26
		Insulation bd. (1 in.)	*2.78*	0.15	0.12	0.11	0.09	0.08	0.07	0.07	0.12	0.12	0.12	0.12	0.12	0.11	0.10	0.11	0.10	27
		Insulation bd. (1½ in.)	*4.17*	0.12	0.11	0.09	0.08	0.08	0.07	0.06	0.10	0.10	0.10	0.10	0.11	0.10	0.09	0.09	0.09	28
		Asbestos-cement bd.[e] (¼ in.)	*0.06*	0.25	0.19	0.15	0.12	0.11	0.09	0.08	0.18	0.17	0.18	0.18	0.19	0.16	0.14	0.15	0.14	29
		Glass fiber bd. (1 in.)	*4.00*	0.13	0.11	0.09	0.08	0.08	0.07	0.06	0.11	0.10	0.10	0.10	0.11	0.10	0.07	0.09	0.09	30

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] To adjust *U* values for the effect of added insulation between framing members, see Table 16.
[c] *U* values would also apply if slab were poured on metal lath, paper-backed wire fabric, or asbestos-cement board (¼ in.).
[d] Concrete assumed to have a thermal conductivity *k* of 12.0 and a density of 140 lb per cu ft.
[e] Concrete assumed to have a thermal conductivity *k* of 0.90 and a density of 30 lb per cu ft.
[f] Gypsum slab ¾ in. thick since this is recommended practice.
[g] Gypsum fiber concrete with 12½ percent wood chips (thermal conductivity $k = 1.66$).
[h] See Table 16 E for *U* value of roof and ceiling construction with roof insulation added to roof deck.

Table 14A .... Coefficients of Transmission (U) of Wood or Metal Construction Flat Roofs and Ceilings[a] (Winter Conditions, Upward Flow)

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based upon an outside wind velocity of 15 mph

Example—Roof J.2



Resistances used are given below in this table or in Table 3 or 4

Construction (Heat flow up)	Resistance (R)
1. Outside surface (15 mph wind)	0.17
2. Built-up roofing, 3/8 in.	0.33
3. Roof insulation (C = 0.72)	1.39
4. Wood deck (1 in.)	0.98
5. Air space[b]	0.85
6. Gypsum wall board (3/8 in.)	0.32
7. Acoustical tile (1/2 in.)—glued	1.19
8. Inside surface (still air)	0.61
Total resistance	5.84
U = 1/R = 1/5.84	0.17

See value 0.17 in boldface type in table below.

Example of Substitution

Resistances used are given below in this table or in Table 3 or 4

Replace item 4 with 2 in. wood deck (exposed to inside) and omit items 5, 6, and 7.	
Total resistance	5.84
Deduct 4. Wood deck (1 in.)	0.98
5. Air space	0.85
6. Gypsum wall board (3/8 in.)	0.32
7. Acoustical tile (1/2 in.) glued	1.19 3.34
Difference	2.50
Add 4. Wood deck (2 in.)	2.03
Total resistance	4.53
U = 1/R = 1/4.53 =	0.22

To Adjust U Values for Construction with Added Insulation in Air Space, See Table 16

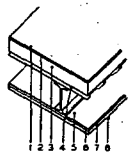
Type of Deck (Built-up Roof in All Cases)		Insulation Added on Top of Deck[d]		TYPE OF CEILING												
				None	Gypsum Bd. (3/8 in. and Plas.)			Metal Lath and Plas.		Insul. Bd. (1/2 in.)	Acoustical Tile					Number
					None	Lt. wt. agg. 1/2 in.	Sand agg. 1/2 in.	Lt. wt. agg. 3/4 in.	Sand agg. 3/4 in.	Plain (1.43) or 1/2 in. plas. sand agg. (1.52)	On Furring		On Gypsum Bd. (3/8 in.)		
		Conductance of insul.	Resistance								1/2 in.	3/4 in.	1/2 in.	3/4 in.	
Resistance →				0.32	0.64	0.41	0.47	0.13	1.47	1.19	1.78	1.51	2.10		
Material	R	C	R	U	U	U	U	U	U	U	U	U	U	U	
				A	B	C	D	E	F	G	H	I	J	K	
Wood[c] 1 in.	0.98	None	—	0.48	0.31	0.28	0.30	0.29	0.33	0.23	0.24	0.21	0.22	0.20	1
		0.72	1.39	0.29	0.22	0.20	0.21	0.21	0.22	0.17	0.18	0.16	0.17	0.16	2
		0.36	2.78	0.21	0.17	0.16	0.16	0.16	0.17	0.14	0.14	0.13	0.14	0.13	3
		0.24	4.17	0.16	0.13	0.13	0.13	0.13	0.14	0.12	0.12	0.11	0.12	0.11	4
		0.19	5.26	0.14	0.12	0.11	0.12	0.12	0.12	0.10	0.11	0.10	0.10	0.10	5
		0.15	6.67	0.11	0.10	0.10	0.10	0.10	0.10	0.09	0.09	0.09	0.09	0.09	6
		0.12	8.33	0.10	0.09	0.08	0.09	0.09	0.09	0.08	0.08	0.08	0.08	0.07	7
Wood[c] 2 in.	2.03	None	—	0.32	0.23	0.22	0.23	0.22	0.24	0.18	0.19	0.17	0.18	0.16	8
		0.72	1.39	0.22	0.18	0.17	0.17	0.17	0.18	0.15	0.15	0.14	0.15	0.13	9
		0.36	2.78	0.17	0.14	0.13	0.14	0.14	0.14	0.12	0.13	0.12	0.12	0.11	10
		0.24	4.17	0.14	0.12	0.11	0.12	0.12	0.12	0.10	0.11	0.10	0.10	0.10	11
		0.19	5.26	0.12	0.10	0.10	0.10	0.10	0.11	0.09	0.10	0.09	0.09	0.09	12
		0.15	6.67	0.10	0.09	0.09	0.09	0.09	0.09	0.08	0.08	0.08	0.08	0.08	13
		0.12	8.33	0.09	0.08	0.08	0.08	0.08	0.08	0.07	0.07	0.07	0.07	0.07	14
Wood[c] 3 in.	3.88	None	—	0.23	0.18	0.17	0.18	0.18	0.19	0.15	0.16	0.14	0.15	0.14	15
		0.72	1.39	0.17	0.14	0.14	0.14	0.14	0.15	0.12	0.13	0.12	0.12	0.11	16
		0.36	2.78	0.14	0.12	0.12	0.12	0.12	0.12	0.11	0.11	0.10	0.10	0.10	17
		0.24	4.17	0.12	0.10	0.10	0.10	0.10	0.10	0.09	0.09	0.09	0.09	0.09	18
		0.19	5.26	0.10	0.09	0.09	0.09	0.09	0.09	0.08	0.09	0.08	0.08	0.08	19
		0.15	6.67	0.09	0.08	0.08	0.08	0.08	0.08	0.07	0.08	0.07	0.07	0.07	20
		0.12	8.33	0.08	0.07	0.07	0.07	0.07	0.07	0.07	0.07	0.07	0.07	0.06	21
Preformed slabs—wood fiber and cement binder															
2 in.	3.60	None	—	0.21	0.17	0.16	0.17	0.17	0.18	0.14	0.15	0.14	0.14	0.13	22
3 in.	5.40	None	—	0.15	0.13	0.13	0.13	0.13	0.13	0.11	0.12	0.11	0.11	0.11	23
Flat metal roof deck	0	None	—	0.90	0.44	0.38	0.42	0.41	0.48	0.29	0.32	0.27	0.29	0.25	24
		0.72	1.39	0.40	0.27	0.25	0.27	0.26	0.29	0.21	0.22	0.19	0.21	0.18	25
		0.36	2.78	0.26	0.20	0.19	0.19	0.19	0.21	0.16	0.17	0.15	0.16	0.15	26
		0.24	4.17	0.19	0.16	0.15	0.15	0.15	0.16	0.13	0.14	0.13	0.13	0.12	27
		0.19	5.26	0.16	0.13	0.13	0.13	0.13	0.14	0.12	0.12	0.11	0.11	0.11	28
		0.15	6.67	0.13	0.11	0.11	0.11	0.11	0.11	0.10	0.10	0.10	0.10	0.09	29
		0.12	8.33	0.11	0.09	0.09	0.09	0.09	0.10	0.09	0.09	0.08	0.08	0.08	30

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] To adjust U values for the effect of added insulation between framing members, see Table 16.
[c] Wood deck 1, 2, and 3 in. is assumed to be 25/32, 1 9/16, and 2 5/16 in. thick, respectively. The thermal conductivity k is assumed to be 0.80.
[d] If a vapor barrier is used beneath roof insulation it will have a negligible effect on the U value. For information on vapor barrier requirements see Chapter 10, Moisture in Building Construction.

Table 14B .... Coefficients of Transmission (U) of Wood or Metal Construction Flat Roofs and Ceilings[a] (Summer Conditions, Downward Flow)

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based upon an outside wind velocity of 7.5 mph

Example—Roof F' 27



Resistances used are given below in this table or in Tables 3 or 4

Construction	Resistance (R)
1. Outside surface (7.5 mph wind)	0.25
2. Built-up roofing (3/8 in.)	0.33
3. Roof insulation (C = 0.24)[d]	4.17
4. Metal deck	0.00
5. Air space[b]	1.85
6. Metal lath and 7. 3/4 in. plas. (sand agg.)	0.15
8. Inside surface (still air)	0.68
Total resistance	7.03
U = 1/R = 1/7.03 =	0.14

See value 0.14 in boldface type in table below.

Example of Substitution

Resistances used are given below in this table or in Tables 3 or 4

Replace item 3 with roof insulation (C = 0.36) and items 6 and 7 with metal lath and 3/4 in. plas. (lt. wt. agg.)	
Total resistance	7.03
Deduct 3. Roof insulation (C = 0.24)	4.17
6. Metal lath and 7. 3/4 in. plas. (sand agg.)	0.15 4.30
Difference	2.73
Add 3. Roof insulation (C = 0.36)	2.78
6. Metal lath and 7. 3/4 in. plas. (lt. wt. agg.)	0.47 3.25
Total resistance	5.98
U = 1/R = 1/5.98 =	0.17

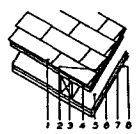
To Adjust U Values for Construction with Added Insulation in Air Space See Table 16

Type of Deck (Built-up Roof in All Cases)		Insulation Added on Top of Deck[d]		Type of Ceiling												
				None	Gypsum Bd. (3/8 in.) and Plas.			Metal Lath and Plas.		Insul. Bd. (1/2 in.)	Acoustical Tile					Number
					None	Lt. wt. agg. 1/2 in.	Sand agg. 1/2 in.	Lt. wt. agg. 3/4 in.	Sand agg. 3/4 in.	Plain (1.43) or 1/2 in. plas. sand agg. 1.52	On Furring		On Gypsum Bd. (1/2 in.)		
		Conductance of insul.	Resistance								1/2 in.	3/4 in.	1/2 in.	3/4 in.	
Resistance →				0.32	0.64	0.41	0.47	0.13	1.47	1.19	1.78	1.51	2.10		
Material	R	C	R	U	U	U	U	U	U	U	U	U	U	U	
				A'	B'	C'	D'	E'	F'	G'	H'	I'	J'	K'	
Wood[c] 1 in.	0.98	None	—	0.40	0.25	0.23	0.24	0.24	0.26	0.19	0.20	0.18	0.19	0.17	1
		0.72	1.39	0.26	0.18	0.17	0.18	0.18	0.19	0.15	0.16	0.15	0.15	0.14	2
		0.36	2.78	0.19	0.15	0.14	0.14	0.14	0.15	0.13	0.13	0.12	0.13	0.12	3
		0.24	4.17	0.15	0.12	0.12	0.12	0.12	0.12	0.11	0.11	0.10	0.11	0.10	4
		0.19	5.26	0.13	0.11	0.10	0.11	0.11	0.11	0.10	0.10	0.09	0.10	0.09	5
		0.15	6.67	0.11	0.09	0.09	0.09	0.09	0.10	0.08	0.09	0.08	0.08	0.08	6
		0.12	8.33	0.09	0.08	0.08	0.08	0.08	0.08	0.07	0.08	0.07	0.07	0.07	7
Wood[c] 2 in.	2.03	None	—	0.28	0.20	0.19	0.19	0.19	0.20	0.16	0.17	0.15	0.16	0.15	8
		0.72	1.39	0.20	0.15	0.15	0.15	0.15	0.16	0.13	0.14	0.13	0.13	0.12	9
		0.36	2.78	0.16	0.13	0.12	0.13	0.12	0.13	0.11	0.11	0.11	0.11	0.10	10
		0.24	4.17	0.13	0.11	0.10	0.11	0.11	0.11	0.10	0.10	0.09	0.10	0.09	11
		0.19	5.26	0.11	0.10	0.07	0.10	0.10	0.10	0.09	0.09	0.08	0.09	0.08	12
		0.15	6.67	0.10	0.09	0.08	0.08	0.08	0.09	0.08	0.08	0.08	0.08	0.07	13
		0.12	8.33	0.08	0.07	0.07	0.07	0.07	0.08	0.07	0.07	0.07	0.07	0.07	14
Wood[c] 3 in.	3.88	None	—	0.21	0.16	0.15	0.16	0.15	0.16	0.13	0.14	0.13	0.13	0.12	15
		0.72	1.39	0.16	0.13	0.12	0.13	0.13	0.13	0.11	0.12	0.11	0.11	0.11	16
		0.36	2.78	0.13	0.11	0.11	0.11	0.11	0.11	0.10	0.10	0.09	0.10	0.09	17
		0.24	4.17	0.11	0.10	0.09	0.09	0.09	0.10	0.09	0.09	0.08	0.09	0.08	18
		0.19	5.26	0.10	0.09	0.08	0.09	0.09	0.09	0.08	0.08	0.08	0.08	0.07	19
		0.15	6.67	0.09	0.08	0.08	0.08	0.08	0.08	0.07	0.07	0.07	0.07	0.07	20
		0.12	8.33	0.08	0.07	0.07	0.07	0.07	0.07	0.06	0.06	0.06	0.06	0.06	21
Preformed slabs—wood fiber and cement binder															
2 in.	3.60	None	—	0.20	0.15	0.14	0.15	0.15	0.15	0.13	0.13	0.12	0.13	0.12	22
3 in.	5.40	None	—	0.14	0.12	0.11	0.12	0.12	0.12	0.10	0.11	0.10	0.10	0.10	23
Flat Metal Roof Deck	0	None	—	0.67	0.33	0.30	0.32	0.31	0.35	0.24	0.26	0.22	0.24	0.21	24
		0.72	1.39	0.35	0.23	0.21	0.22	0.22	0.24	0.18	0.19	0.17	0.18	0.16	25
		0.36	2.78	0.23	0.17	0.16	0.17	0.17	0.18	0.14	0.15	0.14	0.14	0.13	26
		0.24	4.17	0.18	0.14	0.13	0.14	0.14	0.14	0.12	0.12	0.12	0.12	0.11	27
		0.19	5.26	0.15	0.12	0.12	0.12	0.12	0.12	0.11	0.11	0.11	0.10	0.10	28
		0.15	6.67	0.12	0.10	0.10	0.10	0.10	0.10	0.09	0.09	0.09	0.09	0.09	29
		0.12	8.33	0.10	0.09	0.09	0.09	0.09	0.09	0.08	0.08	0.08	0.08	0.08	30

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] To adjust U values for the effect of added insulation between framing members, see Table 16.
[c] Wood deck 1, 2, and 3 in. is assumed to be 25/32, 1 9/16, and 2 5/16 in. thick, respectively. The thermal conductivity k is assumed to be 0.80.
[d] If a vapor barrier is used beneath roof insulation it will have a negligible effect on the U value. For information on vapor barrier requirements see Chapter 10, Moisture in Building Construction.

Table 15 .... Coefficients of Transmission (U) of Pitched Roofs[a, b]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph for heat flow upward and 7.5 mph for heat flow downward



Example—Roof C 4		Example of Substitution	
Resistances used are given below in this table or in Table 3 or 4		*Resistances used are given below in this table or in Table 3 or 4*	
Construction (Heat flow up)	Resistance (*R*)	Find *U* value for same construction with heat flow down (summer conditions)	
1. Outside surface (15 mph wind)	*0.17*	Total resistance	*3.19*
2. Slate shingles (½ in.)	*0.05*	Deduct 1. Outside surface (15 mph wind)	*0.17*
3. Building paper	*0.06*	5. Air space	*0.90*
4. Wood sheathing ($^{25}/_{32}$ in.)	*0.98*	8. Inside surface (still air)	*0.62* 1.69
5. Air space[a]	*0.90*		
6. Gypsum lath (⅜ in.)	*0.32*	Difference	*1.50*
7. Plas. (sand agg.) (½ in.)	*0.09*	Add 1. Outside surface (7.5 mph wind)	*0.25*
8. Inside surface (still air)	*0.62*	5. Air space	*0.69*
		8. Inside surface (still air)	*0.76* 1.90
Total resistance	*3.19*		
$U = 1/R = 1/3.19$	0.31	Total resistance	*3.40*
See value 0.31 in boldface type in table below.		$U = 1/R = 1/3.40$	0.29

To Adjust U Values for Construction with Added Insulation between Framing Members, See Table 16

Direction of Heat Flow →		Upward Flow Winter Conditions					Downward Flow Summer Conditions					
		Rafter Space					Rafter Space					
		Unventilated, Not to be Further Insulated				Insulated	Unventilated, Not to be Further Insulated				Insulated	
Type of Ceiling (Applied Directly to Roof Rafters)		Asphalt shingles building paper		Asbestos-cement slate or tile shingles, building paper on wood sheathing ($^{25}/_{32}$ in.)	Wood shingles on 1 × 4 in. wood strips on 6-in. centers	Roof and sheathing disregarded[d]	Asphalt shingles building paper		Asbestos-cement slate, or the shingles, building paper on wood sheathing ($^{25}/_{32}$ in.)	Wood shingles on 1 × 4 in. wood strips on 6 in. centers	Roof and sheathing disregarded[d]	Number
		On plywood sheathing (⅜ in.)	On wood sheathing ($^{25}/_{32}$ in.)				On plywood sheathing (⅜ in.)	On wood sheathing ($^{25}/_{32}$ in.)				
Resistance →		0.95	1.48	1.09	0.87	—	0.95	1.48	1.09	0.87	—	
Material	*R*	*U*	*U*	*U*	*U*	*U*	*U*	*U*	*U*	*U*	*U*	
		A	B	C	D	E	F	G	H	I	J	
None	—	0.57	0.44	0.53	0.60	0.66	0.51	0.40	0.48	0.53	0.56	1
Gypsum bd. (⅜ in.)	*0.32*	0.34	0.29	0.32	0.35	0.54	0.30	0.26	0.29	0.31	0.47	2
Gypsum lath (⅜ in.) & ½ in. plas. (lt. wt. agg.)	*0.64*	0.30	0.26	0.29	0.31	0.46	0.28	0.24	0.27	0.28	0.41	3
Gypsum lath (⅜ in.) and ½ in. plas. (sand agg.)	*0.41*	0.33	0.28	0.31	0.34	0.52	0.29	0.25	0.28	0.30	0.45	4
Metal lath and ¾ in. plas. (lt. wt. agg.)	*0.47*	0.32	0.27	0.31	0.33	0.50	0.29	0.25	0.28	0.30	0.44	5
Metal lath and ¾ in. plas. (sand agg.)	*0.13*	0.36	0.30	0.34	0.37	0.61	0.32	0.27	0.31	0.33	0.52	6
Insul. bd. (½ in.)	*1.43*	0.25	0.22	0.24	0.25	0.34	0.23	0.20	0.22	0.23	0.31	7
Insul. bd. lath and ½ in. plas. (sand agg.)	*1.68*	0.24	0.21	0.23	0.25	0.33	0.22	0.20	0.22	0.23	0.30	8
Acoustical tile												
(½ in.) on gypsum bd. (⅜ in.)	*1.51*	0.24	0.21	0.23	0.25	0.33	0.22	0.20	0.22	0.23	0.30	9
(½ in.) on furring	*1.19*	0.26	0.23	0.25	0.27	0.37	0.24	0.21	0.23	0.24	0.34	10
(¾ in.) on gypsum bd. (⅜ in.)	*2.10*	0.21	0.19	0.20	0.21	0.26	0.20	0.18	0.19	0.20	0.26	11
(¾ in.) on furring	*1.78*	0.23	0.20	0.22	0.23	0.30	0.21	0.19	0.20	0.21	0.28	12
Wood lath and ½ in. plas. (sand agg.)	*0.40*	0.33	0.28	0.31	0.34	0.52	0.29	0.26	0.28	0.30	0.46	13

[a] See text section Calculating Overall Coefficients for basis of calculations.
[b] Pitch of roof—45 deg.
[c] To adjust *U* values for the effect of added insulation between framing members, see Table 16.
[d] When insulation is installed between rafters, the space above should be ventilated and in this case the roof construction is disregarded in calculation of *U* values. To adjust *U* values for pitched roof construction with added insulation between framing members, use the average of values for horizontal and vertical heat flow in Table 16A and either 16C or 16D depending on direction of heat flow.

Table 16 .... Determination of U Value Resulting from Addition of Insulation or Air Spaces to Uninsulated Building Sections*

PART A. WALLS[a]

U Value[b] Without Added Insulation	Fibrous Insulation[c] Thickness—Inches				One Air Space of Effective Emissivity[d] E			Two Air Spaces of Effective Emissivity[d] E			Three Air Spaces of Effective Emissivity[d] E		
	½	1	2	3	0.82[e]	0.20	0.05	0.82	0.20	0.05	0.82	0.20	0.05
Col. 1	2	3	4	5	6	7	8	9	10	11	12	13	14
0.70	0.304	0.194	0.113	0.080	0.752	0.463	0.380	0.437	0.240	0.189	0.298	0.150	0.112
0.60	0.284	0.186	0.110	0.078	0.630	0.412	0.341	0.392	0.225	0.177	0.276	0.144	0.108
0.45	0.246	0.168	0.104	0.075	0.460	0.331	0.280	0.318	0.195	0.158	0.237	0.130	0.098
0.40	0.230	0.161	0.101	0.074	0.409	0.299	0.258	0.291	0.185	0.149	0.222	0.125	0.095
0.35	0.212	0.152	0.097	0.072	0.354	0.267	0.234	0.262	0.172	0.140	0.205	0.119	0.092
0.30	0.192	0.142	0.093	0.069	0.30	0.234	0.207	0.232	0.158	0.130	0.186	0.112	0.087
0.28	0.184	0.138	0.091	0.068	0.28	0.221	0.196	0.220	0.151	0.125	0.178	0.108	0.084
0.26	0.175	0.133	0.089	0.066	0.26	0.208	0.185	0.207	0.144	0.120	0.169	0.104	0.082
0.24	0.166	0.127	0.087	0.065	0.24	0.194	0.173	0.194	0.137	0.115	0.160	0.100	0.079
0.22	0.156	0.121	0.084	0.064	0.22	0.180	0.161	0.180	0.129	0.110	0.150	0.096	0.076
0.20	0.145	0.115	0.081	0.062	0.20	0.165	0.149	0.166	0.120	0.104	0.140	0.091	0.073
0.18	0.134	0.108	0.078	0.060	0.18	0.150	0.137	0.152	0.112	0.098	0.129	0.086	0.069
0.16	0.123	0.100	0.074	0.057	0.16	0.136	0.124	0.137	0.103	0.090	0.118	0.080	0.065
0.14	0.111	0.092	0.069	0.054	0.14	0.120	0.111	0.122	0.094	0.083	0.106	0.075	0.061
0.12	0.098	0.083	0.064	0.051	0.12	0.105	0.098	0.107	0.084	0.075	0.094	0.068	0.056
0.10	0.085	0.073	0.058	0.047	0.10	0.089	0.084	0.091	0.074	0.066	0.082	0.061	0.051
0.08	0.070	0.062	0.050	0.042	0.08	0.073	0.068	0.074	0.062	0.056	0.068	0.053	0.045

[a] For constructions with air spaces as insulation, coefficients are based on National Bureau of Standards data in *Housing Research Paper* No. 32 (U. S. Government Printing Office, Washington, D. C.).
[b] Based on an indoor-outdoor temperature difference of 70 F deg, and a mean temperature of 50 F. Values are applicable conservatively to winter and summer conditions.
[c] *U* value taken from Tables 5 to 15, based on one nonreflective 3¾-in. air space between framing members.
[d] Thermal conductivity of fibrous or bulk insulation taken as 0.27 Btu per (hr) (sq ft) (F deg per in.).
[e] For values of effective emissivity E of air space, see Table 3, Section B.
[f] Certain *U* values in Column 6 differ from Column 1 because they are adjusted to the specific temperature drop across the air space in question as affected by the *U* value of the construction.

PART B. FLOORS[f]—HEAT FLOW DOWN

U Value[a] Without Added Insulation	Fibrous Insulation[b] Thickness—Inches				One Air Space of Effective Emissivity[i] E			Two Air Spaces of Effective Emissivity[i] E			Three Air Spaces of Effective Emissivity[i] E		
	½	1	2	3	0.82	0.20	0.05	0.82	0.20	0.05	0.82	0.20	0.05
Col. 1	2	3	4	5	6	7	8	9	10	11	12	13	14
0.70	0.305	0.195	0.113	0.080	0.70	0.240	0.114	0.377	0.122	0.057	0.262	0.086	0.042
0.60	0.284	0.186	0.110	0.078	0.60	0.236	0.111	0.346	0.118	0.056	0.246	0.084	0.041
0.50	0.260	0.175	0.106	0.076	0.50	0.210	0.106	0.310	0.114	0.055	0.228	0.082	0.041
0.45	0.246	0.168	0.104	0.075	0.45	0.200	0.103	0.290	0.111	0.055	0.217	0.081	0.040
0.40	0.230	0.161	0.101	0.074	0.40	0.189	0.100	0.268	0.107	0.054	0.205	0.079	0.040
0.35	0.212	0.152	0.097	0.072	0.35	0.176	0.096	0.244	0.103	0.052	0.192	0.077	0.040
0.30	0.192	0.142	0.093	0.069	0.30	0.162	0.091	0.219	0.098	0.051	0.175	0.074	0.039
0.28	0.184	0.138	0.091	0.068	0.28	0.156	0.089	0.208	0.096	0.050	0.168	0.073	0.039
0.26	0.175	0.133	0.089	0.066	0.26	0.150	0.087	0.197	0.094	0.049	0.161	0.072	0.038
0.24	0.166	0.127	0.087	0.065	0.24	0.143	0.084	0.185	0.091	0.048	0.153	0.070	0.038
0.22	0.156	0.121	0.084	0.064	0.22	0.136	0.081	0.173	0.088	0.047	0.145	0.068	0.037
0.20	0.145	0.115	0.081	0.062	0.20	0.128	0.078	0.160	0.084	0.046	0.136	0.066	0.036
0.18	0.134	0.108	0.078	0.060	0.18	0.119	0.074	0.148	0.080	0.045	0.126	0.064	0.036
0.16	0.123	0.100	0.074	0.057	0.16	0.109	0.070	0.133	0.076	0.044	0.116	0.061	0.035
0.14	0.111	0.092	0.069	0.054	0.14	0.099	0.065	0.118	0.071	0.042	0.105	0.058	0.034
0.12	0.098	0.083	0.064	0.051	0.12	0.088	0.060	0.103	0.066	0.040	0.094	0.054	0.033
0.10	0.085	0.073	0.058	0.047	0.10	0.076	0.054	0.089	0.058	0.037	0.081	0.049	0.031
0.08	0.070	0.062	0.050	0.042	0.08	0.063	0.047	0.072	0.050	0.033	0.068	0.044	0.028

[a] For construction with air spaces as insulation, coefficients are based on National Bureau of Standards data in *Housing Research Paper* No. 32 (U. S. Government Printing Office, Washington, D. C.).
[b] Based on a temperature difference of 50 F deg from air to air, and a mean temperature of 50 F.
[c] *U* value taken from Tables 11 and 12, in which it is assumed that the air space between joists or above the suspended ceiling is nonreflective (E = 0.82), and is 8 in. thick.
[d] Thermal conductivity of fibrous or bulk insulation taken as 0.27 Btu per (hr) (sq ft) (F deg per in.).
[e] For values of effective emissivity E of air space, see Table 3, Section B.

Table 16 . . . . Determination of *U* Value Resulting from Addition of Insulation or Air Spaces to Uninsulated Building Sections* (*Continued*)

PART C. CEILINGS—HEAT FLOW UP[j] *(WINTER CONDITION)*

U Value[k] Without Added Insulation	Fibrous Insulation[l] Thickness—Inches				One Air Space of Effective Emissivity[m] E			Two Air Spaces of Effective Emissivity[m] E			Three Air Spaces of Effective Emissivity[m] E		
	½	1	2	3	0.82[n]	0.20	0.05	0.82	0.20	0.05	0.82	0.20	0.05
Col. 1	2	3	4	5	6	7	8	9	10	11	12	13	14
0.70	0.305	0.195	0.113	0.080	0.690	0.472	0.403	0.427	0.262	0.216	0.307	0.180	0.146
0.60	0.284	0.186	0.110	0.078	0.588	0.417	0.362	0.385	0.244	0.204	0.284	0.171	0.139
0.50	0.260	0.175	0.106	0.076	0.488	0.361	0.318	0.339	0.224	0.189	0.258	0.160	0.131
0.45	0.246	0.168	0.104	0.075	0.438	0.331	0.295	0.316	0.212	0.180	0.243	0.154	0.126
0.40	0.230	0.161	0.101	0.074	0.389	0.300	0.270	0.288	0.199	0.170	0.227	0.146	0.121
0.35	0.212	0.152	0.097	0.072	0.340	0.269	0.244	0.260	0.185	0.158	0.209	0.138	0.115
0.30	0.192	0.142	0.093	0.069	0.292	0.237	0.215	0.230	0.168	0.145	0.189	0.129	0.108
0.28	0.184	0.138	0.091	0.068	0.272	0.224	0.203	0.217	0.161	0.140	0.181	0.125	0.104
0.26	0.175	0.133	0.089	0.066	0.253	0.211	0.191	0.204	0.154	0.134	0.172	0.120	0.101
0.24	0.166	0.127	0.087	0.065	0.234	0.199	0.179	0.191	0.146	0.128	0.163	0.115	0.097
0.22	0.156	0.121	0.084	0.064	0.214	0.186	0.166	0.178	0.137	0.120	0.153	0.109	0.093
0.20	0.145	0.115	0.081	0.062	0.195	0.173	0.154	0.164	0.128	0.114	0.143	0.104	0.088
0.18	0.134	0.108	0.078	0.060	0.176	0.159	0.141	0.150	0.119	0.106	0.132	0.097	0.084
0.16	0.123	0.100	0.074	0.057	0.156	0.146	0.128	0.136	0.109	0.098	0.121	0.090	0.079
0.14	0.111	0.092	0.069	0.054	0.137	0.132	0.115	0.120	0.099	0.090	0.109	0.083	0.073
0.12	0.098	0.083	0.064	0.051	0.118	0.118	0.101	0.105	0.088	0.080	0.096	0.075	0.068
0.10	0.085	0.073	0.058	0.047	0.099	0.105	0.088	0.090	0.076	0.071	0.082	0.067	0.062
0.08	0.070	0.062	0.050	0.042	0.079	0.091	0.074	0.073	0.064	0.061	0.067	0.058	0.056

* For construction with air spaces as insulation, coefficients are based on National Bureau of Standards data in *Housing Research Paper* No. 32 (U. S. Government Printing Office, Washington, D. C.).

[j] Based on a temperature difference of 75 deg F from air to air, and a mean temperature of 40 F.

[k] *U* value taken from Tables 11, 12, 13-A, 14-A, and 15, in which it is assumed that the air space between joists or above the suspended ceiling is nonreflective (E = 0.82), and is 8 in. thick.

[l] Thermal conductivity of fibrous or bulk insulation taken as 0.27 Btu per (hr) (sq ft) (F deg per in.).

[m] For values of effective emissivity E of air space, see Table 3, Section B.

[n] Certain *U* values in Column 6 differ from Column 1 because they are adjusted to the specific temperature drop across the air space in question as affected by the *U* value of the construction.

PART D. CEILINGS—HEAT FLOW DOWN[o] *(SUMMER CONDITION)*

U Value[p] Without Added Insulation	Fibrous Insulation[q] Thickness— Inches				One Air Space of Effective Emissivity[r] E			Two Air Spaces of Effective Emissivity[r] E			Three Air Spaces of Effective Emissivity[r] E		
	½	1	2	3	0.82	0.20	0.05	0.82	0.20	0.05	0.82	0.20	0.05
Col. 1	2	3	4	5	6	7	8	9	10	11	12	13	14
0.70	0.305	0.195	0.113	0.080	0.704	0.269	0.119	0.423	0.144	0.061	0.306	0.103	0.046
0.60	0.284	0.186	0.110	0.078	0.602	0.252	0.116	0.384	0.139	0.060	0.284	0.100	0.046
0.50	0.260	0.175	0.106	0.076	0.501	0.231	0.112	0.340	0.133	0.059	0.260	0.097	0.045
0.45	0.246	0.168	0.104	0.075	0.450	0.220	0.108	0.316	0.128	0.058	0.246	0.095	0.044
0.40	0.230	0.161	0.101	0.074	0.400	0.206	0.105	0.291	0.123	0.057	0.230	0.092	0.044
0.35	0.212	0.152	0.097	0.072	0.350	0.192	0.100	0.264	0.118	0.056	0.213	0.090	0.043
0.30	0.192	0.142	0.093	0.069	0.300	0.175	0.095	0.234	0.112	0.055	0.193	0.086	0.042
0.28	0.184	0.138	0.091	0.068	0.280	0.168	0.093	0.222	0.109	0.054	0.185	0.085	0.042
0.26	0.175	0.133	0.089	0.066	0.260	0.160	0.090	0.209	0.105	0.054	0.176	0.083	0.041
0.24	0.166	0.127	0.087	0.065	0.240	0.152	0.087	0.195	0.102	0.053	0.167	0.080	0.040
0.22	0.156	0.121	0.084	0.064	0.220	0.144	0.084	0.182	0.098	0.052	0.157	0.078	0.040
0.20	0.145	0.115	0.081	0.062	0.200	0.135	0.081	0.168	0.094	0.050	0.146	0.075	0.039
0.18	0.134	0.108	0.078	0.060	0.180	0.125	0.077	0.154	0.089	0.049	0.135	0.072	0.038
0.16	0.123	0.100	0.074	0.057	0.160	0.115	0.073	0.139	0.084	0.047	0.124	0.068	0.038
0.14	0.111	0.092	0.069	0.054	0.140	0.104	0.069	0.124	0.078	0.045	0.111	0.064	0.037
0.12	0.098	0.083	0.064	0.051	0.120	0.093	0.063	0.108	0.072	0.042	0.098	0.059	0.035
0.10	0.085	0.073	0.058	0.047	0.099	0.080	0.056	0.091	0.064	0.039	0.084	0.054	0.033
0.08	0.070	0.062	0.050	0.042	0.079	0.068	0.048	0.074	0.054	0.034	0.070	0.046	0.030

* Based on a temperature difference of 25 F deg from air to air, and a mean temperature of 100 F.

[p] *U* value taken from Tables 11, 12, 13-B, 14-B, and 15, in which it is assumed that the air space between joists or above the suspended ceiling is nonreflective (E = 0.82), and is 8 in. thick.

[q] Thermal conductivity of fibrous or bulk insulation taken as 0.27 Btu per (hr) (sq ft) (F deg per in.).

[r] For values of effective emissivity E of air space, see Table 3, Section B.

Table 16 . . . . Determination of *U* Value Resulting from Addition of Insulation to Uninsulated Building Sections (*Concluded*)

(For use with Tables 13A and 13B)

PART E. FLAT ROOFS AND CEILINGS WITH ROOF DECK

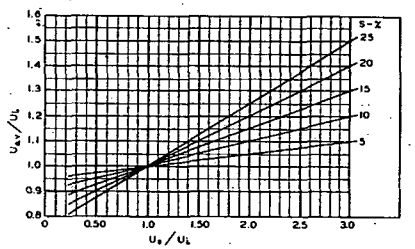
U Value of Roof without Roof-Deck Insulation*	Conductance C of Roof-Deck Insulation					
	0.12	0.15	0.19	0.24	0.36	0.72
	U	*U*	*U*	*U*	*U*	*U*
0.10	0.05	0.06	0.07	0.07	0.08	0.09
0.15	0.07	0.08	0.08	0.09	0.11	0.12
0.20	0.08	0.09	0.10	0.11	0.13	0.16
0.25	0.08	0.09	0.11	0.12	0.15	0.19
0.30	0.09	0.10	0.12	0.13	0.16	0.21
0.35	0.09	0.10	0.12	0.14	0.18	0.24
0.40	0.09	0.11	0.13	0.15	0.19	0.26
0.50	0.10	0.12	0.14	0.16	0.21	0.29
0.60	0.10	0.12	0.14	0.17	0.22	0.33
0.70	0.10	0.12	0.15	0.18	0.24	0.35

* Interpolation or mild extrapolation may be used.

CORRECTION FOR FRAMING

Correction for parallel heat flow through framing and insulated areas may be made by use of Fig. 6. Correction for the effect of framing should be applied after final U_i and U_o values have been obtained for a given construction. In many cases this correction may be omitted.

Example 7: Consider a frame wall with 2-in. blanket insulation which has a U_i value of 0.08. By calculation it is found that heat loss from the area backed by framing members (U_s) is 0.13. U_s/U_i is 1.63. From Fig. 6 if 15 percent of wall area is backed by framing, the value $U_{ai}/U_i = 1.1$. U_{ai} is therefore $1.1 \times 0.08 = 0.088$.



U_{av} = average *U* value for building section.
U_i = *U* value for area between framing members.
U_s = *U* value for area backed by framing members.
S = percentage of area backed by framing members.

Fig. 6 . . . . Correction for Effect of Framing in Insulated Building Sections

VENTILATED ATTICS—HOW TO USE TABLE 17

Table 17 is intended to be used with Table 11, Part D of Table 16, and Fig. 6, or when ceiling resistance is known. Its purpose is to determine the resistance to heat flow of the attic space under various conditions of ventilating air temperatures and rates, ceiling resistance, roof or sol-air temperatures, and surface emissivities.[11] Ventilating air temperature is the outdoor design temperature.

The total resistance, $1/U$, obtained by adding the ceiling and attic resistances can be converted to a U value so that the heat gain may be calculated. The applicable temperature difference is that difference between room air and sol-air temperature or between room air and roof temperatures. (See footnote d, Table 17.)

Table 17 may be used for both pitched and flat residential roofs over attic spaces. When there is an attic floor, the ceiling resistance should be that which applies to the complete ceiling-floor construction.

All values in Table 16, Part D, include the resistance of a non-reflective surface facing the attic space. Therefore, if separate calculations are made, include only the value of a non-reflective surface and not the value of a reflective surface in determining ceiling resistance. The use of Table 17, Part B, will account for this reflection surface.

Example 8: Determine the heat gain for a 1000 sq ft ceiling of ⅜-in. gypsum board and ½-in. light weight aggregate plaster, with no flooring above, when insulated with a 2-in. foil-enclosed fibrous blanket. The blanket is installed so as to form a reflective air space between the ceiling and the blanket. The attic has a gable roof which meets the ventilation requirements in Table 3, Chapter 10. (Use ventilation rate of 0.1 cfm per sq ft.)

Design temperatures are: indoor air = 75 F, outdoor air = 95 F, and sol-air = 160 F.

Solution: From Table 11, for heat flow down, the U value for this ceiling G3 without insulation is 0.40. Referring to Table 16, Part D, with the value 0.40 in Column 1, find U_i = 0.101 in Column 4; enter Column 1 at 0.101 and find U_i = 0.056 in Column 8. Correct for framing (8 in. joists on 16 in. centers) from Fig. 6 and find corrected ceiling coefficient U_c = 0.059; R_c = 17.0.

By interpolation in Table 17, Part B, using 95 F ventilation air and 160 F sol-air temperatures, the effective attic resistance R_a is 8.2.

The overall coefficient for the combined ceiling and attic is:

$$U_o = \frac{1}{R_c + R_a} = \frac{1}{17.0 + 8.2} = 0.04$$

Heat gain = $U_o A (t_o - t_i) = 0.04 \times 1000 \times (160-75) = 3400$ Btuh

COMBINED CEILING AND ROOF COEFFICIENTS

If the attic space between the ceiling and roof is unheated and *not ventilated*, the combined coefficient from room air below the ceiling to exterior air can be calculated from the following formula:

$$R_T = \frac{1}{U_{ce}} + \frac{1}{nU_r} \qquad (4)$$

The combined coefficient U is the reciprocal of R_T, or

$$U = 1/R_T \qquad (5)$$

where

U = combined coefficient to be used with ceiling area.
R_T = total resistance of ceiling and roof.
U_{ce} = coefficient of transmission of ceiling.
U_r = coefficient of transmission of roof.
n = ratio of roof area to ceiling area.

It should be noted that the overall coefficient U should be multiplied by the ceiling area to determine heat loss, and not

Table 17 .... Effective Resistance of Ventilated Attics[a]—(Summer Condition)[11]

PART A. NON-REFLECTIVE SURFACES

Ventilation Air temp., F	Sol-air[d] temp., F	No Ventilation		Natural Ventilation		Power Ventilation[e]					
		Ventilation rate, cfm/ sq ft									
		0		0.1[b]		0.5		1.0		1.5	
		1/U Ceiling resistance, r_u[c]									
		10	20	10	20	10	20	10	20	10	20
80	120	1.9	1.9	2.8	3.4	6.3	9.3	9.6	16	11	20
	140	1.9	1.9	2.8	3.5	6.5	10	9.8	17	12	21
	160	1.9	1.9	2.8	3.6	6.7	11	10	18	13	22
90	120	1.9	1.9	2.5	2.8	4.6	6.7	6.1	10	6.9	13
	140	1.9	1.9	2.6	3.1	5.2	7.9	7.6	12	8.6	15
	160	1.9	1.9	2.7	3.4	5.8	9.0	8.5	14	10	17
100	120	1.9	1.9	2.2	2.3	3.3	4.4	4.0	6.0	4.1	6.9
	140	1.9	1.9	2.4	2.7	4.2	6.1	5.8	8.7	6.5	10
	160	1.9	1.9	2.6	3.2	5.0	7.6	7.2	11	8.3	13
PART B. REFLECTIVE SURFACES[f]											
80	120	6.5	6.5	8.1	8.8	13	17	17	25	19	30
	140	6.5	6.5	8.2	9.0	14	18	18	26	20	31
	160	6.5	6.5	8.3	9.2	15	18	19	27	21	32
90	120	6.5	6.5	7.5	8.0	10	13	12	17	13	19
	140	6.5	6.5	7.7	8.3	12	15	14	20	16	22
	160	6.5	6.5	7.9	8.6	13	16	16	22	18	25
100	120	6.5	6.5	7.0	7.4	8.0	10	8.5	12	8.8	12
	140	6.5	6.5	7.3	7.8	10	12	11	15	12	16
	160	6.5	6.5	7.6	8.2	11	14	13	18	15	20

[a] The term *effective resistance* is used when there is attic ventilation. A value for no ventilation is also included. The effective resistance of the attic may be added to the resistance (1/*U*) of the ceiling (Table 16, Part D) to obtain the effective resistance of the combination based on sol-air (Chapter 13) and room temperature. These values apply to wood frame construction with a roof deck and roofing having a conductance of 1.0 Btu/(sq ft) (hr) (F deg).

[b] When attic ventilation meets the requirements of Table 3 in Chapter 10, 0.1 cfm/sq ft may be assumed as the natural summer ventilation rate for design purposes.

[c] *Resistance Unit*, abbreviated *ru* is one (hr) (sq ft) (F deg) per Btu. Determine ceiling resistance from Tables 11 and 16, and correct for framing by Figure 6. Do not add the effect of a reflective surface facing the attic to the ceiling resistance from Table 16, Part D, as it is accounted for in Table 17, Part B.

[d] Roof surface temperature rather than sol-air temperature (see Chapter 13) may be used if 0.25 is subtracted from the attic resistance shown.

[e] Based on air discharging outward from attic.

[f] Surfaces with effective emissivity E of 0.05 between ceiling joists facing the attic space.

by the roof area. Values of U_r and U_{ce} should be calculated using a value of 2.5 (the reciprocal of one-half the air space resistance, 0.80) rather than the conductances of surfaces facing the attic, since the attic is assumed to be equivalent to an air space.

If the attic contains windows, dormers, and vertical wall spaces, and if their area is small compared to that of the roof, they may be considered part of the roof area. For accuracy, the sum of the coefficients of each individual section, multiplied by its percentage of the total area, should be used as U_r. Where attic wall areas are large or where louvers or vents are used, it is preferable to estimate the attic temperature as illustrated in Chapter 12, and calculate the heat loss through the ceiling by multiplying the value of U_{ce} for the ceiling by the difference in temperature above and below the ceiling.

BASEMENT FLOOR, BASEMENT WALL, AND CONCRETE SLAB FLOOR COEFFICIENTS

The heat transfer through basement walls and floors to the ground is dependent on the temperature difference between the air within and that of the ground, on the material constituting the wall or floor, and on the conductivity of the surrounding earth. The conductivity of the earth will vary with local conditions, and is usually unknown. Tests[12] at the ASHAE Research Laboratory indicate a heat flow of approximately 2.0 Btu per (hr) (sq ft) through an uninsulated concrete basement floor, with a temperature difference of 20 deg between ground temperature and the air temperature 6 in. above the floor (see Table 18).

For basement walls below grade only, the temperature difference for winter design conditions will be greater than for the floor. The test results indicate a unit area heat loss, at mid-height of the basement wall portion below grade approximately twice that of the same floor area.

For concrete slab floors laid in contact with the ground at grade level, tests[13] indicate that for *small floor areas* (equal to

Table 18 .... Coefficients of Transmission (*U*) of Concrete Basement Floors on Ground with Various Types of Finish Flooring

$U = 0.10^a$ Btu per (hr) (sq ft) (Fahrenheit degree temperature difference between the ground and the air over the floor).

[a] Since authentic data are not available, this coefficient is sometimes used for concrete floors on ground. For more recent procedures[13] refer to *National Bureau of Standards Report* BMS-103.

that of a house 25 feet square) the heat loss may be calculated as proportional to the length of exposed edge rather than total area. This amounts to 0.81 Btu per (hr) (linear foot of exposed edge) (Fahrenheit degree difference between the indoor air temperature and the average outdoor air temperature). It should be noted that this may be appreciably reduced by insulating under the ground slab, and also along the edges between the floor and the abutting walls. See also sections on Basement Temperatures and Heat Loss, and on Floor Heat Loss in Basementless Houses, in Chapter 12. In most calculations if the perimeter loss is calculated accurately, no other floor loss need be considered.

GLASS AND DOOR COEFFICIENTS

The *U* values for glass sheets and hollow glass block, given in Sections A, B, and C of Table 19, have been computed by methods ind data given in an ASHVE Research Paper.[8] It is assumed that the surface conductance for convection loss to the air is 4.0 Btu per (hr) (sq ft) (F deg). It is also assumed that the glass loses heat by radiation to the ground and to the clear sky, which together have an effective radiating temperature *below* the air temperature. It is therefore necessary to determine, by trial and error, the temperature of the outdoor glass surface such that the sum of the radiation and convection losses equals the heat conducted through the glass section, and equals the heat delivered to the glass from the heated space. This heat flow, divided by the air-to-air temperature difference, results in a *U* value that is used in the usual manner. The equivalent surface conductance for radiation and convection combined, based on *air-to-surface* temperature difference, therefore varies from about 5.5 for single glass to about 6.6 for double glass for exactly the same environmental design conditions. Curtains, draperies, Venetian blinds, etc., will result in lower glass surface temperatures than when the windows are not covered.

It is assumed that the room air temperature equals the average temperature of the room surfaces *seen* by the glass. Special consideration should be given to those cases where the glass *sees* interior surfaces at temperatures differing greatly from the room air temperature, i.e., such cases as in sun rooms, greenhouses, and some panel heated rooms, or where there is an unusual amount of air motion in the vicinity of the glass. Although based on zero outdoor air, the values change only slightly with different design temperatures, being about 5 percent greater for a 30 F outdoor design temperature.

In computing the Table 19 values, consideration of the dependence of the indoor surface conductances upon temperature and direction of heat flow leads to surface conductances averaging about 1.50 for block and vertical glass, and about *1.80* for horizontal glass, as compared to the value of 1.46 used in computing *U* values given in other tables in this chapter. *These values should therefore be used in estimating the temperature at which condensation on glass surfaces will occur.*

The application factors given in Section D of Table 19 are based upon hot box tests summarized in a research bulletin,[14] and are approximate only. In practice, some variation in heat flow through windows having the same ratio of glass to sash area, may be expected because of difference in construction details and in air-space edge effects. The high conductance of aluminum and steel sash must be taken into consideration where excessive amounts of metal sash and frames are involved. This is particularly important when they are in close proximity to radiation heat sources and are consequently subjected to high differential temperatures.

Table 19 .... Coefficients of Transmission (*U*) of Windows, Skylights and Glass Block Walls

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides). Those for outdoor exposures are based upon the following outdoor conditions: 0 F air temperature, clear skies, no solar radiation, and 15 mph outdoor wind velocity

SECTION A—VERTICAL GLASS SHEETS

Number of Sheets	One	Two			Three		
Air space, inches	None	¼	½	1[a]	¼	½	1[a]
Outdoor exposure	1.13	0.61	0.55	0.53	0.41	0.36	0.34
Indoor exposure	0.75	0.50	0.46	0.45	0.38	0.33	0.32

SECTION B—HORIZONTAL GLASS SHEETS

Number of Sheets	Heat Flow Up				Heat Flow Down			
	One	Two			One	Two		
Air space, inches	None	¼	½	1[a]	None	¼	½	1[a]
Outdoor exposure	1.40	0.70	0.66	0.63	[b]	[c]	—	—
Indoor exposure	0.96	0.59	0.56	0.56	0.60	0.43	0.39	0.38

SECTION C—WALLS OF HOLLOW GLASS BLOCK

Description	Outdoor Exposure	Indoor Partition
5¾ x 5¾ x 3⅞ in. thick	0.60	0.46
7¾ x 7¾ x 3⅞ in. thick	0.56	0.44
11¾ x 11¾ x 3⅞ in. thick	0.52[d]	0.40
7¾ x 7¾ x 3⅞ in. thick with glass fiber dividing the cavity	0.48	0.38
11¾ x 11¾ x 3⅞ in. thick with glass fiber dividing the cavity	0.44	0.36

SECTION D—APPROXIMATE APPLICATION FACTORS FOR WINDOWS—MULTIPLY FLAT GLASS U VALUES BY THESE FACTORS

Window Description	Single Glass		Double Glass[e]		Windows with Storm Sash[f]	
	Percent Glass[g]	Factor	Percent Glass[g]	Factor	Percent Glass[g]	Factor
Sheets	100	1.00	100	1.00		
Wood sash	80	0.90	80	0.95	80	0.90
Wood sash	60	0.80	60	0.85	60	0.80
Steel sash	80	1.00	80	1.20	80	1.00[h]
Aluminum	80	1.10	80	1.30	80	1.10

[a] For 1 in. or greater.

[b] See Chapter 13, Tables 14 and 15, for summer load.

[c] See Chapter 13, Tables 14, 15, and 16, for summer load.

[d] From unpublished data recommended by ASHAE Tech. Adv. Comm. on Heat Flow Through Fenestration.

[e] Unit type double glazing (two lights or panes in same opening).

[f] Use with *U* values for two sheets with 1 in. air space.

[g] Based on area of exposed portion of sash; does not include frame or portions of sash concealed by frame.

[h] For metal storm sash or metal sash with attached storm pane.

Heat transmission coefficients for wood doors, with and without glass storm doors are given in Table 20.

WIND VELOCITY EFFECT ON *U* VALUES

Tables 5 through 8, 13-A, 14-A, 15, parts of Table 19, and Table 20 show values of U for winter calculations, for an outdoor wind velocity of 15 mph. Tables 13-B and 14-B show values of U for summer calculations and an outdoor wind velocity of 7.5 mph. Tables 11, 12, and 15 show values of U for both winter and summer. Tables 9 through 12 are for indoor U values and are based upon still air. All roof coefficient tables also take into account the direction of heat flow. Care must be exercised in selecting the table which applies to the design conditions. Table 21 shows comparative values of U for other wind velocities. When this table is used for summer

Table 20 .... Coefficients of Transmission (*U*) of Solid Wood Doors

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based upon an outside wind velocity of 15 mph.

Nominal Thickness Inches	*Actual Thickness Inches*	*U*[a,b] *Exposed Door*	*U*[a,b] *With Glass Storm Door*[c]
1	$^{25}/_{32}$	0.64	0.37
1¼	1¼6	0.55	0.34
1⅜	1⅜6	0.49	0.32
1¾	1⅜	0.48	0.31
2	1⅝	0.43	0.28
2¼	2⅛	0.36	0.26
3	2⅝	0.31	0.23

[a] Computed using k = 1.10 for wood, f_i = 1.46, f_o = 6.0, and 1.03 for air space.
[b] A U value of 0.85 may be used for single exposed doors containing thin wood panels or single panes of glass, and 0.39 for the same with glass storm doors.
[c] 50 percent glass and thin wood panels.

U values, it is necessary to enter the table at 7.5 mph which can be interpolated between 5 and 10 mph. Any value taken from this table should be rounded to two significant figures.

Example 8: Find the coefficient of transmission U of a frame wall consisting of stucco, $^{25}/_{32}$-in. insulation board sheathing, 2 x 4-in. studs, gypsum lath and plaster, and with 2-in. blanket insulation between studs, for 25 mph wind velocity.

Solution: From Table 5, this wall, F37, with no insulation between studs has a value of U = 0.22. From Table 16, Part A, Col. 4, this wall with 2-in. insulation added has a value of U = 0.084. Entering Table 21 in the 15 mph column, interpolate between 0.080 and 0.090 in 15 mph column and proceed horizontally to the 25 mph column where the U value is found by interpolation to be 0.085.

CALCULATING SURFACE TEMPERATURES

In many heating and cooling load calculations it is necessary to determine the inside surface temperature or the temperature of the surfaces within the structure. As the resistance of any path of heat flow is expressed in Fahrenheit degrees per (Btu)/(hour) (square foot), the resistances through any two paths of heat flow would be proportional to the temperature drop through these paths, and can be expressed as follows:

$$\frac{R_1}{R_2} = \frac{(t_i - t_x)}{(t_i - t_o)} \qquad (6)$$

Table 21 .... Conversion Table for Wall Coefficient *U* for Various Wind Velocities

U for 15 mph[a]	*U* for 0 to 30 mph Wind Velocities					
	0	5	10	20	25	30
0.050	0.049	0.050	0.050	0.050	0.050	0.050
0.060	0.059	0.059	0.060	0.060	0.060	0.060
0.070	0.068	0.069	0.070	0.070	0.070	0.070
0.080	0.078	0.079	0.080	0.080	0.080	0.080
0.090	0.087	0.089	0.090	0.090	0.091	0.091
0.100	0.096	0.099	0.100	0.100	0.101	0.101
0.110	0.105	0.108	0.109	0.110	0.111	0.111
0.130	0.123	0.127	0.129	0.131	0.131	0.131
0.150	0.141	0.147	0.149	0.151	0.151	0.152
0.170	0.158	0.166	0.169	0.171	0.172	0.172
0.190	0.175	0.184	0.188	0.191	0.192	0.193
0.210	0.192	0.203	0.208	0.212	0.213	0.213
0.230	0.209	0.222	0.227	0.232	0.233	0.234
0.250	0.226	0.241	0.247	0.252	0.253	0.254
0.270	0.241	0.259	0.266	0.273	0.274	0.275
0.290	0.257	0.278	0.286	0.293	0.295	0.296
0.310	0.273	0.296	0.305	0.313	0.315	0.317
0.330	0.288	0.314	0.324	0.333	0.336	0.338
0.350	0.303	0.332	0.344	0.354	0.357	0.359
0.370	0.318	0.350	0.363	0.375	0.378	0.380
0.390	0.333	0.368	0.382	0.395	0.399	0.401
0.410	0.347	0.385	0.402	0.416	0.420	0.422
0.430	0.362	0.403	0.421	0.436	0.441	0.444
0.450	0.376	0.420	0.439	0.457	0.462	0.465
0.500	0.410	0.464	0.487	0.509	0.514	0.518
0.600	0.474	0.548	0.581	0.612	0.620	0.626
0.700	0.535	0.631	0.675	0.716	0.728	0.736
0.800	0.592	0.711	0.766	0.821	0.836	0.847
0.900	0.645	0.789	0.858	0.927	0.946	0.960
1.000	0.695	0.865	0.949	1.034	1.058	1.075
1.100	0.742	0.939	1.039	1.142	1.170	1.192
1.200	0.786	1.010	1.129	1.250	1.285	1.318
1.300	0.828	1.080	1.217	1.359	1.400	1.430

[a] U in first column is from previous tables or as calculated for 15 mph wind velocity.

where

R_1 = the resistance from the indoor air to any point in the structure at which the temperature is to be determined.
R_2 = the overall resistance of the wall from indoor air to outdoor air.
t_i = indoor air temperature.
t_x = temperature to be determined.
t_o = outdoor air temperature.

Example 9: Determine the inside surface temperature for a wall having an overall coefficient of heat transmission U = 0.25, indoor air temperature 70 F, and outdoor air temperature −20 F.

Solution:

$$R = 1/f_i = 1/1.46 = 0.684$$

$$R_2 = 1/U = 1/0.25 = 4.00$$

Then, by Equation 6

$$\frac{0.684}{4.00} = \frac{70 - t_x}{70 - (-20)}$$

$$t_x = 54.6 \text{ F}$$

Example 10: Determine the temperature of the bottom of a 4-in. insulated concrete roof slab to which has been glued ½-in. acoustical tile (C = 0.84) as the interior finish. The roof-ceiling overall coefficient of heat transmission U is 0.14 for heat flow up. The indoor air temperature is assumed to be 70 F and the outdoor air temperature −20 F.

Solution:

$$R_1 = \frac{1}{f_i} + \frac{1}{C} = \frac{1}{1.63} + \frac{1}{0.84} = 1.80$$

$$R_2 = \frac{1}{U} = \frac{1}{0.14} = 7.14$$

Then, by Equation 6

$$\frac{1.80}{7.14} = \frac{70 - t_x}{70 - (-20)}$$

$$t_x = 47.3 \text{ F}$$

The concrete surface temperature is of interest since reference to a psychrometric chart or table will show that moisture condensation could occur on this surface under the above conditions (47.3 F) if the relative humidity in the room exceeds 44 percent. Additional roof insulation should be considered above the slab to avoid condensation at this point if higher relative humidities in the room are anticipated.

The same procedure can be used for determining the temperature it any point within the structure.

A chart for determining inside wall surface temperature is given in Fig. 13 of Chapter 30, Panel Heating.

REFERENCES

[1] *Standard Method of Test for Thermal Conductivity by Means of the Guarded Hot Plate*, sponsored by ASHVE, ASTM, ASRE, and NRC, and approved as a tentative code by ASHVE and ASTM in 1942 (ASTM designation C-177-45, Approved, 1945).

[2] F. B. Rowley and A. B. Algren: *Heat Transmission Through Building Materials* (University of Minnesota, *Engineering Experiment Station Bulletin* No. 8, p. 11).

[3] M. S. Kersten: *Thermal Properties of Soils* (University of Minnesota, *Engineering Experiment Station Bulletin* No. 28, June 1949).

[4] G. B. Wilkes and C. M. F. Peterson: Radiation and convection from surfaces in various positions (ASHVE Transactions, Vol. 44, 1938, p. 513).

[5] B. F. Raber and F. W. Hutchinson: Radiation corrections for basic constants used in the design of all types of heating systems (ASHVE Transactions, Vol. 51, 1945, p. 213).

[6] F. B. Rowley, A. B. Algren, and J. L. Blackshaw: ASHVE Research Report No. 869—Surface conductances as affected by air velocity, temperature and character of surface (ASHVE Transactions, Vol. 36, 1930, p. 444).

[7] G. V. Parmelee and R. G. Huebscher: *Forced Convection Heat Transfer from Flat Surfaces* (ASHVE Research Bulletin No. 3, p. 40, also published in ASHVE Transactions, Vol. 53, 1947, p. 245).

[8] G. V. Parmelee and W. W. Aubele; ASHVE Research Report No. 1399—Heat flow through unshaded glass: Design data for load calculations (ASHVE Transactions, Vol. 56, 1950, p. 371).

[9] H. E. Robinson, F. J. Powlitch, and R. S. Dill: *The Thermal Insulating Value of Airspaces* (Housing and Home Finance Agency, *Housing Research Paper* No. 32, U. S. Government Printing Office, 1954).

[10] G. B. Wilkes, F. G. Heebler, and E. R. Queer: Thermal test coefficients of aluminum insulation for buildings (ASHAE Transactions, Vol. 46, 1940, p. 109).

[11] F. A. Joy: Improving attic space insulating values (ASHAE Journal Section, *Heating, Piping and Air Conditioning*, January 1958, p. 223).

[12] F. C. Houghten, S. I. Taimuty, Carl Gutberlet, and C. J. Brown: ASHVE Research Report No. 1213—Heat loss through basement walls and floors (ASHVE Transactions, Vol. 48, 1942, p. 369).

[13] R. S. Dill, W. C. Robinson, and H. E. Robinson: *Measurements of Heat Losses from Slab Floors* (National Bureau of Standards, *Building Materials and Structures Report BMS* 103).

[14] G. V. Parmelee: *Heat Transmission through Glass* (ASHVE Research Bulletin No. 1, July 1947).

BIBLIOGRAPHY

ASHVE Research Reports:

No. 852—F. B. Rowley, A. B. Algren, and J. L. Blackshaw: Effects of air velocities on surface coefficients (ASHVE Transactions, Vol. 56, 1930, p. 123).

No. 895—F. C. Houghten and Paul McDermott: Wind velocities gradients near a surface and their effect on film conductance (ASHVE Transactions, Vol. 37, 1931, p. 301).

No. 914—F. B. Rowley and W. A. Eckley: Surface coefficients as affected by direction of wind (ASHVE Transactions, Vol. 38, 1932, p. 33).

No. 915—F. C. Houghten and Carl Gutberlet: Conductivity of concrete (ASHVE Transactions, Vol. 38, 1932, p. 47).

No. 964—F. B. Rowley: The heat conductivity of wood at climatic temperature differences (ASHVE Transactions, Vol. 39, 1933, p. 329).

No. 966—F. B. Rowley: Insulating value of bright metallic surfaces (ASHVE Transactions, Vol. 40, 1934, p. 413).

No. 1026—F. B. Rowley, A. B. Algren, and Clifford Carlson: Thermal properties of concrete construction (ASHVE Transactions, Vol. 42, 1936, p. 33).

No. 1048—F. B. Rowley, A. B. Algren, and Robert Lander: Thermal properties of concrete construction (ASHVE Transactions, Vol. 43, 1937, p. 33).

No. 1351—G. V. Parmelee and W. W. Aubele: Overall coefficients for flat glass determined under natural weather conditions (ASHVE Transactions, Vol. 55, 1949, p. 39).

G. B. Wilkes and C. M. F. Peterson: Radiation and convection across air spaces in frame construction (ASHVE Transactions, Vol. 43, 1937, p. 351).

L. W. Schad: Insulating effect of successive air space bounded by bright metallic surfaces (ASHVE Transactions, Vol. 37, 1931, p. 285).

J. D. MacLean: Thermal conductivity of wood (ASHVE Transactions, Vol. 47, 1941, p. 323).

G. B. Wilkes and C. O. Wood: The specific heat of thermal insulating materials (ASHVE Transactions, Vol. 48, 1942, p. 493).

D. B. Anderson: Heat loss studies in four identical buildings to determine the effect of insulation (ASHVE Transactions, Vol. 48, 1942, p. 471).

T. D. Phillips: *Effect of Ceiling Insulation upon Summer Comfort* (*National Bureau of Standards Report* BMS52, July 1, 1940).

L. V. Teesdale: *Thermal Insulation Made of Wood-Base Materials, Its Application and Use in Houses* (*U. S. Forest Products Laboratory Report* No. R1740, October 1949).

F. B. Rowley and A. B. Algren: *Heat Transmission through Building Materials* (University of Minnesota, *Engineering Experiment Station Bulletin* No. 8).

Paul D. Close: *Building Insulation* (American Technical Society, Chicago, 1951, 4th ed.).

CHAPTER 10

MOISTURE IN BUILDING CONSTRUCTION

Properties of Water in Air; Water in Building Materials; Vapor Transmission; Permeance and Testing; Visible Condensation; Concealed Condensation in Heated Buildings; Control of Concealed Condensation; Condensation in Cooled Structures.

THE behavior of moisture is too often overlooked or given scant attention in the design and construction of buildings. It is present as a vapor in all air and as adsorbed moisture in most building materials. It may be present also at times in the free liquid state or as ice, in the solid state, within the range of temperatures encountered in many buildings. Problems involving moisture may arise from changes in moisture content, from the presence of excessive moisture, or from effects associated with its changes in state.

Of particular interest is the change from the vapor to the liquid or solid state, known as condensation. This may be associated with a reduction of temperature with time, or may occur as a result of migration of water vapor to regions of lower temperature. Moisture problems involving condensation are therefore most likely to occur in buildings in any climate in which there is a source of water vapor at temperatures above normal, or in cooled structures, and in buildings in cold climates.

Moisture problems in residences occur in winter and become increasingly important as homes are built smaller and tighter. Water vapor originates from such necessary living requirements as cooking, laundering, bathing, and the breathing and perspiration of people. In a typical family of four, the average daily production of water vapor from these sources may be as much as 25 lb, and may be much greater where such appliances as humidifiers, automatic washers, and dryers are used.[1] Another large source of water vapor is sometimes the bare earth in a crawl space or basement. All this water vapor must escape from the dwelling.

PROPERTIES OF WATER VAPOR IN AIR

Water vapor in air is a gas which occupies all the space, along with the air present. In many ways, the water vapor can act independently of the air, since in general its properties do not depend on the presence of the air. It exerts its own vapor pressure, and can move about through air in a space, or move through materials under differences in its own vapor pressure, independently of the air. However, when the air is moved suddenly or is heated or cooled, the water vapor present is similarly affected, so that it is usually necessary to consider it as a part of an air-vapor mixture.

The properties of mixtures of air and water vapor are relatively well known and are discussed in Chapter 3, Thermodynamics. Changes in these properties with heating and cooling can be followed readily with the aid of a psychrometric chart, shown in outline in Fig. 1. The saturation line represents the limiting concentrations of water vapor which can exist as vapor at various temperatures.

A common condition inside buildings, 70 F and 40 percent relative humidity, is represented by point A. This is a condition of partial saturation; i.e., less than 100 percent relative humidity. The vapor pressure of the water present in the air, although not shown on the chart, can be calculated readily from the vapor pressure at saturation and the relative humidity, since relative humidity is very nearly equal to the ratio of the actual vapor pressure to the saturation pressure at the existing temperature.

The increasing relative humidity accompanying cooling from the condition represented by A on the chart to point B can readily be followed. At B, however, at 44.6 F, the relative humidity becomes 100 percent, and the air-vapor mixture is said to be saturated. The temperature at which this particular air-vapor mixture, upon cooling, becomes saturated is its dew-point temperature. Upon further cooling, to 35 F, the original amount of water vapor can no longer be retained and is reduced, in this case, to the condition represented by C, from 0.0633 lb per lb dry air to 0.0427 lb per lb dry air. The process ABC is typical of that which an air-vapor mixture experiences when it comes in contact with a cool window surface. Cooling from B to C results in visible condensation on the glass surface. If the point C were below 32 F, the condensation would be in the form of frost.

Once the temperature drops below the dew point, or frost point if below 32 F, the vapor pressure at the condensing surface is also reduced, thereby establishing a gradient of vapor pressure from the room air to the window surface. This gradient will operate, in conjunction with the convective action within the room, to move water vapor continuously to the window surface to be condensed, so long as the concentration of water vapor in the room is maintained.

A common winter process is that shown by DE, showing air at 20 F, saturated, being heated to 70 F with a resulting large decrease in relative humidity. This explains, in part,

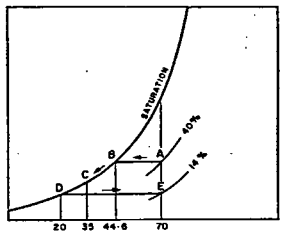


Fig. 1 . . . . Two Typical Heating and Cooling Processes in Air within Buildings Shown on ASHAE Psychrometric Chart

the greatly reduced relative humidities experienced in houses in extreme cold weather, when cold outdoor air enters the house and is heated.

WATER IN BUILDING MATERIALS

The surfaces of most common materials have an affinity for water molecules. Molecular forces of attraction will hold water molecules to the surface, but decrease very rapidly with increase in distance of molecular proportions. The film thickness and therefore the amount of water held in equilibrium with the surrounding atmosphere is roughly proportional to relative humidity. Surface films of water molecules, at low humidities, may be only one molecule thick; at moderate humidities poly-molecular films may be established, while at humidities very close to 100 percent, the films become so thick, relatively, that small pores may become filled and larger capillaries may be partially filled. At saturation conditions all voids in the material may be completely filled.

Some materials such as silica gel, alumina and most natural fibrous materials present very large effective surfaces to the water molecules, so that the amount of water held on the effective surface in these materials may be relatively large, even at moderate humidities. These are said to be hygroscopic. Other materials, such as most metals, not penetrated by the water molecules, present relatively small surfaces and so may take only minute quantities of water, except when wetted directly by liquid.

Substances having a great affinity for water, and their use as dehumidifying agents, are described in Chapter 42. Data on the moisture contents of various common materials in equilibrium with the atmosphere at various relative humidities are given in Table 2 of Chapter 50, and equilibrium moisture content is further discussed in Chapter 53 on Industrial Drying Systems.

Significant dimensional changes take place in many materials used in buildings, with change in moisture content. The best known are those which take place in wood, of the order of 0.1, 2, and 4 percent in the longitudinal, radial and tangential directions, respectively, on a change from air dry at 12 to 15 percent moisture content to oven dry conditions. Most wood-fiber products, including papers, will exhibit moisture expansion consistent with the basic wood properties to a degree dependent on the fiber orientation and arrangement. Data on wood are available in publications on wood technology. Almost all plant and animal fibers experience appreciable moisture changes with changing relative humidity and undergo substantial dimensional changes of the same order as those in wood. Less generally recognized are the dimensional changes that can occur in masonry materials as a result of changes in moisture content.

Water is either an essential or a contributory factor in almost all cases of breakdown of building materials resulting from chemical changes such as the rusting of steel, physical changes such as the spalling of masonry by frost action, or biological processes such as the rotting of wood. The control of water in building constructions may be necessary to ensure adequate service from the materials involved.

Condensation of water vapor, although not the only means by which wetting may be brought about, is nevertheless a most insidious one, particularly in respect to freeze-thaw breakdown, since from its nature it is most likely to occur at points of low temperature at which there may later be risk of freezing while the material remains in a saturated condition.

Moisture in building materials may have a marked effect upon the transmission of heat through them. It has been commonly assumed that moisture when present in a material will remain more or less stationary and will increase the conductivity largely by adding to the path available for heat flow. On this basis, the effect of moisture on heat flow can be accounted for quite simply by the use of suitable coefficients of conductivity in the usual heat-flow equations. The data presented in Chapter 9 on moist soils are of this type.

Evidence to date indicates, however, that in porous materials partially saturated with water there is likely to be a migration of moisture to the cold side under the influence of the temperature gradient. This can occur by a process of evaporation, vapor flow, and condensation within the material, a substantial amount of heat being transferred as latent heat of the vapor, particularly in the case of open fibrous materials. The transmission of heat through moist materials becomes complex whenever conditions are such as to produce any appreciable migration of the moisture, and, consequently, calculations by the usual heat-flow theory alone, are an approximation.

The usual approach to the calculation of moisture migration has been to consider the flow as hydraulic, under the influence of hydrostatic forces when the materials are saturated, and as a vapor flow produced by vapor pressure differences in unsaturated materials. These simple concepts might be adequate were it not for the fact that there are interactions between water molecules and the material through which they are passing, as already mentioned. Further complications may be introduced by the presence of salts and electrical potentials.

It is now recognized that the migration of moisture under conditions of partial saturation in a material having an affinity for water actually occurs as a kind of series-parallel flow of vapor and liquid, with the liquid phase having more and more influence as the moisture content, or the degree of saturation, increases. The two kinds of flow cannot be separated since they are closely coupled everywhere along the flow path by evaporation and condensation. Enough is already known to indicate that the isothermal or constant temperature case of vapor flow under a vapor-pressure gradient is much more manageable than the cases in which there are both temperature and vapor-pressure gradients. Cases of combined heat and moisture flow are now known to be extremely complicated and it is quite clear that when both are occurring, neither one can be adequately dealt with independently of the other. No adequate way of handling the general case theoretically has yet been found, despite efforts being made in many laboratories. The bibliography at the end of this chapter includes some of the more important papers on this subject.

A relatively simple equation for the calculation of water-vapor flow based upon the concept of vapor pressure alone as the driving force, has been in use for a number of years. It can be applied without great difficulty to cases where uniform temperatures or only small temperature gradients exist, and to cases of low or moderate relative humidity. It has also been shown to be useful in other cases, provided that the proper values representative of the conditions to which it is being applied can be found for the flow coefficient to be used in the calculations. The complications inherent in the combined mechanisms of heat and moisture flow are not adequately covered by the variables used in the equation, but appear in the determination of suitable values of the flow coefficient, which, however, may vary greatly for any one material, depending on the conditions of flow.

VAPOR TRANSMISSION THROUGH MATERIALS

The equation presently used in calculating water-vapor transmission through materials is based on a form of Fick's

Law, and is as follows:

$$w = -\mu \frac{dp}{dx} \tag{1}$$

where

w = weight of vapor transmitted through a unit area in unit time.

p = vapor pressure.

x = distance along the flow path.

and hence:

$\frac{dp}{dx}$ = vapor pressure gradient.

μ = permeability.

The close parallel with Fourier's equation for heat flow will be noted. The actual transmission of vapor through a material is extremely complex, so that the coefficient, μ, is not a simple one but is actually a function of relative humidity and temperature, and may vary along the flow path through the material in question.[2]

Integrating Equation 1 from $x = 0$ to $x = l$ and from p_1 to p_2, and rearranging, the following is obtained:

$$w = \frac{\int_{p_1}^{p_2} \mu dp}{p_1 - p_2} \times \frac{(p_1 - p_2)}{l} \tag{2}$$

Let

$$\frac{\int_{p_1}^{p_2} \mu dp}{p_1 - p_2} = \bar{\mu} \tag{3}$$

Then,

$$w = \bar{\mu} \frac{p_1 - p_2}{l} \tag{4}$$

where

l = length of flow path (or thickness of material).

If Equation 1 had been integrated, assuming the coefficient μ to be independent of vapor pressure (and temperature) along the flow path, Equation 4 would have been obtained, but with $\bar{\mu}$ replaced by μ. The coefficient $\bar{\mu}$ is therefore an average permeability coefficient applicable to the varying conditions along the flow path of length l, while the coefficient μ is the *spot* or *differential* permeability.

Equation 4 may be rewritten and units assigned:

$$W = \bar{\mu} A \theta \frac{\Delta p}{l} \tag{5}$$

where,

W = total weight of vapor transmitted, grains.

A = area of cross-section of the flow path, square feet.

θ = time during which the transmission occurred, hours.

Δp = difference of vapor pressure between ends of the flow path, inches of mercury.

l = length of flow path, (or thickness of specimen), inches.

The basic units given are those now favored by the building industry. The permeability μ or $\bar{\mu}$ is therefore expressed in a unit of grains-inches per (square foot) (hour) (inch of mercury vapor pressure difference).

Whenever it is convenient to deal with a material of a stated or implied thickness other than the unit thickness to which μ or $\bar{\mu}$ refer, use may be made of the permeance coefficient M, where $M = \bar{\mu}/l$. The designation *perm* for the unit of permeance is now widely used, and is a convenient substitute for the unit, 1 grain per (square foot) (hour) (inch of mercury vapor pressure difference).[3] The corresponding unit of permeability is *perm-inch*, since it is the permeance of unit thickness. The corresponding flow equation is:

$$W = MA\theta\Delta p \tag{6}$$

Resistance to vapor flow provided by a sheet or board is the reciprocal of the permeance, and correspondingly, the over-all vapor resistance of an assembly (like a wall) of materials in series is the sum of the resistances of its component parts. The over-all permeance of the assembly may be found from the permeances of the individual components[4, 5] in a manner paralleling that used in calculating the over-all coefficient of thermal conductivity from the individual conductances:

$$M_t = \frac{1}{\left(\frac{1}{M_1} + \frac{1}{M_2} + \frac{1}{M_3} \cdots + \frac{1}{M_n}\right)} \tag{7}$$

This simple theory for vapor flow, as in the case of the corresponding simple heat-flow theory, assumes conditions of unidirectional, steady-state flow. Useful calculations can be made for an assembly or sub-assembly for which the inflow and outflow of vapor are equal (a condition at which no condensation occurs) if a permeance applicable to actual conditions can be assigned to each component part. Over-all permeances, vapor pressures and vapor flow can be calculated, and in conjunction with thermal calculations, relative humidities can be determined, and the imminence of condensation predicted. (See *Example 1*, and Fig. 2. See also Chapter 13, Cooling Load, for use of vapor-flow calculations.)

Example 1: A wood frame wall is exposed to indoor conditions 70 F and 50 percent relative humidity (0.37 in. Hg vapor pressure) and outdoor conditions 0 F and 80 percent relative

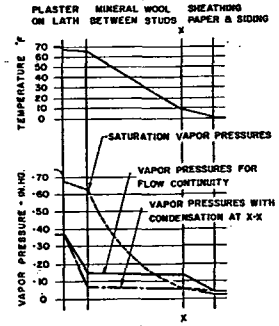


Fig. 2 .... Temperatures and Vapor Pressures under Vapor Flow Conditions in the Insulated Frame Wall of Example 1.

humidity (0.03 in. Hg vapor pressure). The wall consists of painted plaster on gypsum lath on the inside over 2 × 4 in. studs, mineral wool fill between studs, 1 in. wood exterior sheathing, paper, and pine lap siding. Check for possible condensation.

To simplify the example, consider the paint, plaster, and lath as a single element having a permeance $M = 1.0$ perms and a thermal conductance $C = 2.4$, and the exterior sheathing, paper, siding, and paint as another single element for which $M = 2.0$ perms and $C = 0.50$. From Table 1 the value for the permeability of mineral wool fill may be found as $\bar{\mu} = 116$ perm-inches. The thermal conductivity for mineral wool, $k = 0.27$.

Solution: In this particular wall, insulated, and with moderate warm-side relative humidities, condensation is unlikely to occur until the sheathing is reached. To check directly for condensation on the warm side of the exterior sheathing (designated as plane X − X in Fig. 2 for convenient reference) proceed as follows:

Calculate according to the method of Chapter 9 the temperature at plane X − X. *This is found to be 9° F. The saturation* vapor pressure at this temperature is 0.06 in Hg. If condensation is imminent or occurring at X − X, the vapor pressure there will be 0.06 in. Hg. Calculate the permeance for the portion of the wall from the warm side to X − X, and the vapor flow rate to X − X as follows:

$$\text{Permeance of wall to X} - \text{X} = \frac{1}{\frac{1}{1.0} + \frac{3.625}{116}} = 0.97 \text{ perms.}$$

Vapor pressure drop to X − X = 0.37 − 0.06 = 0.31 in. Hg.

Vapor flow to X − X = 0.97 × 0.31 = 0.30 grains per (sq ft) (hr).

Calculate the vapor flow rate from X − X to outdoors as follows:

Permeance of wall from X − X to outdoors = 2.0 perms.

Vapor pressure drop from X − X to outdoors
= 0.06 − 0.03
= 0.03 in. Hg.

Vapor flow rate to outdoors = 2.0 × 0.03 = 0.06 grains per (sq ft) (hr).

It is apparent that continuity of vapor flow is not possible, since at the highest vapor pressure at X − X permitted by the temperature, there is indicated a greater flow to plane X − X than from it to outdoors. Condensation is indicated at a rate of 0.30 − 0.06 = 0.24 grains per (sq ft) (hr).

Whether this condensation rate will be serious must still be decided, since it might readily be absorbed by the sheathing during the condensation period without excessive wetting. When the temperature at X − X is below freezing, as in this case, the condensation will be in the form of frost which may accumulate until released over a short period upon a rise in outdoor temperature.

Condensation will be reduced or avoided if the permeance of the warm side of the wall can be reduced so that the flow to X − X is limited to 0.06 grains per (sq ft) (hr). Permeance required for this is $\frac{0.06}{0.37 - 0.06} = 0.19$ *perms or less. A more* resistant paint film on the plaster, reducing the paint-plaster-lath permeance to 0.19 perms would accomplish this.

When the critical plane for condensation is unknown, or for a more informative, graphical representation of the situation throughout the wall, temperatures and vapor pressures may be calculated and plotted as in Fig. 2. The vapor pressures throughout the wall, for continuity of flow, are calculated in a manner similar to the temperatures; the external vapor pressures are given, and the vapor pressure drops across each element are taken in proportion to resistance to vapor flow.

The curve for saturation vapor pressures at the various *temperatures throughout the wall* is also shown in Fig. 2 and is seen to fall below the curve for vapor pressures with continuity of flow, toward the outer portions of the wall. This indicates that with the given temperatures and vapor pressures, continuity of flow is not possible and that condensation will occur. Condensation on the sheathing is indicated as a definite possibility, and the new vapor-pressure curve can be constructed for this condition, as shown. This new curve does not rise above the saturation-vapor-pressure curve, thereby confirming that the critical plane for condensation was correctly assumed.

With the vapor pressures thus established, the relative humidities may be found, by reference to the saturation vapor pressures. The permeances originally assigned to the various elements may then be re-examined in the light of the service conditions of temperatures and relative humidities indicated, and the analysis repeated, if necessary, using more appropriate *permeance values.*

In a more detailed analysis, individual values might be assigned to the elements forming the outer portion of the *wall which is here dealt with as a composite, homogeneous* element.

The transmission of water vapor as outlined is based on the assumption of a diffusion process. The possibility of vapor being transferred as part of a moving air stream has thus far been ignored, except in *Example 1* in which it is implied that the air circulation on either side of the wall will be sufficient to eliminate surface-film resistances to vapor flow. Differences in total pressure of the air may result in a transfer of vapor with air, augmenting and at times over-riding the effects of the flow produced by vapor-pressure gradients alone. This can be particularly important in the transfer of vapor through cracks and pinholes or through air-permeable building constructions. Similar effects can be obtained with air-permeable materials themselves.[6]

This means of vapor transfer is similar to that of transfer of heat by air leakage in and through building constructions, and requires, for purposes of calculation, information on the nature and amount of the air leakage. It will seldom be important in constructions without air spaces and having parged or plastered surfaces. It may, however, be an important means of vapor transfer through constructions lacking in air tightness, and may contribute to condensation difficulties, since the mechanism of condensation is not dependent upon the way in which the vapor is transferred.

PERMEANCE AND TESTING

The simplest method of finding the permeance of a specimen is to seal it over the top of a cup containing desiccant or water, placing it in a controlled atmosphere, and weighing it periodically. The steady rate of weight gain or loss is normally the water vapor transfer. When the cup contains a desiccant the procedure is called the *dry-cup method* and when the cup contains water, the *wet-cup method.* Usually the surrounding atmosphere is held at 50 percent relative humidity, thus providing, in either method, substantially the same difference of vapor pressure, but the results obtained by the two methods for the same specimens are likely to be much different, the wet-cup method producing the higher values. The relationship between these values can best be understood by reference to Fig. 3, which shows a typical variation of spot permeability with relative humidity at one particular temperature (isothermal conditions) for a material such as wood. The vapor permeability is shown to vary only moderately at low humidities, but to increase at an increasing rate as higher humidities are reached. The dry-cup test of this material carried out with 0 percent relative humidity one side and 50 percent on the other, will experience throughout its thickness, because of the variation in relative humidity, a variation in spot permeability. The average permeability $\bar{\mu}$, is by definition (Equation 3), given by $\frac{\int_{p_1}^{p_2} \mu dp}{p_1 - p_2}$, and since at a fixed temperature there is a linear relationship between vapor pressure and relative humidity, this expression can be seen to correspond to the mean height of the area under the spot permeability curve, between the appropriate relative humidity limits. The average permeability as found for the dry-cup conditions should therefore have the value $\bar{\mu}_1$. Similarly for the wet-cup test between 50 percent and 100 percent relative humidity, the value should be $\bar{\mu}_2$. It is not uncommon for these values for wood and wood-fiber materials to be in the ratio of 1 to 3, or higher. (See Table 1.)

Table 1 .... Permeance and Permeability of Materials to Water Vapor

Material	Permeance Perm	% RH_1-RH_2	Method†	Ref.‡
AIR (still)	120*	92–73	b	4
INSULATION				
Cellular glass	0.0*		d	
Corkboard	2.1–2.6*	75–0	d	5
Corkboard	9.5*	100–45	w	11
Structural Insulating Board (vegetable, uncoated)	20–50*	40–x	t	9
Mineral Wool (unprotected)	116*	100–30	w	8
INTERIOR FINISH				
Plaster on wood lath	11	100–30	w	8
Plaster on metal lath—¾ in.	15	40–x	t	9
Plaster on plain gypsum lath (with studs)	20	40–85	t	4
Gypsum wall board—plain—⅜ in.	50	50–20	v	15
Insulating wall board (uncoated)—½ in.	50–90	40–x	t	9
**PAINT—2 coats				
Asphaltic paint on plywood	0.4	100–30	w	8
Aluminum in varnish on wood	0.3–0.5	95–0	d	12
Enamels, brushed on smooth plaster	0.5–1.5	92–0	b	4
Primers or *Sealers* on insulating wall board	0.9–2.1	40–x	t	9
Various *Primers* + 1 coat flat paint on plaster	1.6–3.	40–x	t	9
Flat paint (alone) on insulating wall board	4	40–x	t	9
Water Emulsion on insulating wall board	30.–85.	40–x	t	9
**PAINT—Exterior, 3 coats				
White *lead & oil* prepared paint on wood siding	0.3–1.0	50–0	d	15
White lead-zinc oxide & linseed oil on wood	0.9	95–0	d	12
WOOD				
Sugar Pine	0.4–5.4*	various	tv	4
Plywood (Exterior type 3 ply D.F.), ¾ in.	0.72	50–	4	13
Plywood (Interior type 3 ply D.F.), ¼ in.	1.86	50–	4	13
MASONRY				
Concrete (1:2:4 Mix)	3.2*	100–45	w	11
Concrete (8 in. cored block wall, limestone agrgt.)	2.4	79–68	t	4
Brick wall—with mortar—4 in.	0.8	50–x	t	10
Tile wall—with mortar—4 in.	0.12	50–x	t	10
	Lb per 500 sq ft	Permeance-Perms: dry cup	Permeance-Perms: wet cup	
**BUILDING PAPERS AND FELTS				
Duplex sheet, asphalt laminae, *aluminum foil* one side	43	0.002	0.176	14
Saturated and coated felt heavy roll roofing	326	0.05	0.24	14
Kraft and *asphalt laminae,* Reinforced 30-120-30	34	0.3	1.8	14
Insulation back up, asphalt-sat., one side glossy	31	0.4	0.6–4.2	14
Asphalt-saturated and coated sheathing paper	43	0.3	0.6	14
Asphalt-saturated sheathing paper	22	3.3	20.2	14
15-pound asphalt felt	70	1.0	5.6	14
15-pound tar felt	70	4.0	18.2	14
Single sheet Kraft, double infused	16	30.8	41.9	14

* These boldface values are permeability in perm-inches.
** Description is a guide only, and does not insure permeance.
† Methods: d—dry cup; w—wet cup; t—two temperatures; b—special cell; v—air velocity both sides; 4—average of four methods.
‡ References. No. 9 also includes *Bulletins* 23 and 25 of the Engineering Experiment Station, University of Minnesota. No. 15 includes data to be published by the Engineering Experiment Station, Pennsylvania State College.

The average permeability $\bar{\mu}$ for any other relative humidities at a particular temperature is given by the mean height of the area under the curve of spot permeability for the material at that temperature, between the appropriate limits of relative humidity. Only average permeabilities (or permeances) are measurable directly in practical tests. However, if several average permeabilities at different relative humidities are known, and can be plotted as for the wet-cup and dry-cup tests shown in Fig. 3, it is possible to construct, by trial and error, a spot permeability curve which will satisfy the condition that the average height of the curve between the appropriate limits for each test must equal the value found in each test. Separate curves are required for each temperature so that a large number of permeance cup tests would be required to cover a range of conditions of both temperature and humidity.

Experience to date shows that the effect of temperature is quite moderate and, for some purposes, differences in temperature at which tests are run may be ignored. Corrections for temperature have been made with some success in the case of certain materials by the use of an equation based on activation energy[7] which states:

$$\mu = \mu_\infty e^{-(E/RT)} \qquad (8)$$

where, in appropriate units,

μ_∞ = permeability at $T = \infty$.
R = gas constant.
E = activation energy.
e = Naperian base of logarithms = 2.718.
T = absolute temperature.

Applications of this equation may be limited to materials such as resins, rubbers, and polymers in which the water molecules enter and move through the molecular structure by a process known as activated diffusion. It may serve also as a useful approximation in other cases.

By use of Equation 8, spot permeability curves for a variety of temperatures can be constructed from a curve for one temperature, provided only that two tests at different temperatures can be run, from which E; the activation energy, can be evaluated. In this way it becomes possible to describe the permeability of a material with reasonable completeness from

the results of as few as five single tests at carefully selected conditions.

No method has yet been found, other than by direct testing for a particular set of conditions, of predicting the effective permeabilities for conditions of temperature gradient along with a humidity (or vapor-pressure) gradient, from data obtained under other conditions, even though this is the general situation in practice. A very considerable amount of information is available on the permeances and permeabilities of various materials, but these are frequently obtained at different conditions so that the results cannot be compared directly, nor do they often give a coverage of a range of conditions to permit construction of basic curves of spot permeability or permeance.

Since dry-cup and wet-cup tests are much easier to carry out than tests at intermediate humidities, many of the data available have been obtained in this way. In the absence of

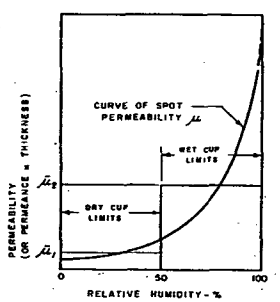


Fig. 3 .... Relation between Dry and Wet Cup Tests and the Spot Permeability for a Material such as Wood

data obtained from tests duplicating the conditions of temperature and relative humidity in service, the permeance of a piece of material may be judged adequately for many purposes if it is tested by both dry and wet methods. It is obvious that any statement of permeance of a specimen should include the conditions of test.

Method of test E96-53T of the American Society for Testing Materials describes procedures for testing for water-vapor transmission of materials in sheet form under five different test conditions. These include the dry-cup and wet-cup methods, based on 50 percent relative humidity outside the cup, at either 73.4 or 90 F, and a fifth condition at an elevated temperature of 100 F with relative humidities of 0 and 90 percent on either side of the sheet. Standard T448m-49 of the Technical Association of the Pulp and Paper Industry covers the determination of water-vapor transmission of paper and paperboard, and calls for the dry-cup method at 73.4 F, based on 50 percent relative humidity outside the cup. *ASTM Standard* C355-54T covers tentative methods of test for water vapor transmission of materials used in building construction.

Unfortunately, there is no general agreement as yet to report results as coefficients of permeance, or of permeability where appropriate, as adopted in this chapter, or to use the same basic units. Test data are frequently reported in terms of weight transmitted per (unit area) (unit time). Such data may be called *water-vapor transmission data* or WVT data and values are either high or low depending on the difference of vapor pressure chosen for the test. When this difference is known, WVT data can be converted to permeance, care being taken if conversion of the basic units (weight, area and time) is also required. The following formula applies:

$$\text{Permeance} = \frac{\text{WVT rating}}{\Delta p} \qquad (9)$$

where

WVT rating = weight of vapor transmitted, grains per (sq ft) (hour).

Δp = vapor-pressure difference in the test, in inches of mercury.

Permeance is expressed in perms.

Table 2 presents the conversion factors applicable to the commonly used units and test methods. Table 1 presents some data on typical building materials showing, in each case, the source and method and, where applicable, the thickness tested.

Water-proofed building papers are listed in *Federal Specifications* UU-P-147, May 24, 1948, according to water-vapor resistance required as:

Class A. For uses where a high degree of water-vapor resistance is required.

Class B. For uses where a lower degree of water-vapor resistance and of water resistance is required.

Class C. For uses where a moderate degree of water-resistance is required.

Class D. For uses where low resistance to water vapor is required.

It may be noted that a paper may be *water-proof*, i.e. possess water resistance, and still have low water-vapor resistance.

Detail requirements in these specifications are given as follows, the specified WVT Test being a dry method at 73 F.

Class A paper shall have a minimum tensile strength in each direction of either 35 lb per inch width, or 20 lb per inch width, as specified in the invitation for bids. Paper of both strengths shall have a minimum water resistance of 24 hr, and a maximum water vapor permeability (WVT) of 4 grams per (square meter) (24 hr) i.e. 0.576 perm.

Class B paper shall have a minimum tensile strength in each direction of either 35 lb per inch width, or 20 lb per inch width, as specified in the invitation for bids. Paper of both strengths shall have a minimum water resistance of 16 hr, and a maximum water-vapor permeability (WVT) of 6 grams per (square meter) (24 hr) i.e. 0.864 perm.

Class C paper shall have a minimum tensile strength in each direction of either 35 lb per inch width, or 20 lb per inch width, as specified in the invitation for bids. Paper of both strengths shall have a minimum water resistance of 8 hr.

Class D paper shall have a minimum tensile strength in each direction of 20 lb per inch width. The paper shall have a minimum water resistance of 10 min., a minimum water-vapor permeability (WVT) of 35 grams per (square meter) (24 hr) i.e. 5.04 perms.

VISIBLE CONDENSATION

Just as moisture collects on the outer surface of a glass of cold water, so does it also condense on other cold materials. In winter, visible condensation may collect on cold closet walls and attic roofs and is commonly observed on frosted window panes. Although condensation, if liquid, may enter an unpainted surface as fast as it forms and thus be unseen, any condensation on a visible surface will for convenience

Table 2 .... Conversion Factors for Vapor Transfer Units

Multiply Number of ... by ... / to Obtain ↓		WVT Units: grams / (24 hrs) (sq m)	WVT Units: grains / (hr) (sq ft)
grams / (24 hrs) (sq m)	by same method*	1	16.7
grains / (hr) (sq ft)	by same method*	0.0597	1
Perms by same method*	method A & B, 73.4 F	0.144	2.41
	method C & D, 90 F	0.0840	1.41
	method E, 100 F	0.0344	0.575

Nominal Test Conditions		% Relative Humidity on the Two Sides of Specimen	
Method	Temperature-F	In cup	Outside cup
A	73.4	0	50
B	73.4	100	50
C	90	0	50
D	90	100	50
E	100	0	90

* Data obtained by one method cannot be reliably converted to another method.

To convert $\frac{\text{grams}}{(24\text{ hr})\ (\text{sq m})\ (\text{mm Hg})}$ to perms multiply by 1.52.

here be called *visible condensation* to distinguish it from *concealed condensation*. Within residences and public buildings, visible condensation in winter may damage decorative finishes and window sash.

Interior visible condensation occurs when any surface is colder than the dew point of the near-by air. The temperature of any such surface—wall, roof, or glass—is dependent upon the air temperature inside and outside the building and the heat transfer coefficient U of the surface structure. Based on a value of 1.65 for the inside surface conductance, Fig. 4 shows the relative humidity in a room at 70 F at which visible condensation will appear at various U values. The curves for single and double glass at their usual U values are included. It should be noted that U values as commonly used are an average for a large area within which there may be spots, such as the studs in an insulated wall, where the transmittance is higher. The inside surface temperature of a wall will, in general, be lower at the bottom due to such things as stratification of inside air and the effects of air leakage and of convection in walls with air spaces. Since condensation appears at any sufficiently cold spot, values from Fig. 4 can be applied only with caution. As a result, the limit of relative humidity for a non-homogeneous wall is lower than might be inferred from its average U value.

Normally, glass is the most likely spot in a heated room for appearance of condensation, but an uninsulated wall surface shielded from radiation by furniture, may be equally cold. In some cases where condensation is barely avoided, the surface, being but little warmer than the dew-point temperature, is exposed to moisture damage—such as swelling, mold or discoloration.

Visible condensation may occur also in summer. It is often seen on basement concrete walls and floors that are cooled by the earth and which being massive, tend to hold a constant temperature from day to day while the weather dew-point temperature rises. When no water vapor is released in the space, the dew point tends to equal that of the outdoors (though it is likely to lag when there is slight ventilation). At times the dew-point temperature rises above the temperature of walls and floors with resulting condensation. If the basement is decorated the trouble may be serious. As an operating problem, the solution may be to reduce ventilation at times of high weather dew point, to warm the walls, or to dehumidify the space. Warming the walls, a slow process at best, is generally accomplished in favorable climates by excess ventilation. In a climate having low temperature at night, it may be feasible to ventilate only at night and thus reduce the moisture content of hygroscopic materials which will then act as desiccants to retard the dew-point rise during the day. In basement walls and floors insulation should be applied in the concrete or on its outside. Insulation placed on the inside of such walls eliminates visible condensation but fos-

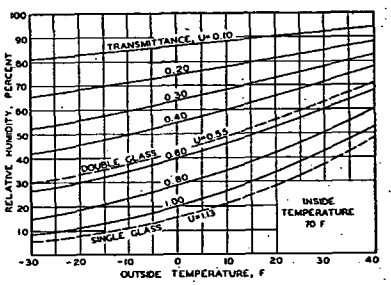


Fig. 4 .... Relative Humidity at Which Visible Condensation Will Appear on Inside Surface

ters concealed condensation. The control of the latter will be discussed. A practical and fully effective technique for such control has, however, not been developed. Usually, sub-surface dwellings designed for occupancy should be dehumidified.

A similar situation is seen in dwellings with concrete floor slabs laid on the ground where such slabs are not a part of the heating system. In a northern climate where high dew-point temperatures occur in summer, condensation or very high relative humidity may damage rugs which are themselves contributors to the trouble since they reduce the floor slab temperature. Slab floors above grade are not so serious a problem as floor slabs on grade but their response to air temperature change is slow. Faster warming is accomplished by the removal of rugs and abundant ventilation at proper times. In their design, floor slabs should have as low specific heat and as high thermal resistance as is consistent with other requirements. Suitable insulation below the slab, especially well-drained gravel, will help somewhat. A top surface cover of insulating value that is unaffected by water on its lower side would be desirable in the less favorable northern climates.

The avoidance of interior visible condensation is partly a construction and partly an operating problem. It is accomplished by reducing the interior dew-point temperature or by raising the surface temperatures that are below the dew point, or both. The dew-point temperature may be lowered by giv-

ing attention to the sources of the moisture, and in winter, may be controlled by ventilation, or possibly by some moisture absorption process. The temperatures of the inside room surfaces in winter may be increased by adding insulation to outside walls, by double glazing of windows, by circulating warm air over the surface, or perhaps by direct heating of the surface. The most expedient method of overcoming a surface condensation difficulty will depend upon special conditions surrounding the problem.

CONCEALED CONDENSATION IN HEATED BUILDINGS

Water vapor produced in a building necessarily raises the vapor pressure above that outdoors thus providing the force that causes its diffusion into exterior walls. The amount of vapor-pressure rise in the building depends on the amount of vapor produced and inversely on its chance to escape. The resulting balance may be expressed in terms of relative humidity if the inside temperature is 70 F. The relative humidity in heated buildings covers nearly all of the possible range. In zero weather it may be only 10 percent in an office, and 85 percent in an industrial plant where humidification is required for a process, or where vapor release is incidental to a process. In residences the relative humidity in cold winter weather ranges from 10 percent to 60 percent, the latter figure applying to a very small, crowded and unventilated dwelling. A 40 percent level is considered representative of a substantial number of modern tightly constructed small houses although the average house relative humidity is probably below 25 percent. Surveys in residences show that the relative humidity increases as would be expected in warmer weather. Fig. 5 represents the results of one such survey.[8]

When water vapor is allowed to enter a wall and condensation occurs on its outer cold elements, it appears as frost or liquid. If the weather temperature rises frequently, frost melts and becoming liquid, is likely to penetrate capillary materials like wood, or run down when the surface is nonabsorbing or is already saturated with water. In weather that is continuously cold for a long period, the frost may build back into a cavity or fibrous insulation and, when it reaches a warmer plane, will run to lower, cooler levels where it forms a mass of ice. Water seepage to the weather side may occur harmlessly in masonry walls when the weather is above freezing but water seepage into the building must obviously be avoided. In typical frame construction with wood sheathing which has large water absorbing capacity, seepage is rare and occurs only after a long period of steady cold weather. More generally, moisture accumulates in wood sheathing and siding through the colder months and reaches a peak in late winter, after which the drying of spring and summer completes the annual cycle.[16] The average winter temperature and its duration are factors in the condensation problem. In Fig. 6 the map of the United States is divided into condensation zones based on winter weather conditions. The solid lines separating the zones follow state lines, and are those recommended by the Housing and Home Finance Agency for the guidance of owners, builders, and architects.[17] The dotted lines are the −20 F, 0 F, and +20 F isotherms of winter design temperature taken from Fig. 1, Chapter 12. It will be noted that there is reasonably good correlation between the zones determined by the two sets of lines. Zone I roughly includes those areas where the design temperature is −20 F or colder; Zone II those for which the design temperature is zero to −20 F, and zone III those at zero and warmer. Within each zone, similar degrees of condensation trouble are to be expected, and similar corrective measures apply.

In roofs, the condensation problem is much the same as in walls. The roof covering may be even more resistant to the escape of vapor than wall coverings such as paint; and while paint is likely to be ruptured by excessive moisture, no such relief occurs in roofs. Thus roofs furnish conspicuous examples of rapid decay in lumber.

In crawl spaces over uncovered damp ground, a large water evaporation sometimes occurs and causes condensation on the outer ends of floor joists and other members that are below the floor line and near the outdoors. Water vapor from the crawl space may also enter walls, be transported by rising air in a stack effect, and even reach the attic by this route when the wall structure permits. Ventilation, as discussed later in this chapter, is an important correction factor in these cases.

Insulation in a wall or roof reduces heat loss and lowers the temperature of the outer elements of the structure, thus increasing the possibility of condensation if the vapor path to the cold surface is not blocked. Since low vapor resistance is a characteristic of fibrous insulation, the needed vapor resistance must be provided by other means. It is to be noted that, in typical residential conditions, condensation does not occur in fibrous insulation itself, except when frost has formed on sheathing and gradually built backward among the fibers. Wet insulation may result from this condition or from liquid condensation seeping down from a higher level.

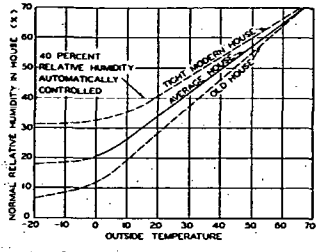


Fig. 5 .... Relative Humidity in Dwellings

CONTROL OF CONCEALED CONDENSATION

An excessive accumulation of moisture in walls (or roofs) can be prevented by one or more of the following measures: (1) providing a vapor barrier to limit vapor entrance into the wall, (2) ventilating the building to reduce vapor pressure therein, or (3) ventilating the wall cavity to remove vapor that has entered.

1. *Vapor Barrier.* A vapor barrier is the principal and most obvious correction, but each measure is more effective if aided by the effect of another. In habitations, some ventilation of the living space, either incidental or planned, is necessary. Also, a small amount of cavity ventilation is essential in cases where the vapor inflow is not completely stopped and the moisture storing capacity of the outer wall elements is slight. This applies to some prefabricated designs using metal siding.

Vapor-barrier sheets are often built into the wall near the warm surface. In wood frame walls they may be applied to the inside surface of the studs. They are sometimes attached to the warm side of the insulating materials, or they may be applied on the cold side of plaster base materials. Special designs may be attached like wall paper to the inside of the wall, when satisfactory from the decorative view point. Sheet barriers often contain asphalt as the vapor-resisting ingredient; metal foils, so placed that they are not too cold, may also be used.

The interior wall board or finish material may itself be vapor resistant, or a barrier coating may be applied to its concealed side if that side will not be too cold. The interior finished surface may be coated with a suitable paint having the required vapor resistance and also serving as the decorative finish, or it may be covered by another coat.

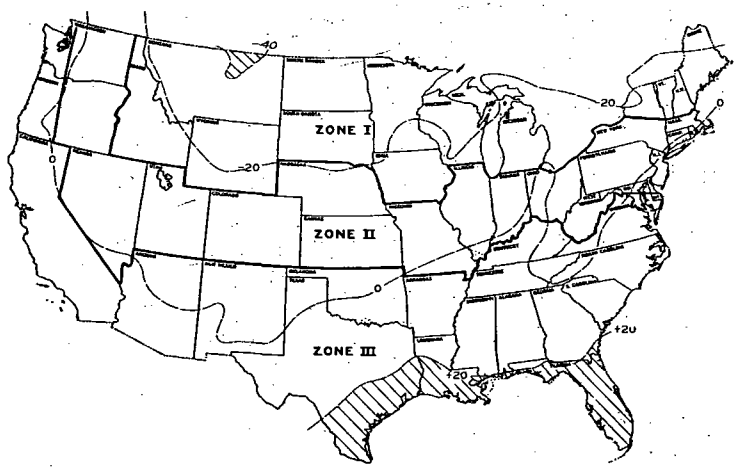
A paint coat on the interior finish, though of adequate resistance, is not likely to be as effective as a sheet barrier properly applied during the wall construction. This applies especially to houses of more than one story having cavities in ceilings which open into the outside walls. Such cavities allow vapor entering the ceiling to diffuse or be transported to the cold areas. Stoppage of this path is difficult and requires normally the painting of the ceiling as well as the walls. Similar treatment may be required on internal partitions, or at least the first stud space adjacent to a cold wall.

The necessary barrier resistance depends on a number of factors. When the vapor flow occurs in annual cycles as in heated buildings, the requirement is not as exacting as it is for a cold storage room where there is no chance for drying out an accumulation of moisture. In a heated structure covered on the outside with materials highly resistant to water vapor such as paint or roll roofing, the winter season is a time of moisture accumulation in the cold outer elements and their safe moisture holding capacity is an important factor in determining the barrier requirement.[16] A house without sheathing requires a better barrier. A prefabricated design with only a sheet of metal outside of insulation requires very high barrier resistance. The interior vapor pressure and the length and severity of the winter are also important.

For typical frame dwellings with wood sheathing and siding in the northern United States, a barrier permeance of one perm or less has been found satisfactory. There are cases, however, in residential construction where a one perm barrier would not be adequate and there are also many industrial applications in which a very much higher vapor resistance is required. In any event, the choice of an adequate barrier implies that its permeance be definitely established. The usually accepted test procedure for this purpose is a dry method at a temperature of 70 F to 80 F. If obtainable at reasonable cost (including good application), a barrier better than required should be chosen for any construction. Despite the theoretical possibility of safely discharging some vapor through a wall, a higher-than-minimum permeance is not preferred.

An exact statement showing which buildings require a vapor barrier is not readily formulated. However, in view of the distressing results its omission may bring, it is tentatively recommended that the walls of every well constructed modern dwelling include a vapor barrier when the construction includes any material that would be damaged by moisture or its freezing. This applies to all condensation zones in Fig. 6 when the *U* value for the wall is lower than 0.25 Btu per (sq ft) (hr) (F deg), and it applies in Zone I and Zone II to walls of higher transmittance.

In applying vapor resistance to a wall, there are certain



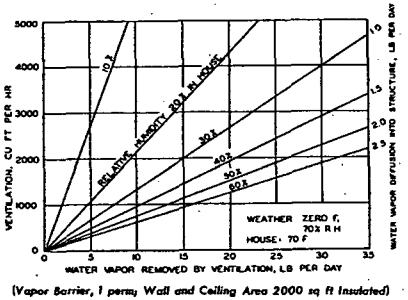
Zones include Areas with Design Temperatures about as follows: Zone I, −20 F and lower; Zone II, 0 F to −20 F; and Zone III, above 0 F. Note that cross hatched areas are outside of Zones I and III.

Fig. 6 .... Condensation Zones in the United States

fundamental principles that should be followed. First, the vapor barrier should be placed as near to the warm surface of the wall as practicable. Second, it should be continuous with no direct openings through the barrier. Good workmanship and application are very important. Workmanship that leaves two openings through the barrier, or around its margin, at different levels, connecting air spaces at only slightly different temperatures, leaves a path for thermosyphon air rotation which will transport large amounts of water vapor from the warmer space to the colder. If a membrane barrier is used back of the plaster or interior finish, its joints should be made over some solid framing member, and not between the studs or in similar places. Usually a two-inch lap over a framing member will make a sufficiently tight joint when the interior finish is applied. Such a lap, however, without backing would not be adequate. Barriers attached to the warm side of insulation should form a continuous unbroken membrane over the entire insulated area. Edges should be lapped over framing members; ends of strips should be fastened by lapping over plates or headers. All openings for electrical fixtures and joints around window and door casings should be carefully sealed. Holes accidentally made in the barrier should be sealed.

2. *Ventilation of Living Space.* The second measure listed for the control of concealed condensation is ventilation of the house. This measure is obviously necessary as an accompaniment to a vapor barrier since, if the barrier blocks entrance into the walls, the water vapor must be removed by other means. No great volume of air change is necessary, however, and normal infiltration alone is frequently all that is required in winter weather.

The effectiveness of ventilation is shown in Fig. 7, which also shows the small amount of water vapor escaping into the barrier-equipped, well-insulated walls and ceilings (2000 sq ft) of a typical small dwelling, the floor being neglected. Evidently, ventilation of 2000 cfh will remove 21 lb of vapor per day with the relative humidity at 40 percent, while at the same time 1.5 lb escapes into the structure. The total vapor production (22.5 lb) is a typical amount. Double glass will be barely safe from visible condensation as will be seen in Fig. 4. By reference to Chapter 6, 2000 cfh appears to be near the minimum for odor control, and ventilation would have to be higher when cooking is done. By reference to Chapter 11, it appears that usual infiltration will normally supply the necessary air change, but that supplementary ventilation may



(Vapor Barrier, 1 perm; Wall and Ceiling Area 2000 sq ft Insulated)

Fig. 7 .... Water Vapor Balance in a Dwelling

be necessary in kitchen and laundry for proper vapor control and for the reduction of peaks in relative humidity that would otherwise occur in those areas.

3. *Ventilation of Structure.* The third measure listed for the control of concealed condensation is ventilation of the structure itself. It is effective in certain cases especially as a supplement to warm-side vapor resistance which is considerable but not of itself fully adequate. Air from outdoors is used. The vents must be shielded from the entrance of rain and insects.

Attics and crawl spaces may be considered as parts of the structure, and for these portions ventilation is practically a necessity. Attic ventilation has long been an established practice but its effectiveness is likely to be diminished by the newer practice of adding insulation to ceilings. Insulation requires added ventilation which in turn necessitates adequate insulation. The recommended ventilation shown in Table 3 for dwellings[17] is based on such insulation. The *net area* refers to the total of all openings free from obstructions. The use of louvers and 8-mesh screen (usually recommended) requires a gross area 2.25 times that listed. In Zone I of Fig. 6 a ceiling vapor barrier is recommended for all constructions. It is also necessary that stray openings from walls into the attic, or around a loose fitting attic door be avoided. The *stack effect* allows a large inflow of warm air from the dwelling, transporting much vapor to a danger area. More desirable ventilation of the house can be arranged.

Crawl spaces under dwellings where the earth is damp and uncovered require a high rate of ventilation. At least four

Table 3 .... Recommended Good Practice[17] for Loft and Attic Ventilation[a]

Flat Roof—Slope Less than 3 Inches in 12 Inches
Condensation Zone I[c]: Total net area of ventilation should be $\frac{1}{300}$th[b] distributed uniformly at the eaves *plus* a vapor barrier in the top story ceiling. Free circulation must be provided through all spaces.
Condensation Zone II and III: Same as for Zone I.
Gable Roof—Slope over 3 Inches in 12 Inches
Condensation Zone I: Total net area of at least 2 louvers on opposite sides located near the ridge to be $\frac{1}{300}$th[b] *plus* a vapor barrier in the top story ceiling.
Condensation Zone II: Same ventilation as for Zone I. A vapor barrier is not considered necessary.
Condensation Zone III: Same as for Zone II.
Hip Roof
Condensation Zone I: Total net area of ventilation should be $\frac{1}{300}$th[b] with $\frac{1}{600}$th[b] distributed uniformly at the eaves *and* $\frac{1}{600}$th[b] located at the ridge with all spaces interconnected. A vapor barrier should also be used in the top story ceiling.
Condensation Zone II: Same ventilation as for Zone I. A vapor barrier is not considered necessary.
Condensation Zone III: Same as for Zone II.
Gable or Hip Roof—With Occupancy Contemplated
Condensation Zone I: Total net area of ventilation should be $\frac{1}{300}$th[b] with $\frac{1}{600}$th[b] distributed uniformly at the eaves and $\frac{1}{600}$th[b] located at the ridge with all spaces interconnected. A vapor barrier should also be used on the warm side of the top full story ceiling, the dwarf walls, the sloping part of the roof, and the attic story ceiling.
Condensation Zone II: Same as for Zone I.
Condensation Zone III: Same as for Zone I except that a vapor barrier is not considered necessary if insulation is omitted.

[a] It is recognized that in many areas increased ventilation may be desirable for summer comfort. For winter comfort, insulation is recommended between a living space and a loft or attic ventilated at these rates.

[b] Refers to area enclosed within building lines at eave level.

[c] The zone numbers refer to Fig. 6.

openings, one at each corner, as high as possible, should be provided.[17] Their total net area may be calculated by the formula:

$$a = \frac{2L}{100} + \frac{A}{300} \qquad (10)$$

where:

L = the perimeter of the crawl space, linear feet.

A = the area of the crawl space, square feet.

a = the total net area of all vents, (or the gross area if a 4-mesh screen is used) square feet.

This ventilation is usually sufficient but cools the first floor so much that insulation is needed. A better treatment is a cover on the damp ground. This cover may be a concrete slab, or merely heavy roll roofing laid on a graded surface with its edges lapped 2 in. (but not necessarily cemented). With this barrier, the vent area may be reduced to 10 percent of that calculated by Equation 10.

In building walls, cavity ventilation can be applied in a moderate climate as the sole vapor control system. In general, however, air passages in walls designed to remove an unrestricted vapor supply are unduly large and may waste considerable heat. On the other hand, a barrier as the only control measure would, in some cases, require so high a resistance as to be impractical. Ventilation of the structure in conjunction with a vapor barrier, is a procedure with important applications, but its general utility has not been fully investigated. Ventilation is most effective when each structural space has a clearly defined air passage with an inlet and outlet. In walls a small thermosyphon effect may be utilized by locating one vent at the bottom and one at the top of each space.

The best time to vapor-proof a building is during its construction. After a building is completed, ventilation of the occupied space is the most easily applied of the three basic control measures. Paint that is chosen for its low vapor permeance can be applied as a barrier on the interior with good results, care being taken that all areas, including parts of partitions and ceilings which offer an indirect vapor path to the cold wall, are covered. Ventilation of the wall cavity is effective in certain cases especially to supplement the foregoing measures. When such venting is required, each cavity space isolated by framing should be separately vented with an inlet and outlet judiciously placed, to accomplish proper air change.

CONDENSATION IN COOLED STRUCTURES

Water vapor is sometimes an even greater problem in cooled than in heated structures, but the basic facts of its migration and condensation on cold surfaces are the same. Refrigerators, cold pipes, and cold vessels all require insulation and should be provided with a vapor barrier. The barrier, as always, should be placed on the warm side of the insulation and as close as possible to the warm surface of any enclosure. It resists the movement of water vapor toward the colder parts of the structure, and its job is even more exacting than that required in residential construction. In the case of an insulated cold pipe line, the process is likely to be uninterrupted for years and there is no chance for vapor that enters the insulation to dry out periodically. Also, no vapor can escape from the cold side, if this is a metal (vapor tight) pipe. For such an application vapor control requires insulating material that is itself very highly resistant to water vapor or a coating whose permeance is the minimum obtainable, not over 0.1 perm. Metal coverings are desirable but difficult to apply.

Similar considerations apply in the case of cold rooms whether constructed inside a heated building or as a separate building. In the latter case, cold rooms operating above freezing provide some periods of vapor reversal in winter but such drying can be of little help. Refrigerators, however, if lined with cement or other vapor permeant material, will allow slight amounts of vapor to pass and to that extent reduce the accumulation of moisture that may have penetrated the barrier. While this is helpful, emphasis must be placed on an adequate warm-side barrier not greater than 0.1 perm.

Summer air cooling for comfort does not normally create serious vapor problems in exterior walls and ceilings. Normally the cooled air is little, if any, colder than the dew point of the outdoor atmosphere and there are no areas of condensation. The design dry-bulb and design wet-bulb temperatures in common use as given in Table 2, Chapter 13, tend to bear this out. For the 315 cities in the United States, for which design values are given, only 78 have design wet-bulb temperatures in common use over 75 F. Outdoor dew-point temperatures, especially peak values, may exceed the design wet-bulb temperatures in common use, but these will seldom be greatly in excess of 75 F for any prolonged period of time. Condensation within exterior walls which are exposed to indoor air at 75 F will seldom be as serious as for the winter condition.

A vapor barrier to prevent condensation within exterior walls under summer cooling conditions would normally be located on the outside of the insulation, whereas the location for the winter condition is on the inner side. The use of vapor barriers at both locations is not without difficulty because they not only restrict the entry of moisture into the insulation but also restrict the escape of any moisture. It is generally considered best in dwelling construction to locate the vapor barrier for the more serious case of condensation in winter, and to disregard the summer case even when some condensation may be expected. The corresponding situation in cold-storage buildings in which there may be a much more serious reversal of vapor flow conditions from winter to summer poses a wall design problem that has not yet been fully resolved.

Some observations made by the National Research Council of Canada, but not yet publicly reported, would indicate that serious wetting within walls can occur in summer under special conditions. The walls of test huts of brick masonry, but finished inside with strapping, insulation, vapor barrier, and plasterboard interior finish, were opened during a sunny period that followed a rain. Extensive wetting was observed in the insulation, and particularly on the back of the vapor barrier. The brick wall which was unusually absorptive had taken on substantial quantities of water during the period of rainfall. Subsequent heating by the sun had driven the moisture as vapor into the wall where it condensed and caused serious wetting. The construction in question did not have protection for the strapping and insulation in the form of parging or paper on the inside of the brick. Walls having absorptive exterior coverings capable of absorbing and storing considerable quantities of water during a rain and providing little resistance to vapor flow into the insulation from the outdoors, may experience serious interior wetting by condensation under such unusual conditions. No wetting occurred in a construction similar to that described above when a saturated sheathing paper was used between the insulation and the brick. It would appear that the provision of moderate resistance to vapor flow such as that provided

by parging or a good exterior grade sheathing paper on the outside of the insulation can be effective in such cases.

Vapor which diffuses inwards under cooling conditions adds to the cooling load. Vapor barriers in walls installed for winter conditions are a help in reducing this. However, if condensation does occur on the back of a vapor barrier installed on the inside of the insulation, the latent heat which is then released on the inside of the insulation may add to the heat gain through the wall.

REFERENCES

[1] S. C. Hite and J. L. Dray: *Research in Home Humidity Control* (Purdue University, *Engineering Experiment Station Research Series* No. 106, November 1948).

[2] S. C. Chang and N. B. Hutcheon: Dependence of water vapor permeability on temperature and humidity (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 437).

[3] F. A. Joy and E. R. Queer: Permeance measurement improved by special cell (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 377).

[4] F. A. Joy, E. R. Queer, and R. E. Schreiner: *Water Vapor Transfer Through Building Materials* (Pennsylvania State College, *Engineering Experiment Station Bulletin* No. 61, December 1948).

[5] J. D. Babbitt: The diffusion of water vapor through various building materials (*Canadian Journal of Research*, February 1939, p. 15).

[6] F. A. Joy and D. R. Fairbanks: Effect of unbalanced air pressure on permeance (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 451).

[7] R. M. Barrer: *Diffusion In and Through Solids* (Cambridge Press, London, 1951).

[8] L. V. Teesdale: *Remedial Measures for Building Construction* (*U. S. Forest Products Laboratory Report* R1710, 1947).

[9] F. B. Rowley, A. B. Algren, and C. E. Lund: *Methods of Moisture Control and Their Application to Building Construction* (University of Minnesota, *Engineering Experiment Station Bulletin* No. 17).

[10] H. J. Barre: *The Relation of Wall Construction to Moisture Accumulation in Fill-Type Insulation* (Iowa State College of Agriculture and Mechanic Arts, *Agricultural Experiment Station Bulletin* No. 271, 1940).

[11] P. F. McDermott: Moisture migration: a survey of theory and existing knowledge (*Refrigerating Engineering*, August 1941, p. 103).

[12] R. I. Wray and A. R. Van Vorst: Permeability of paint films to moisture (*Industrial and Engineering Chemistry*, Vol. 25, 1933, p. 842).

[13] R. R. Britton and R. C. Reichel: *Water Vapor Transmission of Building Materials Using Four Different Testing Methods* (*U. S. Housing and Home Finance Agency Technical Bulletin* No. 12, January 1950).

[14] E. R. Bell, M. G. Seidl, and N. T. Krueger: Water-vapor permeability of building papers and other sheet materials (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 287).

[15] Value from unpublished texts of Pennsylvania State College Experiment Station.

[16] F. A. Joy: Basic concepts of water vapor migration and their application to frame walls (*ASTM Special Technical Publication* No. 119, 1951, p. 2).

[17] *Condensation Control in Dwelling Constructions* (U. S. Housing and Home Finance Agency, 1949).

BIBLIOGRAPHY

H. W. Wooley: *Moisture Condensation in Building Walls* (*National Bureau of Standards Report* BMS 63, December 14, 1940).

F. B. Rowley, A. B. Algren, and C. E. Lund: Condensation of moisture and its relation to building construction and operation (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 231).

P. D. Close: Permissible relative humidities in humidified buildings (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December 1939, p. 766).

F. B. Rowley, A. B. Algren, and C. E. Lund: Condensation within walls (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 95).

H. Edenholm: Moisture movement and moisture distribution in the walls of buildings (*Meddelanden Fran Statens Forsknings-kommitte for Lantmannabyggnader* No. 5, 1945, p. 53, available as Technical Translation TT-361 from the National Research Council of Canada, 1952).

J. D. Babbitt: *Physics of Condensation in Buildings* (*National Research Council of Canada Bulletin* No. 2).

L. V. Teesdale: Comparative resistance to vapor transmission of various building materials (ASHVE TRANSACTIONS, Vol. 49, 1943, p. 124).

Durability of moisture-resistant membrane materials in contact with the ground (Housing and Home Finance Agency, *Housing Research* No. 4, October 1952, p. 23).

Proposed method of test for water vapor transmission of building materials utilizing the Penn State-Armstrong cell (*ASTM Bulletin* No. 215, July 1956, p. 63).

Heat and water vapor transmission apparatus for insulated panels (National Bureau of Standards, *Technical News Bulletin*, November 1954, p. 157).

F. A. Joy and A. W. Sherdon: Automatic permeance measurement by the permeometer (ASHAE TRANSACTIONS, Vol. 59, 1953, p. 435).

F. B. Rowley: A theory covering the transfer of vapor through materials (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 545).

F. G. Hechler, E. R. McLaughlin, and E. R. Queer: Simultaneous heat and vapor transfer characteristics of an insulating material (ASHVE TRANSACTIONS, Vol. 48, 1942, p. 505).

J. A. Paxton and N. B. Hutcheon: Moisture migration in a closed, guarded hot plate (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 301).

C. G. Gurr, T. J. Marshall, and J. T. Hutton: Movement of water in soil due to a temperature gradient (*Soil Science*, November 1952, p. 335).

W. A. Hadley and Ray Eisenstadt: Moisture movement in soils due to temperature difference (ASHAE TRANSACTIONS, Vol. 59, 1953, p. 395).

K. R. Solvason: Moisture in transient heat flow (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 111).

H. F. Winterkorn: Fundamental similarities between electro-osmotic and thermo-osmotic phenomena (*Proceedings 27th Annual Meeting*, Vol. 27, Highway Research Board, 1947, p. 443).

S. C. Chang and N. B. Hutcheon: Performance of desiccants in the dry pan test for water vapor permeance of membranes (*Canadian Journal of Technology*, September 1953, p. 175).

F. A. Joy: Thermal conductivity of insulation containing moisture (*ASTM Special Technical Publication* No. 217, February 1957, p. 65).

J. S. Cammerer: The effect of moisture on heat transmission through building and insulating materials (*Wärme und Kältetechnik*, September 1939, p. 126, available as Technical Translation TT-317 of the National Research Council of Canada, 1952).

H. B. Jesperson: Thermal conductivity of moist materials and its measurement (*Journal of IHVE*, August 1953, p. 157).

D. Krischer: Heat conductivity and water vapor diffusion in materials for insulation against cold (*Wärme und Kältetechnik*, 1941, 43(0), p. 2, translation available from British Building Research Station as Library Communication No. 492).

D. A. De Vries: The thermal conductivity of soil (Mededelingen van de Landbouwhogeschool te wageningen, 1952, 52(1), p. 1, translation available from British Building Research Station as Library Communication No. 759).

C. H. Johansson: Moisture transmission and moisture distribution in building materials (*Wärme-Ventilations-Sanitetstek*, 19: 67, 1948, available as Technical Translation TT-189 from the National Research Council of Canada).

J. D. Babbitt: The movement of moisture through solids (*ASTM Bulletin* No. 212, February 1956, p. 58).

J. R. Philip and D. A. De Vries: Moisture movement in porous materials under temperature gradients (*American Geophysical Union Transactions*, April 1957, p. 222).

CHAPTER 11

INFILTRATION AND VENTILATION

Causes of Infiltration, Infiltration Due to Wind Pressure, Infiltration Due to Temperature Difference, Sealing of Vertical Openings, Infiltration Measurement, Natural Ventilation, Wind Forces, Temperature Difference Forces, Heat Removal, Effect of Unequal Openings, Combined Wind and Temperature Forces, Types of Openings, General Ventilation Rules, Ventilation of Animal Shelters, Garage Ventilation

INFILTRATION is the air leakage through cracks and interstices, around windows and doors, and through floors and walls. Its magnitude depends on the structural design, workmanship, and condition of the building. The rate of infiltration cannot be controlled by the inhabitants of the building to any considerable extent. Natural ventilation is the controlled displacement of air through openings, such as windows, doors, and ventilators as well as through combustion heating devices.

CAUSES OF INFILTRATION

The air leakage which takes place through various apertures in buildings must be estimated in heating and cooling calculations and enough heating or cooling capacity provided to offset the heat lost or gained by the air leakage. The rate of air flow into and out of a building depends on the magnitude of the pressure difference between the inside and outside of the structure and on the resistances presented to this pressure difference. The pressure difference exerted on the building walls by the air may be caused either by wind or by a difference in density of the air inside and outside the building. The effect of the wind depends on the interrelation of the speed and direction of the wind and the exposure of the building. The effect of the difference in the density of the air depends on the magnitude of the indoor-outdoor temperature difference, the height of the rooms, the shape of the openings, and their elevation in the room or building. The effect of the difference in density is often referred to as the chimney or stack effect. The pattern of air flow through any part of the structure depends on both the pressure difference and the openings. In general, when the pressure difference is the result of wind pressure, air will enter the building through openings in the windward walls and leave through openings in the leeward walls or through ventilating ducts in the roof. When the pressure difference is caused by the indoor-outdoor temperature difference the flow will be along the path of least resistance from inlets at lower levels to outlets at higher levels in a heated building or in the opposite direction for an air-conditioned building.

An exact estimate of the amount of infiltration under design conditions is difficult to make. The complicating factors include (1) variations in building construction, particularly as to width of crack or size of openings through which air leakage takes place; (2) variations in wind velocity and direction; (3) exposure of the building with respect to air leakage openings, and with respect to adjoining buildings; (4) variations in outdoor temperatures which influence the chimney effect; (5) relative area and resistance of openings on the windward and leeward sides, and on the lower floors and on the upper floors; and (6) influence of a planned air supply and the related outlet vents. Tight construction is essential for preventing large heat loss due to infiltration.

In view of the meager information available on infiltration, a cooperative investigation was undertaken in 1954 in which the infiltration rates of residences were obtained by actual measurement.[1, 2] In 1956 a study was started at the ASHAE Research Laboratory to obtain additional information on infiltration through building entrances.[3] These studies are yielding quantitative information on factors affecting infiltration rates, but more data are still needed.

INFILTRATION DUE TO WIND PRESSURE

The wind causes a pressure to be exerted on one or two sides of a building. As a result, air comes into the building on the windward side through cracks or porous construction, and a similar quantity of air leaves on the leeward side through like openings. In general, the resistance to air movement is similar on the windward to that on the leeward side. This causes a building up of pressure within the building, and a lesser air leakage than that experienced in single wall tests as determined in the laboratory. It is assumed that actual building leakages, owing to this building up of pressure, will be 80 percent of laboratory test values. While there are cases where this is not true, tests in actual buildings substantiate the factor for the general case. Mechanical ventilating systems are frequently designed to produce positive or negative pressures in an enclosure, which are greater or lower than prevalent wind pressures. In such designs, if the specified rate at which air is to be supplied to, or removed from, the enclosure by positive means, exceeds the infiltration rate, it is common practice to use the greater value in determining the heating capacity to warm the outdoor air.

Infiltration Through Walls

Data on infiltration through brick and frame walls are given in Table 1.[4] The brick walls listed in this table are walls that show poor workmanship, and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 percent; a heavy coat of cold water paint, 50 percent; and three coats of oil paint carefully applied, 28 percent. The infiltration through walls ranges from 6 to 25 percent of that through windows and doors in a 10-story

Table 1 .... Infiltration Through Walls[a]

Expressed in cubic feet per (square foot) (hour)

Type of Wall		Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
Brick Wall[b]							
8¼ in.	Plain	2	4	8	12	19	23
	Plastered[c]	0.02	0.04	0.07	0.11	0.16	0.24
13 in.	Plain	1	4	7	12	16	21
	Plastered[c]	0.01	0.01	0.03	0.04	0.07	0.10
	Plastered[d]	0.03	0.10	0.21	0.36	0.53	0.72
Frame Wall, lath and plaster[e]		0.03	0.07	0.13	0.18	0.23	0.26

[a] The values given in this table are 20 percent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed in chapter footnotes.

[b] Constructed of porous brick and lime mortar—workmanship poor.

[c] Two coats prepared gypsum plaster on brick.

[d] Furring, lath, and two coats prepared gypsum plaster on brick.

[e] Wall construction: bevel siding painted or cedar shingles, sheathing, building paper, wood lath, and three coats gypsum plaster.

office building, with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 percent; or a practically negligible quantity, which indicates the importance of good workmanship for proper sealing at the baseboard. It will be noted from Table 1 that the infiltration through properly plastered walls can be neglected.

The value of building paper, when applied between sheathing and shingles, is indicated by curve 3D-7D, Fig. 1, which represents the effect on outside construction only, without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper, or of the poor construction of the wall containing it. Not only is it difficult to secure and maintain the full effectiveness of the plaster, but also it is highly desirable to have two points of high resistance to air flow with an air space between them. The infiltration indicated in Fig. 1 is that determined in the laboratory, and *should be multiplied by the factor* 0.80 to give proper working values.

Window and Door Leakage

There are two methods of estimating air leakage through window and door cracks, namely, (1) the crack method, and (2) the air change method. The crack method is generally regarded as being more accurate than the air change method, provided the variables, such as crack width and clearance, can be properly evaluated.

Crack Method

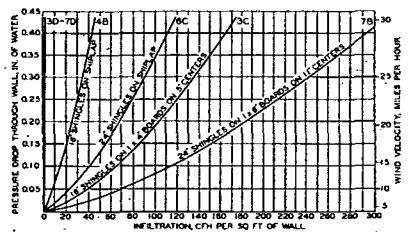
The crack method is based on known air leakage factors for various types of windows, and widths of crack and clearance. The wind velocity and length of crack are also considered when the crack method is employed. The amount of infiltration for various types of windows is given in Table 2.[a] The fit of double-hung wood windows is determined by crack and clearance. Crack thickness is equivalent to one-half the difference between the inside window frame dimension and the outside sash width. The difference between the width of the window frame guide and the sash thickness is considered as the clearance. The length of the perimeter opening or crack for a double-hung window is equal to three times the width, plus two times the height, or in other words, it is the outer sash perimeter length, plus the meeting rail length. All of the window crack in any given room is not necessarily used in estimating the infiltration heat loss by the crack method. The length of crack to be selected in any given case depends on the number of exposed sides, as explained in Chapter 12.

Values of leakage shown in Table 2 for the average double-hung wood window were determined by using, on nine windows tested in the laboratory, the average measured crack and clearance of a large number of windows found in a field survey. In addition, the table gives figures for a poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable figures, provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For a metal pivoted sash, the length of crack is the total perimeter of the movable or ventilating sections. Frame leakage on steel windows may be neglected when they are properly grouted with cement mortar into brick work or concrete. When they are not properly sealed, the linear feet of sash section in contact with steel work at mullions should be figured at 25 percent of the values given in Table 2 for industrial pivoted windows.

When storm sash are applied to well fitted windows, some reduction in infiltration is secured; the application of the sash provides an air space which reduces the heat transmission and helps prevent the frosting of the windows.[a] By applying storm sash to poorly fitted windows, a reduction in leakage of 50 percent may be obtained, the effect, so far as air leakage is concerned, being roughly equivalent to that obtained by the installation of weatherstrips.



3D-7D: 3-D-16 inch shingles on 1 x 4 inch boards on 5 inch centers with paper, 7-D-16 inch shingles, shiplap and paper.

Fig. 1 .... Infiltration Through Various Types of Shingle Construction

Table 2 .... Infiltration Through Windows

Expressed in cubic feet per foot of crack per hour[a]

Type of Window	Remarks	Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3	8	14	20	27	35
	Around frame in masonry wall—calked[b]	1	2	3	4	5	6
	Around frame in wood frame construction[b]	2	6	11	17	23	30
	Total for average window, non-weatherstripped, 1/16-in. crack and 3/64-in. clearance.[c] Includes wood frame leakage[d]	7	21	39	59	80	104
	Ditto, weatherstripped[d]	4	13	24	36	49	63
	Total for poorly fitted window, non-weatherstripped, 3/32-in. crack and 3/32-in. clearance.[c] Includes wood frame leakage[d]	27	69	111	154	199	249
	Ditto, weatherstripped[d]	6	19	34	51	71	92
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted, 1/16-in. crack[e]	52	108	176	244	304	372
	Architectural projected, 1/32-in. crack[h]	15	36	62	86	112	139
	Architectural projected, 3/64-in. crack[h]	20	52	88	116	152	182
	Residential casement, 1/64-in. crack[i]	6	18	33	47	60	74
	Residential casement, 1/32-in. crack[i]	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in. crack[j]	3	10	18	26	36	48
	Heavy casement section, projected, 1/32-in. crack[j]	8	24	38	54	72	92
Hollow Metal, Vertically Pivoted Window[f]		30	88	145	186	221	242

[a] The values given in this table, with the exception of those for double-hung and hollow metal windows are 20 percent less than test values to allow for building up of pressure in rooms, and are based on test data, reported in the papers listed in chapter footnotes.

[b] The values given for frame leakage are per foot of sash perimeter, as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself, and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and non-calked tests.

[c] The fit of the average double-hung wood window was determined as 1/16-in. crack and 3/64-in. clearance by measurements on approximately 600 windows under heating season conditions.

[d] The values given are the totals for the window opening per foot of sash perimeter, and include frame leakage and so-called *elsewhere leakage*. The frame leakage values included are for wood frame construction, but apply as well to masonry construction assuming a 50 percent efficiency of frame calking.

[e] A 1/32-in. crack and clearance represent a poorly fitted window, much poorer than average.

[f] Windows tested in place in building, so that no reduction from test values is necessary, as mentioned in footnote a.

[g] Industrial pivoted window generally used in industrial buildings. Ventilators horisontally pivoted at center or slightly above, lower part swinging out.

[h] Architecturally projected made of same sections as industrial pivoted, except that outside framing member is heavier, and it has refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms. 1/32-in. crack is obtainable in the best practice of manufacture and installation, 3/64-in. crack considered to represent average practice.

[i] Of same design and section shapes as so-called *heavy section casement*, but of lighter weight. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[j] Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice. Known as Intermediate Windows by steel window manufacturers.

[k] With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions, is negligible. With 1/64-in. crack, representing poor installation, leakage at contact with steel framework is about one third, and at mullions, about one-sixth of that given for industrial pivoted windows in the table.

Door Leakage

Doors vary greatly in fit because of their large size and tendency to warp. For a well fitted door, the leakage values for a poorly fitted double-hung wood window may be used. If poorly fitted, twice this figure should be used. If weatherstripped, the values may be reduced one-half. A single door that is frequently opened, as might be the case in a store, should have a value applied three times that for a well fitted door. This extra allowance is for opening and closing losses, and is kept from being greater by the fact that doors are not used as much in the coldest and windiest weather.

The infiltration rate through swinging and revolving doors is generally a matter of judgment by the engineer making *cooling* load determinations. In the absence of adequate research data, the values given in Table 3 may be used to represent current engineering practice. Some tests of infiltration through swinging and revolving doors have been reported.[a] The data in Table 3 are indicative of what might be expected in this connection, but it should be noted that Table 3 is based on a no-wind condition, and therefore is not directly applicable to *heating* design.

A wide range of infiltration rates would be expected for swinging doors because of variations in the indoor-outdoor pressure difference caused by wind, temperature differences, and the degree to which a heating or an air-conditioning

Table 3 .... Infiltration Through 72-Inch Revolving Door and 36-Inch Swinging Door[a, b]
(Cubic feet per person per passage)

Usage	Freely-Revolving Door	Door Equipped with Brake
Infrequent	75	60
Average	60	50
Heavy	40	40
36-Inch Swinging Door	20 to 100	

[a] These figures are based on the assumption that there is no wind pressure and that swinging doors are in use in one wall only. Any swinging doors in other walls should be kept closed to insure air conditioning in accordance with these recommended standards.

[b] From *Application Engineering Standards for Air Conditioning for Comfort* 1947, Air Conditioning & Refrigeration Institute Inc., Washington, D. C. and from experimental data of National Bureau of Standards. Used by permission.

system tended to raise or lower the inside pressure by blower action. The frequency of door usage would also affect the amount of leakage per passage.

Air Change Method

The amount of air leakage may be estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use, and location of the room, as indicated in Table 4. Where it is not possible to determine or predetermine with accuracy the width of crack or clearance of windows, or when other sources of air leakage cannot readily be evaluated, as is often the case, the use of the air change method may be justified.[a]

The values in Table 4 may be used with reasonable accuracy for residences, and are the requirements for each room. The total infiltration allowance for the entire building should be one-half the sum of the infiltration allowances of the individual rooms, since whatever air enters on the windward side, generally leaves the building on the leeward side, and the infiltration requirements therefore do not exist simultaneously on all sides or in all rooms. An allowance of one air change per hour for all sources of air leakage for the entire volume may be considered average for a well constructed residence.

The air leakage, due to opening and closing of doors in vestibules, is sometimes based on the air change method, even though the air leakage estimates for other rooms are based on the crack method. Except for vestibules and reception halls, it is not advisable to attempt to apply the air change method to factories and industrial and commercial buildings, because of the wide variations in the type and percentage of fenestration which is the principal source of air leakage in such buildings.

INFILTRATION DUE TO TEMPERATURE DIFFERENCE

The air exchange due to temperature difference, indoor to outdoor, is a chimney effect, causing air to enter through openings at lower levels, and to leave at higher levels[a] when the building is heated or causing flow in the reverse direction when the building is cooled. This air exchange is usually of considerable importance in tall, single-story buildings with openings near the ground level and near the ceiling; it should also be considered in tall, multi-story buildings unless the sealing between various floors is nearly perfect; and may be appreciable in one-story buildings with basements and attics.

The flow of air through a building under the influence of indoor-outdoor temperature difference is quite complex except for single-story structures without a basement or attic. It can best be understood by visualizing the building as a complex chimney with a number of passages and a number of restrictions. The basement represents the lower section of this imaginary chimney with air moving inward through cracks in the walls and around windows and doors. The air then moves upward through cracks and openings in the floor which acts like a damper in the chimney. Above the floor the chimney has many parallel passages consisting of the several rooms and the hollow wall spaces in certain types of construction. These separate passages are all interconnected by cracks and also communicate with the outdoors through cracks and fissures. The ceiling acts as another damper with air flowing upward through various cracks and openings. This condition is repeated in every story of a multi-story building and can be further complicated by open stairways, elevator shafts, and utility ducts. The attic finally represents the union of all the parallel passages in the chimney with the outward flow of air being again restricted by the roof construction.

Since the chimney effect in a building produces a negative pressure and an inward flow of air at the lower levels and positive pressure and outward flow at the higher levels, a neutral zone[m] exists near midheight where there is no pressure difference between indoors and outdoors, if the openings are about uniformly distributed in a vertical direction. At the neutral zone there would be no air flow through openings in the outside walls as a result of temperature differences.

The infiltration caused by the indoor-outdoor temperature differences can be calculated by means of the crack method described earlier in this chapter for determining the infiltration caused by wind pressure. This is done by determining the equivalent wind velocity that would produce the same rate of infiltration as was caused by the prevailing temperature difference. It is recommended that one-half the total crack length of the building be used for this computation. To determine the infiltration caused by the temperature difference, one-half the crack length of the building is multiplied by the infiltration coefficient from Table 2

Table 4 .... Air Changes Taking Place under Average Conditions in Residences, Exclusive of Air Provided for Ventilation[a]

Kind of Room or Building	Number of Air Changes Taking Place per Hour
Rooms, 1 side exposed	1
Rooms, 2 sides exposed	1½
Rooms, 3 sides exposed	2
Rooms, 4 sides exposed	2
Rooms with no windows or outside doors	½ to ¾
Entrance Halls	2 to 3
Reception Halls	2
Bath Rooms	2

[a] For rooms with weatherstripped windows or storm sash, use ½ these values, where applicable, but never less than ½ air change.

corresponding to the equivalent wind velocity computed from the following equation developed from basic relationships between velocity, pressure, density, and temperature.

$$V_e = B\sqrt{h(t_i - t_o)} \qquad (1)$$

where

- V_e = equivalent wind velocity corresponding to the temperature difference $(t_i - t_o)$, miles per hour.
- h = height of rooms, feet.
- t_i = indoor temperature, Fahrenheit.
- t_o = outdoor temperature, Fahrenheit.
- B = a constant to account for leakage through floor and ceiling and for the number of stories in the building.

The constant B would be 0.12 for a single-story building or for any story of a multi-story building whose floor and ceiling were impervious to air, whereas the value of B has been found to be approximately unity for a single-story frame building with basement and attic having double wood flooring, plastered ceiling, and the walls finished with plasterboard on the inside. The value of this constant would increase as the number of stories increases and for stories farther removed from the neutral zone in either direction in buildings that lack perfect sealing between stories.

Sealing of Vertical Openings

In tall, multi-story buildings, every effort should be made to seal off vertical openings, such as stair-wells and elevator shafts, from the remainder of the building. Stair-wells should be equipped with self-closing doors, and, in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight, and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance need be made for the chimney effect. Instead, the greater wind movement at the greater heights makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One arbitrary rule is to increase the heating surface on floors above neighboring buildings by an amount ranging from 5 percent to 20 percent. This extra heating surface is required only on the windward side and on windy days, and hence, automatic temperature control is especially desirable with such installations.

In stair-wells that are open through many floor levels, although closed off from the remainder of each floor by doors and partitions, the stratification of air makes it advisable to increase the amount of heating surface at the lower levels, and to decrease the amount at higher levels. One rule is to calculate the heating surface of the entire stair-well in the usual way, and to place 50 percent of this in the bottom third, the normal amount in the middle third, and the balance in the top third.

Infiltration and Air for Combustion

Infiltration in buildings normally supplies the air required for combustion by fuel-burning appliances, but in some cases weatherstripping, sealing, and calking may reduce infiltration to the point that special openings must be provided to supply adequate air to the heating appliances. This need for combustion air is recognized in various building codes. For instance, the State Building Construction Code of New York State for One- and Two-Family Dwellings[13] requires that an air inlet area not less than the area of the smoke pipe connection be provided if adequate air supply at all times is not assured. Similarly, the same code for multiple dwellings requires a permanent opening to the outdoor air for rooms containing fuel-burning appliances having a gross capacity in excess of 250,000 Btuh.

NATURAL VENTILATION

Ventilation by natural forces finds application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for moving air into, through, and out of buildings are: (*a*) wind forces, and (*b*) the difference in temperature between the air inside and outside a building. The air movement may be caused by either of these forces acting alone, or by a combination of the two, depending upon atmospheric conditions, building design, and location. The ventilating results obtained will vary, from time to time, due to variation in the velocity and direction of the wind, and the temperature difference. The arrangement, location, and control of the ventilating openings should be such that the two forces act cooperatively rather than in opposition.

WIND FORCES

In considering the use of natural wind forces for producing ventilation, account must be taken of: (1) average wind velocity; (2) prevailing wind direction; (3) seasonal and daily variations in velocity and direction; and (4) local wind interference by nearby buildings, hills, or other obstructions of similar nature.

Values are given in Table 2, Chapter 13, for the average wind velocities for the months June to September in various localities throughout the United States, while Table 1, Chapter 12, lists similar values for the winter. In almost all localities, the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While the tables give no average velocities below 5 mph, there will be times when the velocity is lower, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the velocity falls below one-half of the average for many hours per month. Consequently, if the natural ventilating system is designed for wind velocities of one-half of the average seasonal velocity, it should prove satisfactory in almost every case.

Equation 3 may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings to produce given results:

$$Q = EAV \qquad (3)$$

where

- Q = air flow, cubic feet per minute.
- A = free area of inlet openings, square feet.
- V = wind velocity, feet per minute, = miles per hour × 88.
- E = effectiveness of openings. (E should be taken at 0.50 to 0.60 for perpendicular winds, and 0.25 to 0.35 for diagonal winds.)[12]

The precision of results obtained by the use of Equation 3, depends upon the placing of the openings, as the formula assumes that ventilating openings have a flow coefficient

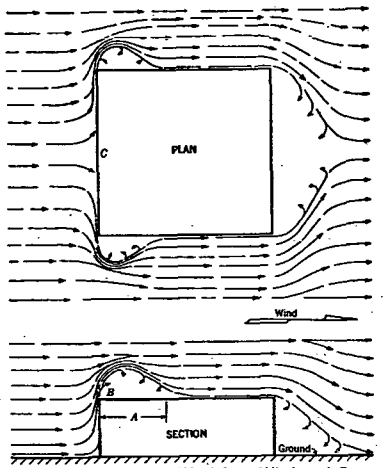


Fig. 2 .... The Jump of Wind from Windward Face of Building. (A—Length of Suction Area; B—Point of Maximum Intensity of Suction; C—Point of Maximum Pressure)

slightly greater than that of a square-edged orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less and, if unusually well placed, the flow will be slightly more than that given by the formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the five places listed:

1. On the side of the building directly opposite the direction of the prevailing wind.
2. On the roof in the low pressure area caused by the jump of the wind (see Fig. 2).
3. On the sides adjacent to the windward face where low pressure areas occur.
4. In a monitor on the side opposite from the wind.
5. In roof ventilators or stacks.

TEMPERATURE DIFFERENCE FORCES[10]

The stack effect produced within a building, when the outdoor temperature is lower than the indoor temperature, is due to the difference in weight of the warm column of air within the building and cooler air outdoors. The flow due to stack effect is proportional to the square root of the draft head, or approximately

$$Q = 9.4A\sqrt{h(t_i - t_o)} \qquad (4)$$

where

Q = air flow, cubic feet per minute.
A = free area of inlets or outlets (assumed equal), square feet.
h = height from inlets to outlets, feet.
t_i = *average* temperature of indoor air in height h, Fahrenheit.
t_o = temperature of outdoor air, Fahrenheit.
9.4 = constant of proportionality, including a value of 65 percent for effectiveness of openings. This should be reduced to 50 percent (constant = 7.2) if conditions are not favorable.

HEAT REMOVAL

In problems of heat removal, knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of air to be passed through the building per minute, to maintain this temperature difference, can be determined by means of Equation 5.

$$Q = \frac{H}{C_p \times \rho \times 60(t_i - t_o)} = \frac{H}{1.08(t_i - t_o)} \qquad (5)$$

where

Q = air removed, cubic feet per minute.
H = heat removed, Btu per hour.
C_p = specific heat of air at constant pressure, 0.24.
ρ = density of standard air, 0.075 pounds per cubic foot.
$t_i - t_o$ = indoor-outdoor temperature differences, Fahrenheit.

EFFECT OF UNEQUAL OPENINGS

The largest flow per unit area of openings is obtained when inlets and outlets are equal, and the preceding equations are based on this condition. Increasing outlets over inlets, or vice-versa, will increase the air flow, but not in proportion to the added area. When solving problems having an unequal distribution of openings, use the smaller area, either inlet or outlet, in the equations, and add the increase as determined from Fig. 3.

COMBINED FORCES OF WIND AND TEMPERATURE

Equations have already been given for determining the air flow due to temperature difference and wind. It must be remembered that when both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two estimated quantities. The flow

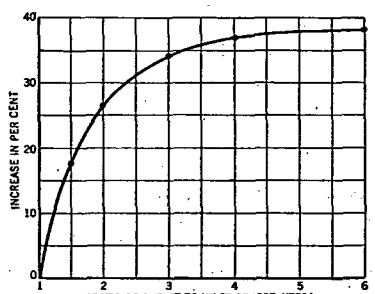


Fig. 3 .... Increase in Flow Caused by Excess of One Opening Over Another

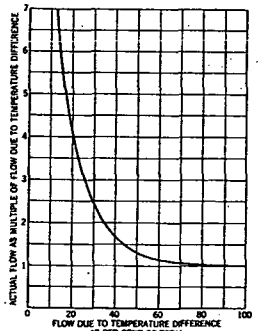


Fig. 4 .... Determination of Flow Caused by Combined Forces of Wind and Temperature Difference

through any opening is proportional to the square root of the sum of the heads acting on that opening.

When the two heads are about equal in value, and the ventilating openings are operated so as to coordinate them, the total air flow through the building is about 10 percent greater than that produced by either head acting independently under conditions ideal to it. This percentage decreases rapidly as one head increases over the other. The effect of the larger head will predominate.

The wind velocity and direction, the outdoor temperature, or the indoor distribution, cannot be predicted with certainty, and refinement in calculations is not justified; consequently, a simplified method can be used. This may be done by using the equations and calculating the flows produced by each force separately, under conditions of openings best suited for coordination of the forces. Then, by determining the ratio of the flow produced by temperature difference to the sum of the two flows, the actual flow due to the combined forces can be approximated from Fig. 4.

Example 1: Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per lb is used in this shop at the rate of 15 gph (7.75 lb per gal). Desired summer temperature difference is 10 deg, and the prevailing wind is 8 mph perpendicular to the long dimension. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution for Temperature Difference Only: The heat $H = 15 \times 7.75 \times 18{,}000 = 2{,}092{,}500$ Btu per hr.

By Equation 5, the air flow required to remove this heat with an average temperature difference of 10 deg is

$$Q = \frac{H}{1.08(t_i - t_o)} = \frac{2{,}092{,}500}{1.08 \times 10} = 193{,}750 \text{ cfm.}$$

This is equal to about 20 air changes per hour. From Equation 4, the inlet (or outlet) opening area should be

$$A = \frac{Q}{9.4\sqrt{h(t_i - t_o)}} = \frac{193{,}750}{9.4\sqrt{30 \times 10}} = 1190 \text{ sq ft}$$

The flow per square foot of inlet or outlet would be $193{,}750 \div 1190 = 162.5$ cfm, with all windows open.

Solution for Wind Only: With 1,190 sq ft of inlet openings distributed around the side walls, there will be about 400 sq ft in each long side and 195 sq ft in each end. The outlet area will be equally distributed on the two sides of the monitor, or 595 sq ft on each side. With the wind perpendicular to the long side, there will be 400 sq ft of opening in its path for inflow, and 595 in the lee side of the monitor for outflow with the windward side closed. The air flow, as calculated by Equation 3, will be:

$$Q = 0.60 \times 400 \times 704 = 168{,}960 \text{ cfm.}$$

This gives 16.9 air changes per hour, which should be more than ample when there is no heat to be removed.

Solution for Combined Heads: Since the windward side of the monitor is closed when the wind is blowing, the flow due to temperature difference must be calculated for this condition, using Fig. 3. This chart shows that, when inlets are twice the size of the outlets, in this case 1,190 sq ft in the side walls and 595 sq ft in the monitor, the flow will be increased 26.5 percent over that produced by equal openings. Using the smaller opening and the flow per square foot obtained previously, the calculated amount for this condition will be

$$595 \times 163 \times 1.265 = 122{,}700 \text{ cfm.}$$

Adding the two computed flows:

Temperature Difference	= 122,700	= 42	percent.
Wind	= 168,960	= 58	percent.
Total	291,660	= 100	percent.

From Fig. 4 it is determined that, when the flow due to temperature difference is 42 percent of the total, the actual flow due to the combined forces will be about 1.6 times that calculated for temperature difference alone, or 196,300 cfm.

The original flow due to temperature difference alone was 193,750 cfm with all openings in use. The effect of the wind is to increase this to 196,300 cfm, even though half of the outlets are closed.

A factor of judgment is necessary in the location of the openings in a building, especially those in the roof, where heat, smoke, and fumes are to be removed. Usually, windward monitor openings should be closed, but if the wind is low enough for the temperature head to overcome it, all windows may be opened.

TYPES OF OPENINGS

Types of openings may be classified as: (1) windows, doors, monitor openings, and skylights; (2) roof ventilators; (3) stacks connecting to registers; and (4) specially designed inlet or outlet openings. The various types and principles of operation are discussed in following paragraphs.

Windows, Doors, and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area, when open. Their movable parts are arranged to open in various ways; they may open by sliding either vertically or horizontally, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top, bottom, or side. Regardless of their design, the air flow per square foot of opening may be considered to be the same under the same conditions. The type of pivoting should receive consideration from the standpoint of weather protection, and certain types may be advantageous in controlling the distribution of incoming air. Deflectors are sometimes used for the same purpose and should be considered a part of the ventilation system.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weatherproof air outlet. They are actuated by the same

forces of wind and temperature head that create flow through other types of openings. The capacity of a ventilator depends upon four things: (1) its location on the roof; (2) the resistance it and the ductwork offer to air flow; (3) the height of draft; and (4) the efficiency of the ventilator in utilizing the kinetic energy of the wind for inducing flow by centrifugal or ejector action.

For maximum flow induction, a ventilator should be located on that part of the roof where it will receive the full wind without interference. If ventilators are installed within the suction region created by the wind passing over the building, or in a light court, or on a low building between two high buildings, their performance will be seriously influenced. Their normal ejector action, if any, may be completely lost.

The base of the ventilator should be of a taper-cone design to produce the effect of a bell-mouth nozzle whose coefficient of flow is considerably higher than that of a square-entrance orifice. If a grille is provided at the base, or if the base or structural members present obstructions, additional resistance is introduced, and the base opening should be increased in size accordingly.

Air inlet openings located at lower levels in the building, should be at least equal to, and preferably larger than, the combined throat areas of all roof ventilators. The air discharge by a roof ventilator depends on wind velocity and temperature difference, and, in general, its performance will be the same as any monitor opening located in the same place but, due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Roof ventilators may be classified as stationary, pivoting, or oscillating, and rotating. Generally, these have a round throat, but the continuous-ridge ventilator would fall in the stationary classification. When selecting roof ventilators, some attention should be given to ruggedness of construction, storm-proofing features, dampers and damper-operating mechanisms, possibility of noise, original cost, and maintenance.

Natural ventilation units may be used to supplement power-driven supply fans, and under favorable weather conditions it may be possible to stop the power-driven units. Units are not subject to code tests for ratings. Generally, they must be selected from manufacturers' tables. It is, therefore, very important to consider the reliability of the ratings used.

Controls

Gravity ventilators may have dampers controlled by hand, thermostat, or wind velocity, in combination with a fan. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself. The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks or vertical flues are really chimneys which function through the effects of the wind and temperature difference. Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction. With little or no wind, the chimney effect depends entirely on temperature difference to produce a removal of air from the rooms where the inlet openings are located.

GENERAL VENTILATION RULES

A few of the important considerations, in addition to those already outlined, are:

1. Inlet openings in the building should be well distributed, and should be located on the windward side near the bottom, while outlet openings are located on the leeward side near the top. Outdoor air will then be supplied to the zone to be ventilated.

2. Inlet openings should not be obstructed by buildings, trees, sign boards, etc., outdoors, nor by partitions indoors.

3. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.

4. In the design of window ventilated buildings, where the direction of the wind is quite constant and dependable, the orientation of the building, together with amount and grouping of ventilation openings, can be readily arranged to take full advantage of the force of the wind. Where the wind's direction is quite variable, the openings should be arranged in side walls and monitors so that, as far as possible, there will be approximately equal areas on all sides. Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force, and others to a suction force, and effective movement through the building will be assured.

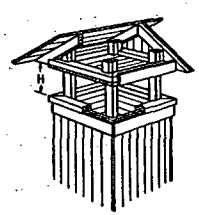
5. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.

6. In order that temperature difference may produce a motive force, there must be vertical distance between openings. That is, if there are a number of openings available in a building, but all are at the same level, there will be no motive head produced by temperature difference, no matter how great the difference might be.

7. In order that the force of temperature difference may operate to maximum advantage, the vertical distance between inlet and outlet openings should be as great as possible. Openings in the vicinity of the neutral zone are least effective for ventilation.

8. In the use of monitors, windows on the windward side should usually be kept closed, since, if they are open, the inflow tendency of the wind counteracts the outflow tendency of temperature difference. Openings on the leeward side of the monitor result in cooperation of wind and temperature difference.

9. In an industrial building where furnaces that give off heat and fumes are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then cooperate with temperature difference,



The opening H should equal one-half the least dimension of the flue. Heavy insulation of the level deck is essential.

Fig. 5 . . . . Recommended Type of Cover for Wooden Outlet Flue

to provide for the most active and satisfactory removal of the heat and gas-laden air.

10. In case it is impossible to locate furnaces in the windward end, that part of the building in which they are to be located should be built higher than the rest, so that the wind, in splashing therefrom, will create a suction. The additional height also increases the effect of temperature difference to cooperate with the wind.

11. The intensity of suction, or the vacuum produced by the jump of the wind, is greatest just back of the building face. The area of suction does not vary with the wind velocity, but the flow due to suction is directly proportional to wind velocity.

12. Openings much larger than the calculated areas are sometimes desirable, especially when an increase in occupancy may occur, or when extremely hot days may be anticipated. In the former case, free openings should be located at the level of occupancy for psychological reasons.

13. In single story industrial buildings, particularly those covering large areas, natural ventilation must be accomplished by taking air in and out of the roof openings. Openings in the pressure zones can be used for inflow, and openings in the suction zone, or openings in zones of less pressure, can be used for outflow. The ventilation is accomplished by the manipulation of openings to get air flow through the zones to be ventilated.

VENTILATION OF ANIMAL SHELTERS[12]

Animal shelters require ventilation to remove moisture, odors and, in the case of dairy stables, excess heat.

Outlets. Outlet flues for natural draft systems should be round or approximately square. A thermal resistance ($1/U$) of not less than two is required in their side walls. They should extend at least two feet above the highest part of the roof. Only one outlet is recommended for each room or pen. The use of several outlet flues may result in excessive up-drafts in some flues, and down-drafts in others.

If flues have roofs or covers, these should be high enough to provide unobstructed openings on all sides, equal in height to one-half the least dimension of the flue, Fig. 5. A level, heavily insulated ceiling under the flue roof, and over the entire area of the flue, is important.

Inlets. Inlets should direct the incoming air vertically upward so as to avoid drafts on the animals, and to insure immediate mixing of incoming air with the room air. A reasonably uniform distribution around the stable or pen is desirable. Inlet flues, that deliver air close to the side walls, stimulate convection currents, which is desirable. When so placed, they also tend to bathe the side walls with cool air, thus reducing temperature difference between the inside and outside of the wall.

From the standpoint of air movement, insulation of inlet flues is not important. Condensation is, however, likely to occur on them, unless the thermal resistance of their walls is at least two.

Controls or throttling devices in inlet flues are seldom required. If used, they are best applied to the inlets and limited to the side of the building facing prevailing winter winds. They should be so made that an opening, at least one inch wide by the width of the flue, will always remain open.

Amounts of heat and water produced by livestock vary not only with the different kinds of animals, but also with age, weight, feed consumption, and production. These facts, and the vagaries of the weather, make exact calculations impossible. The following practical recommendations are based on numerous, carefully checked observations.

It is desirable to keep the relative humidity of livestock shelters below 85 percent. Temperatures may be as indicated in the discussion for each kind of animal.

Dairy Stables

The most commonly accepted temperatures for dairy stables, where cows are confined in stanchions or tie stalls, are from 45 to 55 F. These temperatures are readily maintained in winter weather by the body heat of the herd in well constructed, well stocked, and well ventilated stables. Stable volume in excess of 600 cu ft, and exposed wall area in excess of 130 sq ft per 1000 lb animal weight are, in general, undesirable.

Side walls should have an overall thermal resistance of from 2 to 5, depending on the temperature zone and wind exposure. Thermal resistance of the ceiling should be 50 percent greater than that of the side walls. In stables of this size and so insulated, a ventilation rate of 3200 to 3800 cfh per (1000 lb of animal) usually insures good conditions.

The following recommendations do not apply to so-called pen stables or loafing barns in which there is a thick manure and bedding pack on the floor, and in which doors are normally kept open.

Outlets. One flue will serve a stable 200 ft long. In stables over 120 ft long, the flue should be about midway between the ends or, if the stable is L-shaped, near the angle. In shorter stables, it may be at any convenient location.

The exhaust point in the stable should be not more than 18 in. above the floor. This permits removal of only the coolest air, and prevents rapid fluctuations in stable temperature.

A basic rule for finding the cross-section of the outlet flue is

$$A_o = \frac{176N}{\sqrt{h}} \qquad (6)$$

where

A_o = area of the outlet flue, square inches.
N = weight of animal population, thousands of pounds.
h = vertical distance from top of inlet flues to top of outlet flue, feet.

For large flues the flue area obtained from the basic formula may well be reduced according to the chart, Fig. 6, because of a decrease in friction.

Example 2: Assume a stable in which the vertical height from the top of the inlets to the top of the outlet flue is 32.5 ft, and in which 38 cows, averaging 1300 lb, will be housed. Determine required size of outlet flue.

Solution: From Equation 6

$$A_o = \frac{176 \times (38 \times 1.300)}{\sqrt{32.5}} = 1525 \text{ sq. in.}$$

From Fig. 6 the factor to be applied to this area is 94.5

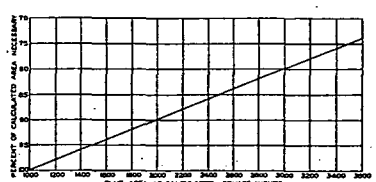


Fig. 6 . . . . Modification of Flue Area for Outlets Exceeding 1000 Sq In.

percent. Therefore, the area of flue required is 1525 × 0.945 = 1441 sq in.

Inlets. Inlet flues, each approximately 60 sq in. in area, have given good results. One such flue should be provided for each 3500 lb animal weight. They should deliver air from points 12 to 18 in. below the ceiling.

Sheep Barns

Shelters used for breeding and feeding stock usually have enough openings, so that no special provision for ventilation is required. Barns for winter lambing flocks, however, require ventilation systems. Fermentation in the floor pack of manure produces heat, vapor, and odor. These must be added to the ventilation load regularly produced by the animals.

Outlets. In practice, results obtained by the basic formula, Equation 7, when modified by the use of the chart, Fig. 6, have given good results:

$$A_o = \frac{4A_f}{\sqrt{h}} \quad (7)$$

where

A_o = area of the outlet, square inches.
A_f = floor area, square feet.
h = vertical height from top of inlet openings to top of outlet flue, feet.

The bottom of the outlet flue should be 15 to 24 in. above the surface of the manure pack.

Inlets. Provide one inlet, 60 sq in. in area, for each 150 sq ft of floor area. Inlets should be well distributed around the side walls, and designed to deliver air near the ceiling (see section on Dairy Stables).

Swine Barns

Community swine barns, because of the extent of slop feeding and the absence of daily cleaning of the pens, are the most difficult farm buildings to ventilate satisfactorily. Farrowing pens, to which supplemental heat is supplied, present less of a problem. In all cases, good floor drainage to remove urine and excess spilled water is important.

Temperatures of from 50 to 55 F are usually recommended for farrowing pens. It is desirable to maintain temperatures above freezing in all other pens in community houses. For barns that are well stocked and adequately ventilated, this requires walls with an overall thermal resistance of from 3 to 6, and ceilings with 40 to 50 percent greater resistance.

Outlets. The outlet flue should draw air from a level of 15 to 18 in. above the floor. Equation 8 is the basic formula for flue area, and gives reasonably good results when modified according to the chart, Fig. 6.

$$A_o = \frac{5 \times A_f}{\sqrt{h}} \quad (8)$$

Inlets. At least one inlet flue should be used for each pen. Swine are given or select definite nesting places, and care must be exercised to avoid having inlets located over them.

Total inlet area should be approximately 70 percent of outlet area. The area of individual inlets is best determined from the total area required, and the number that can be so installed as to meet previous specifications. Inlets should deliver air from points 12 to 15 in. below the ceiling, or from a level deflector on a sloping ceiling.

Poultry Laying Houses

From the standpoint of ventilation, poultry laying houses may be divided into cold houses and warm houses. The former are uninsulated, except in the ceiling. The indoor-outdoor temperature difference is seldom more that 5 F deg. In the warm house, because of insulation or supplemental heat, the temperature seldom falls below 32 F, and is usually above 45 F.

Outlets. The outlet flue is best placed near the middle of the pen. It is advisable to limit the length of pens to 80 ft.

The area of the outlet flue may be determined from Equation 9, and modified according to the chart, Fig. 6.

$$A_o = \frac{2.5 \times A_f}{\sqrt{h}} \quad (9)$$

For a cold house, the bottom of the flue should be at the level of the insulated ceiling. In warm houses, the bottom of the flue should be 12 to 15 in. above the level of the floor litter.

Inlets. Inlets may be approximately 60 sq in. in area. The total inlet area may be equal to 70 percent of the outlet area. Inlets in cold and in warm houses should deliver air from points 12 to 24 in. above the floor. In cold houses which normally do not have storm sash, windows may be raised enough to give the desired area, and be fitted with baffle boards to direct the air straight and upward.

In houses less than 20 ft in width, all inlets may be on one side. In wider houses, a rather uniform distribution of inlets is essential.

GARAGE VENTILATION

Because of hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be over-emphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. The ASHAE Code of Minimum Requirements for Heating and Ventilating Garages,[14] adopted in 1935, specifies that openings for natural ventilation shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 percent of the floor area. The code further states that where it is impracticable to operate such a system of natural ventilation, a mechanical system shall be used, and shall provide for either the supply of 1 cfm of outdoor air for each square foot of floor area, or for exhaust of the same amount of air, discharging it to the outdoors as a means of flushing the garage.

Cooperative research[15] on garage ventilation, undertaken by the ASHAE Committee on Research at Washington University, St. Louis, Mo., and at the University of Kansas, and tests conducted at the ASHAE Research Laboratory, have resulted in authoritative papers on the subject.

Some of the conclusions based on work at these laboratories are:

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line, than does downward ventilation for the same rate of carbon monoxide production, air change, and the same temperature at the 30-in. level.

2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.

3. In the average case, upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage, than that obtained with mixing of the exhaust gases and the air supplied. However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing, and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.

4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cfh, with an average rate of 35 cfh.

5. An air change of 350,000 cfh per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

Individual Exhaust Ducts

Motor vehicle servicing areas may be provided with underground or overhead exhaust systems discharging outdoors. Each service location should be provided with a flexible branch duct, the inlet end of which is sized to slip over the tail pipe of the engine, or over any deflector attached to the tail pipe. The minimum branch size and ventilation rate for the largest class of vehicle regularly serviced at each location are shown in Table 5.

Table 5 .... Minimum Tail Pipe Connection Size and Ventilation Rate per Vehicle

Vehicle	Ventilation Rate, cfm	Flexible Pipe, Dia, in.	Diameter of Rigid Branch Connection to Flexible Pipe, in.
Auto up to 200 hp.....	100	3	4
Auto or truck above 200 hp.....	200	4	4½
Diesel truck or bus	400	4½	5½

Where each tail pipe exhaust branch is provided with an automatic device to close the branch inlet when it is not in use, the capacity of the fan for the system should be based on the maximum number of branches in simultaneous use. Where the branch inlets are open when not in use, the system capacity should be based on the total number of branches connected to the main.

Individual straight lengths of gas tight duct, not over 20 ft in length, terminating outdoors, may be provided where exhaust systems are not installed. The duct should fit tightly over the tail pipe and be of a diameter not less than the diameter of the tail pipe.

REFERENCES

[1] D. R. Bahnfleth, T. D. Moseley, and W. S. Harris: ASHAE Research Report No. 1614—Measurement of infiltration in two residences, Part I—Technique and measured infiltration (ASHAE Transactions, Vol. 53, 1957, p. 439).

[2] D. R. Bahnfleth, T. D. Moseley, and W. S. Harris: ASHAE Research Report No. 1615—Measurement of infiltration in two residences, Part II—Comparison of variables affecting infiltration (ASHAE Transactions, Vol. 63, 1957, p. 453).

[3] T. C. Min: Winter infiltration through swinging-door entrances in multi-story buildings (ASHAE Journal Section, *Heating, Piping and Air Conditioning*, February 1958, p. 121).

[4] F. C. Houghten and Margaret Ingels: ASHVE Research Report No. 736—Infiltration through plastered and unplastered brick walls (ASHVE Transactions, Vol. 33, 1927, p. 377). G. L. Larson, D. W. Nelson, and C. Braats: ASHVE Research Report No. 826—Air infiltration through various types of brick wall construction (ASHVE Transactions, Vol. 35, 1929, p. 183). G. L. Larson, D. W. Nelson, and C. Braats: ASHVE Research Report No. 851—Air infiltration through various types of brick wall construction (ASHVE Transactions, Vol. 36, 1930, p. 99). G. L. Larson, D. W. Nelson, and C. Braats: ASHVE Research Report No. 868—Air infiltration through various types of wood frame construction (ASHVE Transactions, Vol. 36, 1930, p. 397).

[5] F. C. Houghten and C. C. Schrader: ASHVE Research Report No. 686—Air leakage through the openings in buildings (ASHVE Transactions, Vol. 30, 1924, p. 105). C. C. Schrader: ASHVE Research Report No. 704—Air leakage around window openings (ASHVE Transactions, Vol. 30, 1924, p. 313). F. C. Houghten and M. E. O'Connell: ASHVE Research Report No. 803—Air leakage studies on metal windows in a modern office building (ASHVE Transactions, Vol. 34, 1928, p. 321). F. C. Houghten and M. E. O'Connell: ASHVE Research Report No. 815—Air leakage through a pivoted metal window (ASHVE Transactions, Vol. 34, 1928, p. 519). W. M. Richtmann and C. Braats: ASHVE Research Report No. 817—Effect of frame calking and storm windows on infiltration around and through windows (ASHVE Transactions, Vol. 34, 1928, p. 547). G. L. Larson, D. W. Nelson, and R. W. Kubasta: ASHVE Research Report No. 909—Air infiltration through double-hung wood windows (ASHVE Transactions, Vol. 37, 1931, p. 571). J. E. Emswiler and W. C. Randall: The weathertightness of rolled section steel windows (ASHVE Transactions, Vol. 34, 1928, p. 527). J. E. Emswiler and W. C. Randall: Pressure differences across windows in relation to wind velocity (ASHVE Transactions, Vol. 36, 1930, p. 83). D. O. Rusk, V. H. Cherry, and L. Boelter: Air infiltration through steel framed windows (ASHVE Transactions, Vol. 39, 1933, p. 169).

[6] A. P. Kratz and S. Konzo: Fuel saving resulting from the use of storm windows and doors (ASHVE Transactions, Vol. 42, 1936, p. 87).

[7] A. M. Simpson and K. B. Atkinson: The infiltration problem of multiple entrances (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, June 1936, p. 345). A. M. Simpson: Infiltration characteristics of entrance doors (*Refrigerating Engineering*, June 1936).

[8] F. C. Houghten and J. L. Blackshaw: Indices of air change and air distribution (ASHVE Transactions, Vol. 39, 1933, p. 261).

[9] F. C. Houghten, J. L. Blackshaw, and Carl Gutberlet: ASHVE Research Report No. 994—Wind velocities near a building and their effect on heat loss (ASHVE Transactions, Vol. 40, 1934, p. 387). F. C. Houghten and Carl Gutberlet: ASHVE Research Report No. 1069—Heating requirements of an office building as influenced by the stack effect. (ASHVE Transactions, Vol. 43, 1937, p. 437). H. L. Alt: Flue action in high buildings (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, May 1932, p. 376). Axel Marin: Influence of stack effect on the heat loss in tall buildings (ASHVE Transactions, Vol. 40, 1934, p. 377).

[10] J. E. Emswiler: Neutral zone in ventilation (ASHVE Transactions, Vol. 32, 1926, p. 59).

[11] *State Building Construction Code Applicable to One- and Two-Family Dwellings* (New York State Building Code Commission).

[12] W. C. Randall and E. W. Conover: Predetermining airation of industrial buildings (ASHVE Transactions, Vol. 37, 1931, p. 605).

[13] F. L. Fairbanks: Dairy stable ventilation (ASHVE Transactions, Vol. 34, 1928, p. 181). A. J. Offner: Cow barn ventilation (ASHVE Transactions, Vol. 39, 1933, p. 149). For additional information on this subject see: M. A. R. Kelley: *Technical Bulletin* (U. S. Department of Agriculture, 1930). F. L. Fairbanks: Air conditioning of farm buildings (*Agricultural Engineering*, November 1937, p. 485). F. L. Fairbanks and A. M. Goodman: *Dairy Stable Ventilation* (Cornell University, *Cornell Extension Bulletin* No. 151, revised 1949). F. L. Fairbanks and A. M. Goodman: *The Ventilation of Poultry Laying Houses* (Cornell University, *Cornell Extension Bulletin* No. 315, revised 1950).

[14] Code of minimum requirements for heating and ventilating garages (ASHVE Transactions, Vol. 41, 1935, p. 30). W. C. Randall and L. W. Leonhard: Airation study of garages (ASHVE Transactions, Vol. 36, 1930, p. 233). A. S. Langsdorf

and R. R. Tucker: ASHVE RESEARCH REPORT No. 874—Carbon monoxide concentration in garages (ASHVE TRANSACTIONS, Vol. 36, 1930, p. 511). F. C. Houghten and Paul McDermott: ASHVE RESEARCH REPORT No. 935—Carbon monoxide distribution in relation to the ventilation of an underground ramp garage (ASHVE TRANSACTIONS, Vol. 38, 1932, p. 439). F. C. Houghten and Paul McDermott: ASHVE RESEARCH REPORT No. 934—Carbon monoxide distribution in relation to the ventilation of a one-floor garage (ASHVE TRANSACTIONS, Vol. 38, 1932, p. 424). F. C. Houghten and Paul McDermott: ASHVE RESEARCH REPORT No. 967—Carbon monoxide distribution in relation to the heating and ventilation of a one-floor garage (ASHVE TRANSACTIONS, Vol. 39, 1933, p. 395). A. H. Sluss, E. K. Campbell, and L. M. Farber: Carbon monoxide surveys of two garages (ASHVE TRANSACTIONS, Vol. 40, 1934, p. 263).

BIBLIOGRAPHY

J. B. Dick: *Experimental Studies in Natural Ventilation of Houses* (Department of Scientific and Industrial Research, *Building Research Station Note* No. 180, Garston, Watford, England).

J. B. Dick: Measurement of ventilation using tracer gas technique (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May 1950, p. 131).

C. W. Coblents and P. R. Achenbach: Design and performance of a portable infiltration meter (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 477).

Garage ventilation

R. R. Sayers, A. C. Fieldner, W. P. Yant, and B. G. Thomas: *Experimental Studies on the Effect of Ethyl Gasoline and Its Combustion Products* (*U. S. Bureau of Mines Monograph* No. 2, 1927).

A. C. Fieldner, Yandell Henderson, J. W. Paul, R. R. Sayers, et al: Ventilation of vehicular tunnels (ASHVE JOURNAL, January–December 1926).

S. H. Ash and L. L. Naus: *Use of Diesel Engines in Tunnels* (*U. S. Bureau of Mines Information Circular* No. 7222, 1942).

L. B. Berger: *Ventilation Involved in the Use of Gasoline Powered Equipment in Enclosed Spaces* (*U. S. Bureau of Mines Information Circular* No. 7404, 1947).

J. C. Holtz, L. B. Berger, M. A. Elliott, and H. H. Schrenk: *Diesel Engines Underground: Composition of Exhaust Gas from Engines in Proper Mechanical Condition* (*U. S. Bureau of Mines Report of Investigations* No. 3508, 1940).

L. B. Berger, M. A. Elliott, J. C. Holtz, and H. H. Schrenk: *Diesel Engines Underground: Use of Diesel Locomotives in Construction of the Delaware Aqueduct: Effect of Exhaust Gases upon Quality of Tunnel Air* (*U. S. Bureau of Mines Report of Investigations* No. 4032, 1947).

CHAPTER 12

HEATING LOAD

General Procedure, Design Outdoor Weather Conditions, Indoor Temperatures, Attic Temperatures, Temperatures in Unheated Spaces, Ground Temperatures, Basement Temperatures and Heat Loss, Heat Losses from Floor Slabs, Transmission Heat Loss, Infiltration Heat Loss, Selection of Wind Speeds, Auxiliary Heat Sources, Intermittently Heated Buildings, Residence Heat Loss Problems

PRIOR to designing a heating system, an estimate must be made of the maximum probable heat loss of each room or space to be heated, based on maintaining a selected indoor air temperature during periods of design outdoor weather conditions. The heat losses may be divided into two groups, namely: (1) the transmission losses or heat transmitted through the confining walls, floor, ceiling, glass, or other surfaces; and (2) the infiltration losses or heat required to warm outdoor air which leaks in through cracks and crevices, around doors and windows, or through open doors and windows, or heat required to warm outdoor air used for ventilation.

GENERAL PROCEDURE

The general procedure for calculating heat losses of a structure is:

1. Select the design outdoor weather conditions: temperature, wind direction, and wind speed. The data on climatic conditions given in Tables 1 and 2 will be helpful, but should be used with judgment as suggested in the section Design Outdoor Weather Conditions.

2. Select the indoor air temperature that is to be maintained in each room during the coldest weather. (See Table 3.)

3. Estimate temperatures in adjacent unheated spaces and the attic. (See section Attic Temperatures.) The attic temperature need not be estimated if the combined roof and ceiling coefficient is used.

4. Select or compute the heat transmission coefficients for outside walls and glass; also for inside walls, floors, or top-floor ceilings, if these are next to unheated space; include roof if next to heated space. (See Chapter 9. If the design wind speed is appreciably different from 15 mph, the appropriate change in the heat transmission coefficients in Tables 5 to 15 of Chapter 9 can be found in Table 20 of that chapter.)

5. Determine net area of outside wall, glass, and roof next to heated spaces, as well as any cold walls, floors, or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building, using inside dimensions.

6. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling, and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet, and the temperature difference between the indoor and outdoor air. (See Items 1, 2, and 3.)

7. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack, wind speed, and the temperature difference between the indoor and outdoor air; the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 11.)

8. When positive ventilation using outdoor air is provided by an air heating or an air-conditioning unit, the heat required to warm the outdoor air to room temperature must be provided by the unit; if mechanical exhaust from the room is provided, in amount equal to the outdoor air drawn in by the unit, the natural infiltration losses must also be provided for by the unit. If no mechanical exhaust is used, and the outdoor air supply equals or exceeds the amount of natural infiltration that would occur without ventilation, the natural infiltration may be neglected.

9. The sum of the heat losses due to transmission (Item 6) through the outside walls and glass, as well as through any cold floors, ceilings, or roof, plus the heat equivalent (Item 7) of the cold air entering by infiltration, or required to replace mechanical exhaust, represents the total heat loss equivalent for any building.

10. In buildings that have a reasonably steady internal heat release of appreciable magnitude from sources other than the heating system, a computation of this heat release under design conditions should be made for deduction from the total of the heat losses computed in items 1–9. This is especially important for heating systems of high initial cost or those for which a demand charge is based on installed capacity.

DESIGN OUTDOOR WEATHER CONDITIONS

The ideal solution to the basic problem which confronts the designer of a heating system is to design a plant that has a capacity at maximum output just equal to the heating load which develops when the most severe weather conditions for the locality occur.

In most cases, economics interferes with the attainment of this ideal. Studies of weather records show that the most severe weather conditions do not repeat themselves every year. If heating systems were designed with adequate capacity for the maximum weather conditions on record, there would be considerable excess capacity during most of the operating life of the system.

The weather records of principal cities of the United States have been analyzed to determine the probability of occurrence of certain low temperatures. The results of this analysis appear in Table 1. Data for Canadian cities obtained by a different analytical method appear in Table 2.

Five probabilities of outdoor temperatures are offered for American cities and four for Canadian cities. The analytical methods used for determining the several design temperatures cited in Table 1 for U. S. and Table 2 for Canadian cities are described in the footnotes of the tables.

In many cases, occasional failure of a heating plant to maintain a pre-selected indoor design temperature during brief periods of severe weather is not critical. However, the successful completion of some industrial or commercial processes may depend upon close regulation of indoor temperatures. These are special cases and require extra study before assigning design temperatures.

Before selecting an outdoor design temperature from

(Text continued on p. 155)

Table 1 .... Winter Climatic Conditions—United States[a]

Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7	Col. 8	Col. 9
State	Station[b]	Elevation[c] Ft.	Design Winter Temperatures—F Frequency of Recurrence, Years: Once in 40	Once in 20	Once in 13	Once in 10	Once in 5	Avg Wind Speed, Dec, Jan, Feb[d] Mph
Ala	Anniston A	599	5	9	12	14	18	—
	Birmingham A	610	6	10	12	14	18	8.0
	Mobile C	143	17	20	22	24	27	9.9
	Montgomery C	293	11	15	18	19	23	7.5
Ariz	Flagstaff A	6993	−12	−7	−4	−2	3	7.7
	Phoenix C	1122	33	35	36	37	39	5.4
	Yuma A	199	34	36	38	39	41	6.7
Ark	Bentonville C	1295	−8	−4	−1	0	5	6.7
	Fort Smith A	458	0	4	6	8	12	8.3
	Little Rock A	257	2	6	8	10	15	8.3
Calif	Eureka C	115	28	30	32	33	35	7.3
	Fresno A	331	29	31	32	32	34	5.4
	Independence C	—	6	10	12	14	19	—
	Los Angeles C	534	37	39	41	41	44	6.4
	Sacramento C	116	27	29	30	31	33	7.2
	San Diego A	19	41	42	43	44	46	6.3
	San Francisco C	164	34	36	37	38	40	7.5
	San Jose C	100	37	38	38	38	39	—
Colo	Denver C	5398	−19	−15	−12	−10	−5	7.5
	Durango C	6550	−12	−8	−6	−4	−1	5.0
	Grand Junction A	4587[e]	−11	−6	−3	−1	4	4.4
	Leadville C	10,200	−14	−11	−9	−8	−5	11.1
	Pueblo A	4799	−24	−18	−14	−12	−5	7.9
Conn	Bridgeport A	7	−6	−3	−1	1	4	10.5
	Hartford A	15	−7	−4	−3	1	3	8.7
	New Haven A	6	−5	−2	0	1	5	9.4
	New London C	15	−2	1	3	4	7	9.3
	Norwalk C	37	−7	−3	−1	0	4	10.5
D. C	Washington C	128	5	8	10	11	14	7.8
Fla	Jacksonville C	104	24	27	28	30	33	9.0
	Key West C	23	50	52	53	54	56	10.6
	Pensacola C	67	19	22	24	25	29	10.9
	Tampa A	24	32	35	36	38	41	8.6
Ga	Atlanta A	1054	5	9	11	13	17	11.7
	Augusta A	143	15	18	20	21	25	6.5
	Macon A	356	14	18	20	21	24	6.7
	Savannah A	48	19	22	24	25	28	9.6
Idaho	Boise A	2842	−20	−14	−10	−7	0	9.1
	Lewiston A	1413	−24	−16	−12	−8	0	4.1
	Pocatello A	4444	−27	−20	−17	−14	−7	8.9
Ill	Cairo C	319	−8	−3	0	2	7	9.8
	Chicago C	601[e]	−17	−13	−11	−9	−5	11.7
	Peoria A	660	−20	−15	−13	−11	−6	8.3
Ind	Evansville A	385	−12	−7	−4	−2	3	9.6
	Fort Wayne A	801	−13	−9	−7	−5	−2	10.4
	Indianapolis C	816	−14	−10	−8	−6	−1	11.3
	Lafayette A	600	−15	−11	−9	−7	−2	11.3
	Terre Haute A	578	−12	−8	−6	−4	0	10.2
Iowa	Charles City C	1023	−28	−24	−21	−20	−16	7.9
	Davenport C	648	−18	−14	−12	−10	−5	10.6
	Des Moines C	805	−19	−15	−13	−12	−8	10.1
	Dubuque C	740	−22	−18	−16	−14	−9	7.1
	Keokuk C	637	−21	−16	−13	−11	−6	8.3
	Sioux City A	1093	−22	−18	−16	−15	−11	11.5
Kan	Concordia C	1425	−17	−13	−11	−9	−5	7.7
	Dodge City A	2594	−16	−11	−9	−7	−2	10.6

Table 1 .... Winter Climatic Conditions—United States[a] (Continued)

Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7	Col. 8	Col. 9
State	Station[b]	Elevation[c] Ft.	Design Winter Temperatures—F Frequency of Recurrence, Years: Once in 40	Once in 20	Once in 13	Once in 10	Once in 5	Avg Wind Speed, Dec, Jan, Feb[d] Mph
Kan	Iola C	963	−15	−10	−7	−5	0	8.3
	Topeka C	991	−15	−11	−8	−7	−2	9.2
	Wichita A	1372	−13	−9	−6	−4	0	12.4
Ky	Bowling Green A	536	−5	−1	2	4	9	9.5
	Hopkinsville C	540	−5	0	2	4	8	9.5
	Irvington C	646	−7	−3	0	1	6	9.7
	Lexington A	979	−9	−5	−2	−1	4	13.3
	Louisville C	563	−9	−5	−2	0	4	9.8
	Shelbyville C	825	−9	−5	−2	0	5	13.3
La	New Orleans C	85	21	24	26	27	30	8.6
	Shreveport A	179	8	12	14	16	20	8.8
Me	Eastport C	100	−14	−11	−9	−8	−5	12.6
	Portland A	61	−15	−11	−9	−7	−3	10.4
Md	Baltimore C	114	3	6	8	9	13	8.2
Mass	Boston C	356	−6	−2	0	1	4	12.4
Mich	Alpena C	615	−16	−12	−10	−8	−5	11.0
	Detroit A	619	−10	−6	−4	−3	1	12.0
	Escanaba C	645	−25	−21	−18	−17	−12	9.5
	Grand Haven C	622	−8	−5	−3	−1	2	12.4
	Grand Rapids C	706	−9	−6	−4	−2	1	12.1
	Houghton C	1081	−24	−20	−18	−16	−11	8.7
	Lansing C	861	−14	−10	−8	−7	−3	9.8
	Ludington C	650	−13	−9	−7	−5	−1	11.9
	Marquette C	721	−22	−18	−16	−14	−10	10.6
	Sault Ste. Marie A	721	−24	−21	−19	−17	−14	8.9
Minn	Duluth A	1133	−32	−29	−27	−25	−22	13.4
	Minneapolis C	945	−28	−25	−23	−21	−18	11.3
	Moorehead C	940	−35	−32	−29	−27	−23	9.7
	St. Paul A	830	−28	−25	−23	−21	−18	9.5
Miss	Corinth C	470	−2	3	6	8	14	11.6
	Meridian A	294	8	12	14	16	20	6.3
	Vicksburg C	316	9	13	15	17	21	8.3
Mo	Columbia C	739	−16	−11	−9	−7	−2	8.9
	Hannibal C	759	−19	−14	−12	−9	−4	9.0
	Kansas City C	741[e]	−15	−11	−8	−6	−2	10.3
	St. Louis C	646	−12	−8	−5	−3	1	11.8
	Springfield A	1270	−12	−8	−5	−4	1	11.0
Mont	Billings A	3584	−42	−35	−31	−28	−21	12.4
	Havre C	2498	−49	−43	−39	−36	−29	9.4
	Helena C	4175[e]	−53	−44	−39	−36	−27	7.4
	Kalispell A	2965	−43	−35	−31	−27	−19	5.2
	Miles City C	2400	−44	−38	−35	−32	−26	5.6
Neb	Lincoln C	1189	−22	−18	−15	−14	−9	10.6
	North Platte A	2789	−21	−17	−15	−13	−9	7.9
	Omaha A	978	−24	−19	−17	−15	−10	9.7
	Valentine C	2627	−27	−23	−21	−19	−15	9.2
Nev	Reno A	4402	−4	0	3	5	9	6.0
	Winnemucca C	4293	−17	−12	−9	−6	0	8.1
N. H	Concord A	339	−17	−13	−11	−9	−5	6.2
N. J	Atlantic City C	45	3	6	8	9	12	15.8
	Charlotteburg C	—	−6	−3	−1	1	4	—
	Elizabeth C	33	−4	0	2	3	7	11.2
	Flemington C	134	−2	1	3	5	8	10.5
	New Brunswick C	80	−1	2	4	5	9	10.5

Table 1 .... Winter Climatic Conditions—United States[a] (Continued)

Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7	Col. 8	Col. 9
State	Station[b]	Elevation[c] Ft.	Design Winter Temperatures—F Frequency of Recurrence, Years					Avg Wind Speed, Dec, Jan, Feb[d] Mph
			Once in 40	Once in 20	Once in 13	Once in 10	Once in 5	
N. J.	Plainfield C	120	−1	2	4	5	8	11.2
	Somerville C	75	−3	0	2	3	7	10.5
	Trenton C	144	−4	0	2	3	7	10.9
N. M.	Albuquerque A	5310	3	6	8	10	14	7.3
	Roswell A	3612	−4	1	4	6	11	7.1
	Santa Fe C	7045	−2	1	3	4	7	7.0
N. Y.	Albany C	114	−16	−12	−9	−8	−4	10.5
	Binghamton C	915	−12	−9	−7	−5	−2	6.8
	Buffalo A	693	−11	−7	−5	−4	0	17.1
	Canton C	458	−28	−25	−22	−21	−17	10.5
	Ithaca C	888	−9	−6	−4	−3	0	11.3
	New York C	425	1	3	5	6	9	16.8
	Oswego C	363	−12	−9	−7	−5	−1	12.1
	Rochester A	543	−9	−6	−4	−3	1	9.6
	Syracuse A	424	−16	−12	−10	−8	−4	11.2
N. C.	Asheville C	2280	−2	2	5	6	11	9.5
	Charlotte C	809	9	12	14	16	19	7.3
	Hatteras C	4	16	19	21	22	25	14.3
	Raleigh C	405	9	12	14	15	18	7.9
	Wilmington C	78	16	19	20	22	25	9.4
N. D.	Bismarck A	1650	−38	−34	−31	−29	−24	9.1
	Devils Lake C	1481	−37	−34	−32	−30	−27	10.1
	Grand Forks C	832	−37	−34	−31	−30	−26	13.2
	Williston C	1919	−44	−38	−35	−33	−27	8.6
Ohio	Cincinnati C	772	−9	−5	−3	−1	3	8.5
	Cleveland C	669	−11	−7	−5	−3	1	14.7
	Columbus C	812	−9	−5	−3	−2	2	11.6
	Dayton A	1002	−10	−6	−4	−2	2	11.1
	Sandusky C	608	−10	−6	−4	−2	2	11.0
	Toledo A	622	−11	−7	−5	−4	0	12.1
Okla.	Oklahoma City C	1264	−9	−4	−1	1	6	11.5
Ore.	Baker C	3501	−24	−17	−14	−11	−5	5.6
	Portland C	98	1	7	10	12	18	7.3
	Roseburg C	523	14	17	19	20	24	3.9
Pa.	Erie C	771	−9	−5	−3	−2	2	13.6
	Harrisburg A	335	0	2	4	5	8	7.6
	Philadelphia C	18	1	4	6	7	11	11.0
	Pittsburgh C	929	−9	−6	−3	−2	2	11.6
	Reading C	311	−2	1	3	4	8	9.0
	Scranton C	877	−8	−4	−2	−1	3	7.6
R. I.	Block Island C	46	3	5	7	8	10	20.6
	Kingston C	100	−5	−2	0	1	4	11.7
	Providence C	77	−4	−1	1	2	5	12.1
S. C.	Charleston C	59	17	20	22	23	27	10.5
	Columbia C	401	14	17	19	20	24	8.0
S. D.	Huron C	1342	−27	−23	−21	−20	−17	10.7
	Pierre A	1718	−28	−25	−22	−21	−17	11.7
	Rapid City A	3165	−28	−24	−22	−20	−16	8.0
Tenn.	Chattanooga C	952	0	5	8	10	14	7.7
	Knoxville A	950	−3	2	5	7	12	7.2
	Memphis C	348	0	4	6	8	13	9.3
	Nashville C	714	−4	0	3	5	10	9.8
Tex.	Abilene A	1751	0	5	7	9	14	10.1
	Amarillo A	3590	−10	−5	−2	0	5	12.1
	Corpus Christi A	40	16	20	23	24	29	11.0
	Dallas A	487	1	6	8	10	15	10.6

Table 1 .... Winter Climatic Conditions—United States[a] (Concluded)

Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7	Col. 8	Col. 9
State	Station[b]	Elevation[c] Ft.	Design Winter Temperatures[e]—F Frequency of Recurrence, Years					Avg Wind Speed, Dec, Jan, Feb[d] Mph
			Once in 40	Once in 20	Once in 13	Once in 10	Once in 5	
Tex.	El Paso A	3920	15	18	20	21	25	9.0
	Fort Worth C	708	0	5	8	10	15	10.5
	Galveston C	128	16	20	23	24	29	11.2
	Houston C	198	13	17	19	21	25	10.5
	Palestine C	555	4	8	11	13	18	8.0
	San Antonio A	782	12	16	19	20	25	8.3
	Taylor C	570	5	9	12	14	19	9.8
Utah	Salt Lake City C	4346	−8	−3	−1	1	6	7.8
Vt.	Burlington A	331	−23	−19	−17	−15	−11	11.6
	Northfield C	840	−24	−21	−19	−17	−13	8.8
Va.	Cape Henry C	24	13	16	17	18	21	14.0
	Lynchburg A	947	7	10	11	12	15	8.1
	Norfolk C	91	11	14	15	17	20	12.1
	Richmond C	180	6	9	11	12	16	8.1
	Wytheville C	2400	−3	1	3	4	8	7.3
Wash.	Seattle C	104	9	13	15	17	21	9.8
	Spokane A	2357	−28	−20	−16	−13	−5	6.2
	Tacoma C	279	9	13	15	16	20	8.0
	Tatoosh Island C	110	12	16	18	20	25	18.9
	Walla Walla C	949	−23	−16	−12	−8	−1	5.3
W. Va.	Elkins A	1970	−10	−7	−4	−3	1	6.2
	Parkersburg C	685	−7	−3	−1	1	5	7.2
Wis.	Green Bay C	598	−27	−23	−20	−18	−14	10.5
	La Crosse C	725	−27	−23	−20	−19	−14	9.3
	Madison C	1008	−25	−21	−19	−17	−13	10.1
	Milwaukee C	744	−24	−20	−17	−15	−10	12.1
Wyo.	Cheyenne A	6139	−26	−22	−19	−17	−12	13.3
	Lander A	5569	−39	−33	−30	−27	−21	3.9
	Yellowstone Park C	6239	−45	−38	−34	−30	−23	8.8

Notes for Table 1

[a] Data compiled from U. S. Weather Bureau records by H. C. S. Thom.
[b] Col. 2. The stations followed by letter A are airport stations; all others are city office stations and are followed by the letter C.
[c] Col. 3. Elevations marked c are ground elevations at stations. All other elevations given are the actual elevations of the thermometer bulb above mean sea level.
[d] Wind speeds are from U. S. Weather Bureau records for the period ending February 1948.
[e] The winter design temperatures shown in Columns 4–8 inclusive for U. S. cities are minimum average daily temperatures that will recur with a probable frequency of once in the number of years shown at the head of the column based on the U. S. Weather Bureau records for the 30-year period between 1921 and 1950. For example, the temperatures in Column 6 will recur once every 13 years as minimum average daily values, the temperatures in Column 7, once every 10 years, etc.

Tables 1 or 2, a heating engineer should consider each of the following factors:

1. Is heat capacity of structure high or low?
2. Is structure insulated?
3. Is structure exposed to high wind?
4. Is the ventilation or infiltration load high?
5. Is there more glass area than normal in the structure?
6. During what part of the day will it be used?
7. What is the nature of occupancy?
8. Will there be long periods of operation at reduced indoor temperature?
9. What is the amplitude between maximum and minimum daily temperature in the locality?
10. Are there local conditions which cause significant variation from temperatures reported by Weather Bureau?
11. What auxiliary heating devices will there be in the building?
12. What is the expected cost of fuel or energy?

For the usual structure under typical conditions the ASHAE Technical Advisory Committee on Heating and Air Conditioning Loads recommends that the design temperatures in Column 6 be used for U. S. cities. The U. S. Weather Bureau data show that an average daily temperature lower than that shown in Column 6 will occur with a probability of once in 13 years.

If a careful study of preceding factors 1 to 12 shows that the indoor temperature of the structure in question will respond more quickly than normal to extreme outdoor temperatures that persist for less than twelve hours, then design temperatures from Columns 7 or 8 may be used. On the other hand, if the structure is less responsive than normal to changes in outdoor temperature, outdoor design temperatures from Columns 5 or 4 may be selected.

The designer must keep in mind, before reaching a final

Table 2 . . . . Winter Climatic Conditions[a]—Canada

Col. 1 Prov of Canada	Col. 2 Station[b]	Col. 3 Elevation[c] Ft	Col. 4 Period of Record[d]	Col. 5 Lowest Temp on Record[d] F	Col. 6 Average Annual Min. Temp[e] F	Col. 7 Winter Design Temp[g] –F 1%	Col. 8 2½%	Col. 9 5%	Col. 10 10%	Col. 11 Avg Wind Speed[f] Mph
Alta.	Calgary A	3540	1921–1950	−46	−31	−35	−29	−25	−16	10.1
	Edmonton A	2219	1921–1950	−55	−39	−39	−33	−29	−21	7.6
	Grande Prairie A	2190	1942–1950	−62	−47	−43	−39	−34	−27	7.9
	Lethbridge A	3018	1921–1950	−45	−32	−38	−32	−28	−19	15.0
	McMurray A	1216	1921–1950	−64	−51	−48	−42	−37	−30	
	Medicine Hat A	2365	1921–1950	−49	−34	−41	−35	−31	−22	9.0
B. C.	Estevan Point C	20	1924–1950	7	22	14	17	21	27	9.9
	Fort Nelson A	1230	1938–1950	−61	−47	−42	−38	−33	−28	3.7
	Penticton A	1121	1921–1950	−16	−1	−14	−6	−1	4	
	Prince George A	2218	1921–1950	−58	−42	−43	−32	−25	−16	7.2
	Prince Rupert C	170	1921–1950	−3	12	5	8	12	18	8.0
	Vancouver A	22	1938–1950	0	13	8	11	15	21	7.7
	Victoria C	228	1921–1950	6	20	12	15	19	25	12.3
Man.	Brandon C	1200	1921–1950	−49	−40	−36	−32	−28	−24	
	Churchill A	115	1932–1950	−50	−43	−43	−42	−40	−37	14.7
	The Pas A	894	1921–1950	−54	−42	−43	−39	−30	−26	6.4
	Winnipeg A	786	1921–1950	−44	−35	−33	−29	−25	−21	12.0
N. B.	Campbellton C	42	1938–1950	−31		−14	−11	−8	−3	
	Fredericton C	164	1923–1950	−38	−24	−9	−6	−3	2	9.2
	Moncton A	248	1921–1950	−36	−23	−11	−8	−5	0	14.9
	Saint John C	119	1921–1950	−21	−12	−6	−3	0	5	13.8
Nfld.	Corner Brook C	40	1933–1950	−25	−12	−4	−1	2	6	
	Gander A	482	1937–1950	−16	−10	−6	−3	0	4	17.2
	Goose Bay A	144	1942–1950	−35	−30	−29	−26	−24	−20	10.3
	St. John's A	463	1926–1950	−10	−2	−1	1	4	7	19.3
N.W.T.	Aklavik C	30	1927–1950	−62	−42	−50	−46	−43	−39	
	Fort Norman C	300	1925–1950	−65	−54	−46	−42	−39	−35	
	Frobisher A	68	1942–1950	−49	−44	−51	−47	−43	−39	
	Resolute C	56	1948–1950	−55	−53	−45	−42	−40	−36	9.2
	Yellowknife A	682	1942–1950	−60	−51	−49	−47	−45	−41	
N.S.	Halifax C	83	1921–1950	−21	−6	0	4	7	11	9.6
	Halifax A	136	1944–1950	−11		−3	2	5	9	12.4
	Sydney A	197	1921–1950	−23	−9	−7	1	4	8	13.1
	Yarmouth A	136	1921–1950	−11	1	4	7	10	13	13.5
Ont.	Fort William A	644	1921–1950	−41	−29	−30	−24	−20	−15	9.6
	Hamilton C	303	1939–1950	−17	−6	−4	0	5	10	
	Kapuskasing A	752	1921–1950	−53	−42	−34	−30	−27	−22	10.0
	Kingston C	340	1921–1950	−31	−17	−15	−11	−6	−1	10.8
	Kitchener C	1100	1921–1950	−29	−11	−8	−3	2	7	
	London A	912	1921–1950	−27	−11	−5	−1	3	8	11.9
	North Bay A	1210	1925–1950	−46	−33	−24	−20	−16	−11	11.3
	Ottawa A	339	1921–1950	−38	−26	−18	−15	−11	−7	11.1
	Peterborough C	648	1921–1950	−38	−21	−15	−11	−6	−1	
	Sioux Lookout A	1227	1931–1950	−50	−39	−38	−33	−29	−24	8.5
	Sudbury C	837	1921–1950	−45		−21	−17	−12	−5	
	Timmins C	1100	1921–1950	−55	−34	−30	−26	−22	−17	
	Toronto C	379	1921–1950	−22	−7	−4	0	5	10	14.1
	Toronto A	578	1938–1950	−24	−13	−9	−4	1	6	12.1
	Windsor A	637	1921–1950	−10	−2	0	3	7	11	12.3
P. E. I.	Charlottetown C	74	1921–1950	−23	−11	−6	−3	0	4	11.3
Que.	Arvida C	375	1932–1950	−42	−31	−23	−19	−16	−11	8.2
	Knob Lake A	1605	1948–1950	−59	−48	−44	−40	−34	−30	
	Mont Joli A	150	1943–1950	−28		−14	−11	−8	−3	13.3
	Montreal C	187	1921–1950	−29	−16	−12	−9	−6	−2	12.3
	Montreal A	98	1942–1950	−28	−20	−14	−11	−8	−4	13.1
	Port Harrison C	66	1942–1950	−49	−42	−43	−39	−35	−31	13.4
	Quebec City C	296	1921–1950	−32	−19	−16	−12	−9	−4	12.4
	Seven Islands A	190	1944–1950	−43		−23	−20	−17	−12	
	Sherbrooke C	620	1921–1950	−39	−24	−15	−12	−9	−5	8.2
	Three Rivers C	50	1921–1950	−43	−24	−17	−13	−10	−5	
Sask.	Prince Albert A	1414	1921–1950	−56	−46	−48	−41	−37	−30	4.9
	Regina A	1884	1921–1950	−54	−39	−39	−34	−30	−25	12.1
	Saskatoon A	1645	1921–1950	−54	−41	−45	−37	−33	−26	9.7
	Swift Current A	2677	1921–1950	−54	−34	−39	−33	−29	−24	14.6
Y. T.	Dawson C	1062	1921–1950	−73	−54	−62	−56	−50	−39	
	Whitehorse A	2289	1941–1950	−62	−51	−49	−43	−37	−26	8.7

Notes for Table 2 . . . . Winter Climatic Conditions[a]—Canada

[a] Data compiled from the *Climatological Atlas of Canada* for the years indicated.

[b] Col. 2. The stations followed by letter A are airport stations; all others are city office stations and are followed by letter C.

[c] Col. 3. The elevations listed are ground elevations of the station.

[d] Col. 4. The periods of record indicated apply only to the lowest temperature ever recorded shown in Col. 5, and generally extend from a summer month of the first year indicated through the spring months of the last year indicated.

[e] Average of readings of one lowest temperature obtained for each year for period of record. For Canada, in some cases, more than one location have been used.

[f] Most of the wind speeds are based on periods ending in 1947 which are somewhat shorter than the periods for the minimum temperatures. The three months, December, January, and February were used.

[g] The winter design temperatures shown in columns 7-10 inclusive for Canadian cities are the Fahrenheit temperature values at or below which 1, 2½, 5, and 10 percent, respectively, of the January hourly outdoor temperatures occur for the 10 years from 1941 to 1950. The tabulated values are based on hourly temperature observations for some of the cities and upon the difference between the mean monthly temperature and the design temperature for the remainder of the cities listed.

decision on outdoor design temperature, that if the indoor-outdoor design temperature difference is exceeded, the indoor temperature will fall. The question is to determine how large a drop will occur and whether it can be tolerated.

Finally, there is a factor, perhaps intangible, that should not be ignored. It is the performance expected by the owner from the system. In order to judge whether expected performance can be assured the designer needs a full understanding of the basis on which the capacities of all the system components are derived or determined, the limits of accuracy of published performance data, and the accelerating capability of certain types of equipment. There is no substitute for engineering judgment in problems of this type. Judgment is developed from experience and continued study.

Canadian Winter Design Temperatures

Canadian winter design temperatures are based on temperature frequencies and show the percentage of time that the temperature might be expected to fall below certain values. Since January is normally the coldest month of the winter in most of Canada, and to simplify the analysis and understanding, these values are based on January data only.[a] In using these values to calculate the heat loss of a building, the engineer is thus designing for the coldest month of the year.

Winter design temperature on a 2½ percent basis is the temperature value expressed in degrees Fahrenheit at or below which 2½ percent of the January outdoor temperatures occur. It is suggested that this 2½ percent basis value be normally used in Canada. However, the engineer has the opportunity of selecting the 5 or 10 percent basis value if it is not so important that the pre-selected indoor design temperature be maintained, or on the other hand the 1 percent basis value if cost is not as important as the maintenance of indoor temperatures.

INDOOR TEMPERATURES

The indoor air temperature that must be maintained within a building is understood to be the dry-bulb temperature at the breathing line, 5 ft. above the floor, or at the seating level, 30 in. above the floor, and not less than 3 ft from the outside walls. Indoor air temperatures usually specified, vary in accordance with the intended use of the building. Table 3 presents values that conform to good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 6. In other words, a person may feel warm or cool at the same dry-bulb temperature, depending on the relative humidity and air motion. The optimum winter

Table 3 . . . . Winter Indoor Dry-Bulb Temperatures Usually Specified[a]

Type of Building	F
SCHOOLS—	
Classrooms	72–74
Assembly rooms	68–72
Gymnasiums	55–65
Toilets and baths	70
Wardrobe and locker rooms	65–68
Kitchens	66
Dining and lunch rooms	65–70
Playrooms	60–65
Natatoriums	75
HOSPITALS—	
Private rooms	72–74
Private rooms (surgical)	70–80
Operating rooms	70–95
Wards	72–74
Kitchens and laundries	66
Toilets	68
Bathrooms	70–80
THEATERS—	
Seating space	68–72
Lounge rooms	68–72
Toilets	68
HOTELS—	
Bedrooms and baths	75
Dining rooms	72
Kitchens and laundries	66
Ballrooms	65–68
Toilets and service rooms	68
HOMES	73–75
STORES	65–68
PUBLIC BUILDINGS	72–74
WARM AIR BATHS	120
STEAM BATHS	110
FACTORIES AND MACHINE SHOPS	60–65
FOUNDRIES AND BOILER SHOPS	50–60
PAINT SHOPS	80

[a] The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion. These three factors considered together constitute what is termed the *effective temperature*. (See Chapter 6.) When relative humidity is not controlled separately, optimum dry-bulb temperature for comfort will be slightly higher than shown in Table 3.

effective temperature for sedentary persons, as determined at the ASHAE Research Laboratory, is 67–68 ET.

As explained in Chapter 6 for so-called still air conditions, a relative humidity of approximately 50 percent is required to produce an effective temperature of 68 ET when the dry-bulb temperature is 72.5 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 percent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 percent or less. Consequently, in using the values listed in Table 3, consideration should be given to the actual relative humidity to be maintained, if provision is to be made for humidification. If no humidification is to be provided, the higher temperatures may not even produce comfort on cold days; if humidity is to be maintained at 50 percent, the lower temperatures will apply.

In rooms having large glass areas, when sun is not shining, or in rooms with walls having a high heat transmission coefficient, the lowered surface temperature will cause a feeling of coolness even though the air temperature in the room is at or above the temperatures indicated in the table. In rooms of this character, it is desirable to design for even higher temperatures than those listed, unless a compensating higher temperature surface is installed to offset the low-temperature surfaces.

The indoor temperatures specified in Table 3 may be used for panel heated spaces as well as for spaces heated by warm air, radiators, or convectors. It is true that warm panel surfaces tend to produce a comfortable environment at a lower room-air temperature than when warm panels are not present, but field experience in the United States has indicated that actual reductions in air temperature are slight in operation.

Temperature at Proper Level. In making the actual heat loss computations, however, for the various rooms in a building it is often necessary to modify the temperatures given in Table 3 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level.

Temperature at Ceiling. The air temperature at the ceiling is generally higher than at the breathing level due to stratification of air resulting from the tendency of the warmer or less dense air to rise. An allowance for this fact should be made in calculating ceiling heat losses, particularly in the case of high ceilings. However, the exact allowance to be made may be somewhat difficult to determine as it depends on many factors, including (1) the type of heating system, (2) ceiling height, and (3) the indoor-outdoor temperature differential. The type of heating system is particularly important, as the temperature gradient from floor to breathing level to ceiling may depend to a large extent on whether a direct radiation, unit heater, or warm air system is used, and in the latter case, whether the air is moved mechanically or by gravity. The temperature of the heating medium is also a factor.

It is impracticable to establish rigid rules for determining the temperature difference to use in all cases. However, for residences and structures having ceiling heights under 10 ft, the comparatively small temperature differential between the breathing level and ceiling generally may be neglected without serious error. For higher ceilings, an allowance of approximately 1 percent per foot of height above the breathing level may be made for ceiling heights up to 15 ft and approximately $\frac{1}{16}$ of 1 deg per foot of height above this level. The values in Table 4 are calculated on this basis. For direct radiation and gravity warm air systems, the allowance should be increased from 50 percent to 100 percent over those given in Table 4. These rules should, however, be used with considerable discretion, as they do not apply to some types of heating systems such as those using panel and baseboard radiation, where very low temperature differences between the floor and the ceiling may exist.

Temperature at Floor Level. According to tests at the University of Illinois,[5, 6, 8] the temperature at the floor level ranged from about 2 to 6 deg below that at the breathing level, or somewhat greater than the difference between the breathing level and ceiling temperatures. Tests at the University of Wisconsin[7] indicated a somewhat smaller differential between the floor and breathing level temperatures. As a general rule, if the breathing level to ceiling temperature differential is neglected (as with ceiling heights under 10 ft), the breathing-level-to-floor differential may also be neglected, as the two are somewhat compensating, especially where both floor and ceiling losses are calculated for the same space. In other cases, the 10 ft temperature differentials in Table 4 may be used in arriving at the floor heat loss, these differentials to be subtracted from the breathing level temperature.

ATTIC TEMPERATURES

Frequently, it is necessary to estimate the attic temperature, and in such cases Equation 1 can be used for this purpose:

$$t_a = \frac{A_c U_c t_c + t_o(A_r U_r + A_w U_w + A_g U_g)}{A_c U_c + A_r U_r + A_w U_w + A_g U_g} \qquad (1)$$

where

t_a = attic temperature, Fahrenheit.
t_c = indoor temperature near top floor ceiling, Fahrenheit.
t_o = outdoor temperature, Fahrenheit degrees.
A_c = area of ceiling, square feet.
A_r = area of roof, square feet.
A_w = area of net vertical attic wall surface, square feet.
A_g = area of attic glass, square feet.
U_c = coefficient of transmission of ceiling, based on surface conductance of 2.20 (upper surface, see Chapter 9). 2.20 = reciprocal of one-half the air space resistance.
U_r = coefficient of transmission of roof, based on surface conductance of 2.20 (lower surface, see Chapter 9).
U_w = coefficient of transmission of vertical wall surface.
U_g = coefficient of transmission of glass.

Example 1. Calculate the temperature in an unheated attic, assuming the following conditions: $t_c = 70$; $t_o = 10$; $A_c = 1000$; $A_r = 1200$; $A_w = 100$; $A_g = 10$; $U_c = 0.50$; $U_r = 0.40$; $U_w = 0.30$; $U_g = 1.13$.

Solution: Substituting these values in Equation 1:

$$t_a = \frac{(1000 \times 0.40 \times 70) + 10[(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13)]}{(1000 \times 0.40) + (1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13)}$$

$$t_a = \frac{34{,}413}{1041} = 33.1 \text{ F.}$$

Equation 1 neglects the effect of any interchange of air such as would take place through attic vents or louvers intended to preclude attic condensation. Test data[9, 10] indicate reduction in temperature difference between attic air and weather is linear with attic ventilation rates between 0 and 0.5 cfm per sq ft of ceiling area. A ventilation rate of 0.5 cfm per sq ft reduces this attic-to-weather temperature difference approximately 50 percent, while a ventilation rate of 0.1 cfm per sq ft reduces this temperature difference about 10 percent. When attic ventilation meets the requirements of Table 3 in Chapter 10, 0.5 cfm per sq ft is the approximate ventilation rate under design conditions. Therefore, the attic temperature found from Equation 1 above can be reduced accordingly, depending upon the estimated ventilation rate. However, since this affects the overall heat loss of a residence with an insulated ceiling only one or two percent, it can be neglected without serious error.

Neither does this equation take into consideration such factors as heat exchange between chimney and attic or solar radiation to and from the roof. Because of these latter effects, actual attic temperatures are frequently higher than the calculated values using Equation 1. The attic temperature may be calculated in the usual manner by means of Equation 1, allowing the full value of the roof. The error resulting from this assumption will generally be considerably less than if the roof were neglected (as is sometimes the practice) and the attic temperature assumed to be the same as the outdoor temperature. When relatively large louvers are installed, as is customary in the southern states, the attic temperature is often assumed as the average between the indoor and outdoor temperatures.

For a shorter, approximate method of calculating heat losses through attics, the combined ceiling and roof coefficient may be used, as described in Chapter 9.

Table 4 .... Approximate Temperature Differentials Between Breathing Level and Ceiling, Applicable to Certain Types of Heating Systems[a]

Ceiling Height Ft	Breathing Level Temperature (5 ft Above Floor)									
	60	65	70	72	74	76	78	80	85	90
10	3.0	3.3	3.5	3.6	3.7	3.8	3.9	4.0	4.3	4.5
11	3.6	3.9	4.2	4.3	4.4	4.6	4.7	4.8	5.1	5.4
12	4.2	4.6	4.9	5.0	5.2	5.3	5.5	5.6	6.0	6.3
13	4.8	5.2	5.6	5.8	5.9	6.1	6.2	6.4	6.8	7.2
14	5.4	5.9	6.3	6.5	6.7	6.8	7.0	7.2	7.7	8.1
15	6.0	6.5	7.0	7.2	7.4	7.6	7.8	8.0	8.5	9.0
16	6.1	6.6	7.1	7.3	7.5	7.7	7.9	8.1	8.6	9.1
17	6.2	6.7	7.2	7.4	7.6	7.8	8.0	8.2	8.7	9.2
18	6.3	6.8	7.3	7.5	7.7	7.9	8.1	8.3	8.8	9.3
19	6.4	6.9	7.4	7.6	7.8	8.0	8.2	8.4	8.9	9.4
20	6.5	7.0	7.5	7.7	7.9	8.1	8.3	8.5	9.0	9.5
25	7.0	7.5	8.0	8.2	8.4	8.6	8.8	9.0	9.5	10.0
30	7.5	8.0	8.5	8.7	8.9	9.1	9.3	9.5	10.0	10.5
35	8.0	8.5	9.0	9.2	9.4	9.6	9.8	10.0	10.5	11.0
40	8.5	9.0	9.5	9.7	9.9	10.1	10.3	10.5	11.0	11.5
45	9.0	9.5	10.0	10.2	10.4	10.6	10.8	11.0	11.5	12.0
50	9.5	10.0	10.5	10.7	10.9	11.1	11.3	11.5	12.0	12.5

[a] The figures in this table are based on an increase of 1 percent per foot of height above the breathing level (5 ft) up to 15 ft and $\frac{1}{16}$ of one degree for each foot above 15 ft. This table is generally applicable to forced air types of heating systems. For direct radiation or gravity warm air, increase values 50 percent to 100 percent.

TEMPERATURES IN UNHEATED SPACES

The heat loss from heated rooms into unheated rooms or spaces must be based on the estimated or assumed temperature in such unheated spaces. This temperature will lie in the range between the indoor and outdoor temperatures, depending on the relative areas of the surfaces adjacent to the heated room and those exposed to the outside. If the respective surface areas adjacent to the heated room and exposed to the outdoors are approximately the same, and if the coefficients of transmission are approximately equal, the temperature in the unheated space may be assumed to be the mean of the indoor and outdoor design temperatures. If, however, the surface areas and coefficients are unequal, the temperature in the unheated space should be estimated by means of Equation 2.

$$t_u = \frac{t_i(A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.}) + t_o(A_aU_a + A_bU_b + A_cU_c + \text{etc.})}{A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.} + A_aU_a + A_bU_b + A_cU_c + \text{etc.}} \qquad (2)$$

where

t_u = temperature in unheated space, Fahrenheit.
t_i = indoor design temperature of heated room, Fahrenheit.
t_o = outdoor design temperature, Fahrenheit.
A_1, A_2, A_3, etc. = areas of surface of unheated space adjacent to heated space, square feet.
A_a, A_b, A_c, etc. = areas of surface of unheated space exposed to outdoors, square feet.
U_1, U_2, U_3, etc. = coefficients of transmission of surfaces of A_1, A_2, A_3, etc.
U_a, U_b, U_c, etc. = coefficients of transmission of surfaces A_a, A_b, A_c, etc.

Example 2: Calculate the temperature in an unheated space adjacent to a heated room having surface areas (A_1, A_2, and A_3) in contact therewith of 100, 120, and 140 sq ft and coefficients (U_1, U_2, and U_3) of 0.15, 0.20, and 0.25, respectively. The surface areas of the unheated space exposed to the outdoors (A_a and A_b) are respectively 100 and 140 sq ft, and the corresponding coefficients are 0.10 and 0.30. The sixth surface is on the ground and is neglected in this example. Assume $t_i = 70$ and $t_o = -10$.

Solution: Substituting in Equation 2:

$$t_u = \frac{70[(100 \times 0.15) + (120 \times 0.20) + (140 \times 0.25)] + -10[(100 \times 0.10) + (140 \times 0.30)]}{(100 \times 0.15) + (120 \times 0.20) + (140 \times 0.25) + (100 \times 0.10) + (140 \times 0.30)}$$

$$t_u = \frac{4660}{126} = 37 \text{ F.}$$

The temperatures in unheated spaces having large glass areas and having two or more surfaces exposed to the outdoors (such as sleeping porches and sun parlors), generally are assumed to be the same as outdoors.

GROUND TEMPERATURES

Ground temperatures to be assumed for estimating basement heat losses usually will differ in the case of basement walls and floors, the temperatures under the floors generally being higher that those adjacent to walls. Factors affecting these temperatures will be discussed.

Temperatures Under Basement Floors

The temperature of the ground under basement floors[11] is affected by heat sources within the basement and is not influenced by atmospheric conditions. In computing losses

Table 5 .... Below Grade Heat Losses for Basement Walls and Floors

Ground Water Temperature[a]	Basement Floor Loss[b] Btu/Sq Ft	Below Grade Wall Loss[b] Btu/Sq Ft
40	3.0	6.0
50	2.0	4.0
60	1.0	2.0

[a] See Fig. 4, Chapter 41.
[b] Based on basement temperature of 70 F and U of 0.10

through basement floors, the ground temperatures may be assumed to be the same as water temperatures at depths of 30 to 60 ft given in Fig. 4, Chapter 41. Test observations indicate that heat losses through basement floors frequently are over-estimated.[12]

Temperatures Adjacent to Basement Walls

Ground temperatures near the surface and under open spaces vary with the climate, the season of the year, and the depth below the surface. The nearer the surface (during the cold weather) the lower will be the ground temperature. Frost will penetrate to a depth of over 4 ft in some localities if not protected by snow. A thick blanket of snow will result in a higher ground temperature near the surface. Therefore, in localities where the ground is covered with a heavy blanket of snow throughout the winter, the ground temperatures near the surface will be higher than when little or no snow is present.

Complete data on ground temperatures adjacent to buildings are not available, but since the recommended transmission coefficient for basement walls in contact with the soil is only 0.10, any reasonable, assumed ground temperature will not materially affect the calculated heat loss.

BASEMENT TEMPERATURES AND HEAT LOSS

The allowance to be made for basement heat loss depends on whether the basement is to be heated or not.

If the basement is *heated* to a specified temperature, the heat loss should be calculated in the usual manner, based on the proper wall and floor coefficients (see Chapter 9) and the outdoor air and ground temperatures. Heat loss through windows and walls above grade should be based on outdoor temperatures and the proper air-to-air coefficients. Heat loss through basement walls below grade should be based on the floor and wall coefficients for surfaces in contact with the soil, and on the proper ground temperature.

The heat loss values for below-grade basement walls and floors given in Table 5 are sufficiently precise for general practice.

If a basement is completely below grade and is *not heated*, the temperature in the basement normally will range between that in the rooms above and the ground temperature. Basement windows will, of course, lower the basement temperature when it is cold outdoors, and heat given off by the heating plant will increase the basement temperature. In any case, the exact basement temperature is indeterminate if the basement is not heated. In general, it is found that the transient heat from the heating plant warms the air near the basement ceiling sufficiently to make it unnecessary to make an allowance for floor heat loss from rooms located over the basement.

The temperature in crawl spaces below floors will vary widely depending on the number and size of wall vents, the amount of warm piping present and type of piping insulation. It is necessary, therefore, to evaluate the conditions and to select an appropriate temperature by judgment.

HEAT LOSSES FROM FLOOR SLABS

Two types of concrete floors used in basementless houses are (*a*) the unheated floor, relying for warmth on heat delivered above floor level by the heating system, and (*b*) the heated floor containing heated pipes or ducts that constitute a radiant slab or portion thereof for complete or partial heating of the house.

For type (*a*) the floor heat loss, economically considered, is of minor importance since it comprises generally about 10 percent of the total heat loss of the house. From the comfort standpoint, however, it may be most important, since houses with cold floors are not successfully heated. In this connection, it should be remembered that a well insulated floor does not assure comfort if downdrafts from windows or exposed walls create pools of chilly air over considerable areas of the floor. For this reason a floor of type (*a*) should not be used in a severe climate except with a heating system that delivers enough heat near the floor to counteract the downdrafts of the exterior walls and the heat transmission through the floor.

The results of some experiments[12, 13] with type (*a*) unheated floor slabs indicate that the heat loss from a concrete slab floor on grade is more nearly proportional to the perimeter than to the area of the floor, and that the heat loss can be estimated by means of the equation:

$$H_F = FP\,(t_i - t_o) \qquad (3)$$

where

H_F = heat loss of the floor, Btu per hour.
P = perimeter or exposed edge of the floor, linear feet.

Table 6 .... Heat Loss of Concrete Floors at or Near Grade Level per Foot of Exposed Edge

Outdoor Design Temperature F	Heat Loss per Foot of Exposed Edge, Btu/hr: Recommended 2-in. Edge Insulation	Heat Loss per Foot of Exposed Edge, Btu/hr: 1-in. Edge Insulation
−20 to −30	50	55
−10 to −20	45	50
0 to −10	40	45
Outdoor Design Temperature F	**1-in. Edge Insulation**	**No Edge Insulation[a]**
−20 to −30	60	75
−10 to −20	55	65
0 to −10	50	60

[a] This construction not recommended; shown for comparison only.

F = heat loss coefficient, Btu per (hour) (linear foot of exposed edge) (degree difference in temperature between the indoor air and the outdoor air). (F ranges between 0.81 for a floor with no edge insulation to 0.55 for a floor with edge insulation.)
t_i = indoor air temperature, Fahrenheit.
t_o = outdoor air temperature, Fahrenheit.

In most instances the values given in Table 6 for edge loss are of sufficient precision for use.[14] The insulation shown extending under the floor horizontally for 2 ft can also be located along the foundation wall with equal effectiveness if the insulation extends 24 in. below the floor level.

Example 3: Calculate the heat loss from the floor of a 12 ft x 15 ft room with two exposures. The floor is an unheated concrete slab which is insulated at the edge with 2 in. of insulation extending horizontally for a distance of 2 ft from the edge, and the house is located in an area with an outdoor design temperature of −15 F.

Solution: From Table 6 the heat loss per foot of exposed edge is 45 Btu per hr. The length of exposed edge is 12 ft + 15 ft = 27 ft, and the total edge loss 27 x 45 = 1215 Btu per hr.

Floors of type (*b*) containing heating pipes or ducts, are now in common use. The heat loss downward into the ground and outward through the edges of the floor slab is called the reverse loss.

The results[15] of an investigation in which a warm air perimeter duct was embedded in four types of concrete floor-slab and foundation constructions has verified the indication that Equation 3 can be used to calculate the reverse loss when warm air perimeter heating ducts are used. To make the results of this application more usable, values corresponding to those shown in Table 6 for unheated floors are given in Table 7 for concrete floors with a warm air perimeter duct.

The desirability of edge insulation is apparent. One inch of water-resistant material is the minimum thickness of edge insulation that should be used, but a 2-in. thickness is recommended.[14, 16, 17] The values of edge loss in Table 7 indicate that the reverse heat loss of heated slabs is likely to be about 20 percent of the total heat loss of many types of present day houses, and may exceed 20 percent if only one inch of insulation is used at the edge of the floor.

The concrete floor slab is usually placed on a gravel fill 4 in. thick or more, both to insulate the floor from the earth and to retard the rise of ground water by capillarity. A waterproof membrane should be installed over the gravel fill. Obviously, it is important that such floors be laid several inches above grade, and that effective subsoil drainage be provided to avoid slabs soaked by rain or melting snow, and consequent excessive heat loss.

TRANSMISSION HEAT LOSS

The basic formula for the loss of heat by transmission through any surface is given in Equation 4:

$$H_t = AU\,(t_i - t_o) \qquad (4)$$

where

H_t = heat loss transmitted through the wall, roof, ceiling, floor, or glass, Btu per hour.
A = area of wall, glass, roof, ceiling, floor, or other exposed surface, square feet.
U = coefficient of transmission, air-to-air, Btu per (hour) (square foot) (Fahrenheit degree temperature difference).

Table 7 .... Floor Heat Loss to be Used When Warm Air Perimeter Heating Ducts Are Embedded in Slabs[a]
Btu per (hour) (linear foot of heated edge)

Outdoor Design Temperature, F	Edge Insulation: 1-in. Vertical Extending Down 18 in. Below Floor Surface	1-in. L-Type Extending at Least 12 in. Deep and 12 in. Under	2-in. L-Type Extending at Least 12 in. Down and 12 in. Under
−20	105	100	85
−10	95	90	75
0	85	80	65
10	75	70	55
20	62	57	45

[a] Factors include loss downward through inner area of slab.

t_i = indoor temperature near surface involved (this may not necessarily be the so-called breathing-line temperature), Fahrenheit degrees.
t_o = outdoor temperature, or temperature of adjacent unheated space or of the ground, Fahrenheit degrees.

Example 4: Calculate the transmission loss through an 8-in. brick wall having an area of 150 sq ft, if the indoor temperature t_i is 70 F and the outdoor temperature t_o is −10 F.

Solution: The coefficient of transmission U of a plain 8-in. brick wall is 0.41 (Chapter 9, Table 6). The area A is 150 sq ft. Substituting in Equation 4:

$$H_t = 150 \times 0.41 \times [70 - (-10)] = 4{,}920 \text{ Btu per hour.}$$

Transmission Loss Through Ceilings and Roofs

The transmission heat loss through top floor ceilings, attics, and roofs may be estimated by either of two methods:

1. By substituting in Equation 4 the ceiling area A, the indoor-outdoor temperature difference $(t_i - t_o)$ and the proper value of U:
   a. *Flat roofs.* Select the coefficient of transmission of the ceiling and roof from Tables 11 to 14, Chapter 9, or use appropriate coefficients in Equation 1 if side walls extend appreciably above the ceiling of the floor below.
   b. *Pitched roofs.* Select the combined roof and ceiling coefficient from Table 15, Chapter 9 or calculate the combined roof and ceiling coefficient by means of Equations 4 and 5, Chapter 9, where these formulas are applicable as explained in Chapter 9.
2. By estimating the attic temperature (based on the indoor and outdoor design temperatures) by means of Equation 1, and substituting for t_o in Equation 4, the value of t_o thus obtained, together with the ceiling area A and the ceiling coefficient U. This applies to *pitched roofs.* In the case of *flat roofs* it is not necessary to calculate the attic temperatures, as the ceiling-roof heat loss can be determined as suggested in preceding paragraph 1a.

INFILTRATION HEAT LOSS

The infiltration heat loss includes (1) the *sensible heat loss* or the heat required to warm the outdoor air entering by infiltration, and (2) the *latent heat loss* or the heat equivalent of any moisture which must be added.

Sensible Heat Loss

The formula for the heat required to warm the outdoor air which enters a room by infiltration to the temperature

of the room, is given in Equation 5:

$$H_s = 0.240\ Q_\rho\ (t_i - t_o) \qquad (5)$$

where

H_s = heat required to raise temperature of air leaking into building from t_o to t_i, Btu per hour.
0.240 = specific heat of air.
Q = volume of outdoor air entering building, cubic feet per hour (see Chapter 11).
ρ = density of air at temperature t_o, pounds per cubic foot.

It is sufficiently accurate to use $\rho = 0.075$ in which case Equation 5 reduces to

$$H_s = 0.018\ Q\ (t_i - t_o) \qquad (5a)$$

The volume Q of outdoor air entering per hour depends on the wind speed and direction, the width of crack or size of openings, the type of openings, and other factors, as explained in Chapter 11. Where the crack method is used for estimating leakage, it is more convenient to express the air leakage heat loss in terms of the crack length:

$$H_s = B\ L\ (t_i - t_o) \qquad (5b)$$

where

B = air leakage per (hour) (foot of crack) (Chapter 11) for the wind velocity and type of windows or door crack involved, multiplied by 0.018.
L = length of window or door crack to be taken into consideration, feet.

Example 5: What is the infiltration heat loss per hour through the crack of a 3 x 5 ft average, double-hung, non-weather-stripped, wood window, based on a wind speed of 15 mph? Assume indoor and outdoor temperatures to be 70 F and zero, respectively.

Solution: According to Table 2, Chapter 11, the air leakage through a window of this type (based on 1/16 in. crack and 3/64 in. clearance) is 39 cu ft per (ft of crack) (hour). Therefore, $B = 39 \times 0.018 = 0.70$. The length of crack L is $(2 \times 5) + (3 \times 3)$, or 19 ft; $t_i = 70$ and $t_o = 0$. Substituting in Equation 5b,

$$H_s = 0.70 \times 19 \times (70 - 0) = 931 \text{ Btu per hour.}$$

Crack Length to be Used for Computations

For designers who prefer to use the crack method, the basis of calculation is as follows: The amount of crack used for computing the infiltration heat loss should be not less that half of the total length of crack in the outside walls of the room. For a building having no partitions, air entering through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building. In a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack.

In small residences the total infiltration loss of the house is generally considered to be equal to the sum of the infiltration losses of the various rooms. However, this is not necessarily accurate as at any given time infiltration will take place only on the windward side or sides and not on the leeward side. Therefore, for determining the total heat requirements of larger buildings it is more accurate to base the total infiltration loss on the wall having the most total crack, but in no case on less than half of the total crack in the building.

Number of Air Changes to be Used for Computations

Since a certain amount of judgment is required regarding quality of construction, weather conditions, use of room, and other factors in estimating infiltration by any method, some designers base infiltration upon an estimated number of air changes rather than upon the length of window cracks. Table 4 of Chapter 11 indicates air changes commonly used, but should be taken only as a guide.

When calculating infiltration losses by the air change method, Equation 5a may be used by substituting for Q the volume of the room multiplied by the number of air changes obtained from Table 4, Chapter 11. For further discussion of the method see section on Air Change Method in Chapter 11.

Latent Heat Loss

When it is intended to add moisture to air leaking into a room in order to maintain proper winter comfort conditions, it is necessary to determine the heat required to evaporate the water vapor added. This heat may be calculated by the equation

$$H_l = Q_\rho\ (W_i - W_o)\ h_{fg} \qquad (6)$$

where

H_l = heat required to increase moisture content of air leaking into building from W_o to W_i, Btu per hour.
Q = volume of outdoor air entering building, cubic feet per hour.
ρ = density of air at temperature t_i, pounds per cubic foot.
W_i = humidity ratio of indoor air, pounds per pound of dry air.
W_o = humidity ratio of outdoor air, pounds per pound of dry air.
h_{fg} = latent heat of vapor at W_i, Btu per pound.

If the latent heat of vapor h_{fg} is assumed to be 1060 Btu per lb, Equation 6 reduces to:

$$H_l = 79.5\ Q\ (W_i - W_o) \qquad (6a)$$

Equations 5a, 5b, and 6a may also be used for determining the sensible and latent heat gains due to infiltration in cooling load computations.

SELECTION OF WIND SPEEDS

The effect of wind on the heating requirements of any building should be given consideration for two reasons:

1. Wind movement increases the heat transmission of walls, glass, and roof, affecting poor walls to a much greater extent than good walls.

2. Wind increases materially the infiltration of cold air through the cracks around doors and windows, and even through the building materials themselves (see Tables 1 and 2, Chapter 11).

Theoretically, as a basis for design, the most unfavorable combination of temperature and wind speed should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outdoor temperature, than on a quiet day with a much lower outdoor temperature. However, the combination of wind and temperature, which is the worst, would differ with different buildings, because wind speed has a greater effect on buildings which have relatively high infiltration losses. It would be possible to compute the heating load for a building for several different combinations of temperature and wind speed which records show to have occurred, and to select the worst combination, but designers generally do not feel that such a degree of refinement is justified.

A complete correlation between the design temperatures in Column 6 of Table 1, and the simultaneous maximum wind speed has not been made at the present time. If a designer prefers the air change method for computing infiltration load, such correlation is not too important. Designers who use the crack method may use a leakage rate at 15 mph wind speed unless local experience has established that other speeds are more appropriate. Abnormally high wind speed may have significant effect on the transmission coefficient, U. (See Table 20, Chapter 9.)

Exposure Factors

Many designers use empirical *exposure* factors to increase the calculated heat loss of rooms or spaces on the side or sides of the building exposed to the prevailing winds. However, the use of exposure factors is unnecessary when THE GUIDE method of calculating heat losses is used. Therefore, exposure factors may be regarded as *factors of safety* for the rooms or spaces exposed to the prevailing winds, to allow for additional capacity for these rooms or spaces, or to *balance the radiation*, particularly in the case of multi-story buildings. Tall buildings may have severe infiltration heat losses, induced by their stack effect (see Chapter 11), that will require special consideration. Although the exposure allowance frequently is assumed to be 15 percent, the actual allowance to be made, if any, must to a large extent be a matter of experience and judgment of the designer, since there are at present no authentic test data available from which rules could be developed for the many conditions encountered in practice.

AUXILIARY HEAT SOURCES

The heat supplied by persons, lights, motors, and machinery always should be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only after careful consideration of all local conditions. In many cases, these heat sources should not affect the size of the heating plant at all, although they may have a marked effect on the operation and control of the system. In general, where audiences are present, the heating system must have sufficient capacity to bring the building to the stipulated indoor temperature before the audience arrives. In industrial plants, quite a different condition exists, and heat sources, if always available during occupancy, may be substituted for a portion of the heating requirements. In no case should the actual heating installation (exclusive of heat sources) be reduced below that required to maintain at least 40 F in the building.

Electric Motors and Machinery

Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat. This heat is retained in the room if the product manufactured is not removed until its temperature is the same as the room temperature.

If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used. In some mills this is the chief source of heating, and it is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round. Table 8 shows the heat output equivalent of various sources of heat in a factory. For information concerning the heat supplied by persons, refer to data given in Chapter 6, and also Table 27, Chapter 13. For appliances see Table 28, Chapter 13.

INTERMITTENTLY HEATED BUILDINGS

In the case of intermittently heated buildings additional heat is required for raising the temperature of the air, the building materials, and the material contents of the building to the specified indoor temperature. The rate at which this additional heat must be supplied depends upon the heat capacity of the structure and its material contents, and upon the time in which these are to be heated.[18]

This additional heat may be computed and allowed for as conditions require, but inasmuch as the heating system proportioned for taking care of the heat losses will usually have a capacity about 100 percent greater than that required for average winter weather, and inasmuch as most buildings may either be continuously heated or have more time allowed for heating up during the few minimum temperature days, no allowance usually is made, except in the size of boilers or furnaces. For churches, auditoriums, and other intermittently heated buildings, additional capacity should be provided.

Table 8 .... Heat Equivalents of Various Sources[a]

Machinery (Motor in room)	= Motor Hp/efficiency x 2544 Btuh
Machinery (Motor outside room)	= Motor Hp x 2544 Btuh
Electric Lights	= Kilowatts x 3413 Btuh
Gas (Producer = 150) (Manufactured = 535)	(Natural = 1000) Btu/cu ft

[a] Additional values are given in Chapter 13, Table 28.

RESIDENCE HEAT LOSS PROBLEMS

The following *Examples 6* and *7* will illustrate the procedure for calculating the heat loss of a residence, uninsulated and insulated, in accordance with the recommendations given in this chapter.

Example 6: Calculate the heat loss of the residence shown in Fig. 1 located in the vicinity of Chicago. From Table 1, design outdoor conditions are − 10 F and 12 mph wind speed. Indoor temperature from Table 3 is assumed to be 70 F. The attic is unheated. Assume ground temperature to be 50 F (see Fig. 4, Chapter 41) under basement and garage floors and 32 F adjoining basement walls. Estimate infiltration losses by the air change method. No wall, ceiling, or roof insulation is to be considered in this problem, but all first and second floor windows, except in the garage, are to have storm sash. The building is constructed as follows (heat transmission coefficients U are in parentheses):

Walls: Brick veneer, building paper, wood sheathing, studding, metal lath, and plaster (0.29). Walls of dormer over garage, same except wood siding in place of brick veneer (0.26).

Attic Walls: Brick veneer, building paper, wood sheathing on studding (0.42).

Basement Walls: 10 in. concrete (0.10).

Roof: Asphalt shingles on wood sheathing on rafters (0.44).

Ceiling (Second Floor): Metal lath and plaster (0.74).

Windows: Double-hung wood windows averaging 70 percent glass (0.45; from Chapter 9, Table 18, Section D, the U value for wood windows with storm sash is 0.53 × application factor; by interpolation this factor is 0.85). Steel casement sash in ga-

Table 9 . . . . Heat Loss Calculation Sheet for Uninsulated Residence (Fig. 1)

A Room or Space	B Part of Structure or Infiltration Air Changes	C Net Area or Air Volume	D Coefficient	E Temp. Diff.[a]	F Heat Loss (Btu per hour)	G Totals (Btu per hour)
Bedroom A and Closet	Walls	238 sq ft	0.29	80	5520	
	Glass	40 sq ft	0.45	80	1440	
	Ceiling	252 sq ft	0.74	39.8[d]	7410	
	Infiltration (¾)[e]	1510 cfh[b]	0.018[c]	80	2180	16,550
Bedroom B and Closet	Walls	156 sq ft	0.29	80	3620	
	Glass	40 sq ft	0.45	80	1440	
	Ceiling	170 sq ft	0.74	39.8[d]	5000	
	Infiltration (¾)[g]	1020 cfh[b]	0.018[c]	80	1470	11,530
Bedroom C and Closet	Walls	114 sq ft	0.29	80	2650	
	Glass	27 sq ft	0.45	80	970	
	Ceiling	129 sq ft	0.74	39.8[d]	3800	
	Infiltration (¾)[g]	874 cfh[b]	0.018[c]	80	1260	8,680
Bedroom D and Closet	Walls	118 sq ft	0.29	80	2740	
	Glass	20 sq ft	0.45	80	720	
	Ceiling	110 sq ft	0.74	39.8[d]	3240	
	Floor over garage	110 sq ft	0.26	35[e]	960[p]	
	Infiltration (¾)[g]	660 cfh[b]	0.018[c]	80	950	8,610
Bathroom 1	Walls	30 sq ft	0.29	80	690	
	Glass	14 sq ft	0.45	80	500	
	Ceiling	55 sq ft	0.74	39.8[d]	1620	
	Infiltration (1)[g]	440 cfh[b]	0.018[c]	80	630	3,440
Bathroom 2	Walls	79 sq ft	0.26	80	1640	
	Glass	9 sq ft	0.45	80	320	
	Ceiling	35 sq ft	0.74	39.8[d]	1060	
	Floor over garage	35 sq ft	0.26	35[e]	320[p]	
	Infiltration (1)[g]	280 cfh[b]	0.018[c]	80	400	3,740
Living Room	Walls	267 sq ft	0.29	80	6200	
	Walls (adjoining garage)	94 sq ft	0.39[f]	35[e]	1280[p]	
	Glass	50 sq ft	0.45	80	1800	
	Floor	294 sq ft				
	Infiltration (1½)[h]	3745 cfh[b]	0.018[c]	80	5400	14,680
Dining Room	Walls	166 sq ft	0.29	80	3850	
	Glass (doors)	35 sq ft	0.85	80	2380	
	Glass (windows)	20 sq ft	0.45	80	720	
	Floor	168 sq ft				
	Infiltration (1½)[i]	2140 cfh[b]	0.018[c]	80	3080	10,030
Kitchen and Entrance to Garage	Walls	96 sq ft	0.29	80	2220	
	Walls (adjoining garage)	51 sq ft	0.39[f]	35[e]	700[p]	
	Glass	18 sq ft	0.45	80	650	
	Door	17 sq ft	0.51	35	300	
	Floor	125 sq ft				
	Infiltration (1½)[j]	1595 cfh[b]	0.018[c]	80	2300	6,170
Lavette and Vestibule	Walls	82 sq ft	0.29	80	1900	
	Walls (adjoining garage)	85 sq ft	0.39[f]	35[e]	1160[p]	
	Glass	9 sq ft	0.45	80	320	
	Door	19 sq ft	0.51	80	780	
	Floor	30 sq ft				
	Infiltration (1½)[k]	383 cfh[b]	0.018[c]	80	550	4,710
Entrance Hall	Walls	39 sq ft	0.29	80	900	
	Door	21 sq ft	0.38	80	640	
	Ceiling[r]	87 sq ft	0.74	39.8[d]	2560	
	Infiltration (2)[l]	1110 cfh[b]	0.018[c]	80	1600	5,700
Garage	Walls	167 sq ft	0.29	45[e]	2180	
	Glass	53 sq ft	1.13	45	2700	
	Doors	44 sq ft	0.51	45	1010	
	Infiltration (1½)[m]	2360 cfh[b]	0.018[c]	45	1910	
	Floor	29 ft[s]	0.81	45	1060	
	Gain adjoining rooms				−4420[p]	4,450
Recreation Room[q]	Walls	220 sq ft	0.10	38	840	
	Glass	8 sq ft	1.13	80	720	
	Floor	287 sq ft	0.10	20	570	
	Infiltration (1)[n]	2010 cfh[b]	0.018[c]	80	2890	5,020
					TOTAL	103,310

Notes for Table 9... Heat Loss Calculation for Uninsulated Residence (Fig. 1)

[a] The indoor-outdoor temperature difference is 70− (−10) or 80 F except where otherwise noted.
[b] Volume of infiltration, cfh = (no. air changes) x (floor or ceiling area) x (ceiling height).
[c] From Equation 5a.
[d] The ceiling heat losses are calculated by estimating the attic temperature and then calculating the loss through the ceiling using the proper temperature difference. This unheated attic is not ventilated during winter months. The attic temperature is estimated from Equation 1 to be 30.2 F when the outdoor temperature is −10 F and room temperature is 70 F. The temperature difference is then 70—30.2 or 39.8 deg. For the insulated residence, attic temperature becomes 4.6 F and temperature difference 70—4.6 = 65.4 deg.
[e] Temperature in garage assumed to be 35 F.
[f] Coefficient for wall adjoining garage calculated on basis of metal lath and plaster on both sides of studs (U = 0.39).
[g] One-half of value from Table 4, Chapter 11, for storm windows or weatherstripping.
[h] Exposed on two sides, weatherstripped windows offset by fireplace. Use 1½.
[i] Window on one side weatherstripped but double-doors are hard to close tightly. Hence, conservative value of 1½.
[j] Assuming kitchen vent, door to vestibule usually open, allow full table value of 1½.
[k] One-half value in Table 4, Chapter 11, increased to 1½ by nearby outside door in vestibule.
[l] Full value in Table 4, Chapter 11, to allow for frequent opening of outside door.
[m] Two sides exposed, large doors but large volume. Use value 1½ as given in Table 4, Chapter 11.
[n] Two small unweatherstripped windows in protected location, but fireplace, indicate 1 change.
[p] Heat losses from these rooms into garage are heat gains for garage.
[q] Neglect heat loss to basement, as losses from boiler, piping, etc., will probably keep basement near, if not above, 70 F.
[r] Upstairs hall ceiling figures with downstairs. Heat should be provided downstairs for both.
[s] Linear feet of exposed edge.

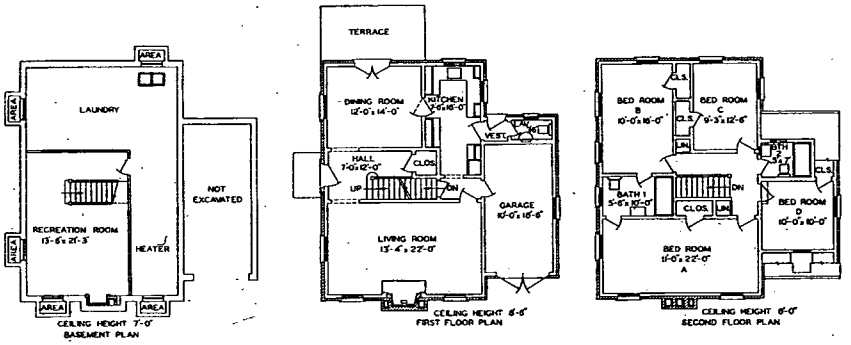


Fig. 1 . . . . Floor Plans of Residence

rage and basement (1.13; from Chapter 9, Table 18, U is 1.13 for all glass and the application factor is 1.00). French doors in dining room 50 percent glass, no storm doors (0.85; from Chapter 9, Table 18, U is 1.13 for all glass; by interpolation the application factor is 0.75).

Floor (Bedroom D): Maple finish flooring on yellow pine subflooring; metal lath and plaster ceiling below (0.26).

Floor (Basement and Garage): 4 in. stone concrete on 3 in. cinder concrete (0.10).

Solution: The calculations for this problem are given in Table 9, and a summary of the results in Table 10. The values in column F of Table 9 were obtained by multiplying together the figures in columns C, D, and E. The heat losses are calculated to the nearest 10 Btu. See reference notes for Table 8 for further explanation of data.

Attention is called to the summary of heat losses (Table 10) for the uninsulated residence. As storm windows are used in this instance the *glass* and *door* transmission heat losses of 19.8 percent are relatively small. The infiltration losses of 14.0 percent are also comparatively small because the storm windows are equivalent to weatherstripping. In this problem, the wall, ceiling, and floor transmission losses comprise 66.2 percent of the total.

Example 7: Calculate the heat loss of residence shown in Fig. 1 based on the same conditions as in *Example 6* but having construction improved or insulated to obtain coefficients as follows:

Walls, 0.13; *Walls of Dormer over Garage*, 0.12; *Attic Walls*, 0.28; *Walls Adjoining Garage*, 0.18; *Basement Walls (Recreation Room)*, 0.10.

Roof, 0.53.

Ceiling (Second Floor), 0.15.

Windows (Same as in *Example 6*).

Floor (Bedroom D), 0.18.

Solution: The procedure for calculating the heat losses is similar to that for *Example 6*. A summary of the results is given in Table 11.

Table 10 .... Summary of Heat Losses of Uninsulated Residence (Btu Per Hour)

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	5520	7410	...	1440	2180	16,550
Bedroom B	3620	5000	...	1440	1470	11,530
Bedroom C	2650	3800	...	970	1260	8,680
Bedroom D	2740	3240	960	720	950	8,610
Bathroom 1	690	1620	...	500	630	3,440
Bathroom 2	1640	1060	320	320	400	3,740
Living Room	7480	...	...	1800	5400	14,680
Dining Room	3850	...	...	3100	3080	10,030
Kitchen	2920	...	...	950	2300	6,170
Lavette	3060	...	...	1100	550	4,710
Entrance Hall	900	2560	...	640	1600	5,700
Garage	−960[a]	−1280[b]	1060	3710	1910	4,440
Recreation	840	...	570	720	2890	5,020
Design Totals	34,950	23,410	2,910	17,410	24,620	103,300
Operating Totals[c]	34,950	23,410	2,910	17,410	12,310	90,990
Percentages[d]	38.4	25.8	3.2	19.1	13.5	100.0

[a] Wall heat loss of 2180 Btuh minus wall heat gains of 1230, 700, and 1160 Btuh. [b] Heat gains of 960 and 320 Btuh. [c] Based on ½ computed infiltration. [d] Based on operating totals.

Table 11 .... Summary of Heat Losses of Insulated Residence (Btu Per Hour)

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	2480	2460	...	1440	2180	8,560
Bedroom B	1620	1660	...	1440	1470	6,190
Bedroom C	1190	1260	...	970	1260	4,680
Bedroom D	1230	1080	690	720	950	4,670
Bathroom 1	310	540	...	500	630	1,980
Bathroom 2	760	250	220	320	400	1,950
Living Room	3370	...	...	1800	5400	10,570
Dining Room	1730	...	...	3100	3080	7,910
Kitchen	1320	...	...	950	2300	4,570
Lavette	1390	...	...	1100	550	3,040
Entrance Hall	410	850	...	640	1600	3,500
Garage	−470[a]	−910[b]	1060	3710	1910	5,300
Recreation	840	...	570	720	2890	5,020
Design Totals	16,180	7,190	2,540	17,410	24,620	67,940
Operating Totals[c]	16,180	7,190	2,540	17,410	12,310	55,630
Percentages[d]	29.1	12.9	4.6	31.3	22.1	100.0

[a] Wall heat loss of 980 Btuh minus wall heat gains of 590, 320, and 540 Btuh. [b] Heat gains 690 and 220 Btuh. [c] Based on ½ computed infiltration. [d] Based on operating totals.

REFERENCES

[1] H. C. S. Thom: Revised winter outside design temperatures (ASHAE Transactions, Vol. 63, 1957, p. 111).

[2] M. K. Thomas: A method for determining winter design temperatures (ASHAE Transactions, Vol. 61, 1955, p. 387).

[3] A. P. Krats and S. Konzo: *Investigation of Oil-Fired Forced Air Furnace Systems in the Research Residence* (University of Illinois, *Engineering Experiment Station Bulletin* No. 318).

[4] A. P. Krats, W. S. Harris, M. K. Fahnestock, and R. J. Martin: *Performance of a Hot-Water Heating System in the I-B-R Research Home at the University of Illinois* (University of Illinois, *Engineering Experiment Station Bulletin* No. 349).

[5] A. P. Krats and W. S. Harris: *A Study of Radiant Baseboard Heating in the I-B-R Research Home* (University of Illinois, *Engineering Experiment Station Bulletin* No. 358).

[6] W. S. Harris: *Performance of a One-Pipe Steam System in the I-B-R Research Home* (University of Illinois, *Engineering Experiment Station Bulletin* No. 383).

[7] G. L. Larson, D. W. Nelson, and John James. ASHVE Research Report No. 1011—Tests of three heating systems in an industrial type of building (ASHVE Transactions, Vol. 41, 1935, p. 185).

[8] F. A. Joy, J. J. Zabrony, and S. Bhaduri: *Insulating Value of Reflective Elements in an Attic Under Winter Conditions* (Pennsylvania State University, October 1956).

[9] F. A. Joy: Improving attic space insulating values (ASHAE Journal Section, *Heating, Piping and Air Conditioning*, January 1958, p. 223).

[10] F. B. Rowley, A. B. Algren, and C. E. Lund: *Methods of Moisture Control and Their Application to Building Construction* (University of Minnesota, *Engineering Experiment Station Bulletin* No. 17).

[11] F. C. Houghten, S. I. Taimuty, Carl Gutberlet, and C. J. Brown: ASHVE Research Report No. 1213—Heat loss through basement walls and floors (ASHVE Transactions, Vol. 48, 1942, p. 369).

[12] R. S. Dill, W. C. Robinson, and H. E. Robinson: *Measurements of Heat Losses from Slab Floor* (U. S. Department of Commerce, National Bureau of Standards, *Building Materials and Structures Report* BMS 103).

[13] H. D. Bareither, A. N. Fleming, and B. E. Alberty: *Temperature and Heat Loss Characteristics of Concrete Floors Laid on the Ground* (University of Illinois, *Small Homes Council Technical Report*).

[14] *Concrete Floors for Basementless Houses* (University of Illinois, *Small Homes Council Circular* No. F 4.3).

[15] J. R. Jamieson, R. W. Roose, and S. Konzo: Warm-air perimeter heating, Part III—Heat losses from floor slab (ASHVE Transactions, Vol. 58, 1952, p. 217).

[16] *Warm-Air Perimeter Heating* (National Warm Air Heating and Air Conditioning Association, Manual 4).

[17] *Slab-on-Ground Construction for Residences* (Building Research Advisory Board, National Research Council, Publication No. 385).

[18] *Heat Requirement Tables for Intermittently Heated Buildings* (A. and M. College of Texas, College Station, Texas, *Engineering Experiment Station Bulletin* No. 60) contains a set of tables applicable to either intermittent heating or cooling. Further information may be found in a paper by E. G. Smith: A method of compiling tables for intermittent heating (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, June 1942, p. 386).

CHAPTER 13

COOLING LOAD

Cooling Load Calculations; Design Conditions; Instantaneous Heat Load; Solar Radiation; Periodic Heat Flow; Tables for Calculating Solar Heat Gain Through Walls, Roofs, and Glass; Instantaneous Heat Gain vs. Cooling Loads; Load from Interior Partitions, Ceiling, and Floors; Load from Outdoor Air, Ventilation and Infiltration; Effect of Outdoor Air on Load; Heat Sources Within Conditioned Space; Moisture Transfer Heat Load; Miscellaneous Heat Loads; Apparatus Dew Point and Required Air Quantity Through Conditioning Equipment; Minimum Entering Air Temperature; Example Cooling Load Calculation

THE variables affecting cooling load calculations are numerous, often difficult to define precisely, and always intricately interrelated. Most of the components of the cooling load vary in magnitude over a wide range during a 24-hour period, and as the cyclic changes in load components are not usually in phase with each other, careful analysis is required to establish the resultant maximum cooling load for a building or zone. A zoned system must often handle peak loads in different zones at different hours.

Economic considerations must be of particular influence in the selection of equipment for cooling season operation in comfort air conditioning, and this fact, coupled with present inadequacies in available data and knowledge of the air-conditioning art, places a premium on the experienced judgment essential to successful design or practice. Variations in the weather, building occupancy, and other factors affecting load, necessitate carefully coordinated controls to regulate simultaneously the components and the equipment in order to maintain the desired room conditions.

The calculation procedures presented in this chapter deal with the various instantaneous rates of heat gain, both sensible and latent, in a conditioned space. There may be an appreciable difference between the *net instantaneous rate of heat gain* and the *total cooling load* at any instant. This difference is caused by the storage and subsequent release of heat by the structure and its contents. This thermal-storage effect may be quite important in determining an economical cooling equipment capacity. The lack of any adequate means of treating this storage quantitatively in its entirety for a complete structure, must be recognized in judging the procedures and data presented for calculating individual components of the net rate of instantaneous heat gain.

Solar heating calculations involve the same principles as cooling load calculations. Many of the data on solar radiation given in this chapter can be used in calculations for solar heating.

COOLING LOAD CALCULATIONS

Summer cooling load calculations, whether for industrial or comfort applications, require consideration of the following factors:

A. Design Conditions: (1) indoor conditions; (2) outdoor conditions; (3) ventilation rate.

B. Instantaneous Heat Load, Sensible and Latent: (1) load from solar radiation, sky radiation, and from outdoor-indoor temperature differential for glass areas and exterior walls and roofs, modified by periodic heat flow or lag factors depending on the type of structure; (2) load due to heat gain through interior partitions, ceilings, and floors; (3) load due to ventilation, either natural or mechanical; (4) load due to heat sources within the conditioned space such as people, lights, power equipment, and appliances; (5) load due to moisture transfer through permeable building materials; and (6) miscellaneous heat sources.

C. Determination of Air Quantity and Apparatus Dew Point.

These factors will be discussed in turn. The material presented leads to an illustrative procedure for a cooling load calculation, and a numerical example is given to demonstrate the calculations involved.

DESIGN CONDITIONS

Indoor Conditions

Indoor air conditions for human health and comfort have been and continue to be the subject of much discussion and research.

The effective temperature index, explained in Chapter 6, is probably the best available source of design criteria for comfort air-conditioning systems for buildings or enclosures in which the air and inside surface temperatures remain substantially equal; a condition that can safely be assumed for most ordinary comfort air-conditioning installations. Other sources of design specifications are to be found in the requirements of codes and ordinances, and in the varied long-term experiences of manufacturers, contractors, and engineering specialists.

Past experience, cumulative over many years, indicates that indoor design conditions, for which summer air-conditioning equipment is selected, should not exceed a temperature of 80 F or a relative humidity of 50 percent for the average job in the United States. If these conditions are exceeded, complaints of discomfort may be expected, especially with continuous occupancy. For very brief occupancy only, a slightly higher peak-load design temperature may be employed. In regard to the lower limit of humidity, complaints are not encountered for store installations operated down to 35 percent relative humidity or, for office jobs, somewhat lower. These observations apply to normal commercial practice in this country only. For extremes, such as tropical or very hot regions, it is regarded as more practicable to design for a peak-load outdoor-indoor temperature difference of about 15 to 20 F deg.

Table 1 offers typical design conditions for average requirements encountered. The values in line 1 would also apply in general for localities having a summer outdoor design tem-

Table 1 .... Design Room Conditions Usually Specified for Summer Average Peak Load in Comfort Air Conditioning[a]

Type of Installation	Dry-Bulb Temp	Wet-Bulb Temp[b]	Relative Humidity Percent	Grains Per Lb[b]	Effective Temp[c]
1. Ample capacity	78	65	50	72.7	72.2
2. Practical application	80	67	51	78.5	74.0
3. Occupancy—15 to 40 min	82	68	49	80.0	75.3

[a] Values in Table 1 are for *peak load* conditions. It is general practice to operate a system at approximately 75 F and 50 percent relative humidity at other than peak load.

[b] Psychrometric data for standard barometric pressure.

[c] Fig. 10, Chapter 6, air movement 15 to 25 fpm.

perature of 90 F or less; and the *15 to 40 min* occupancy values, or even somewhat higher dry-bulb temperatures, would indicate acceptable conditions for very hot localities. Table 1 *is to be used with good judgment*, for there is no universal rule that may be applied to indoor design conditions.

Guarantees of conditions to be maintained for summer operation are based upon a definite set of load conditions. At other than the guarantee load, the conditions produced by a system are determined by the balance of imposed load and equipment capacity, and by the method adopted for regulating the system operation. Complete specifications of indoor design conditions would include part-load and overload operation, particularly from the viewpoint of economy.

In the field of *industrial air conditioning*, indoor design conditions are established by the requirements of goods and processes, in addition to the comfort and efficiency of the workers. Many typical indoor design conditions for products in industrial air conditioning are given in Chapter 50.

The load calculations for either comfort or industrial air conditioning are usually made in accordance with a guarantee. In comfort applications, for example, this is frequently 80 F and 51 percent relative humidity or 78 F and 50 percent relative humidity. The system is *normally operated* at 75 or 76 F and approximately 50 percent relative humidity at all times as long as the equipment has capacity to maintain these conditions. During very hot weather, the room temperature will rise above the control point and the equipment will operate continuously. Normally, industrial jobs are operated at design conditions at all times if the optimum conditions are selected for the benefit of the product.

The indoor design conditions suggested have had reference to conditions to be maintained at the level of occupancy. For extremely high ceilings in public or industrial buildings, only the zone from 10 to 15 ft above the floor may be cooled to the full extent. The air temperature at the ceiling would be much higher, and this should be kept in mind when calculating the convective portion of the roof heat gain. A reduction of outdoor-to-indoor air temperature differential may be assumed in such instances but radiation from the inner roof surface is not diminished.

Outdoor Conditions

Summer climatic conditions and suggested design wet-bulb and dry-bulb temperatures are given in Table 2 for various locations in the United States. The highest temperature ever recorded is for the period of record shown. In some cases it should be noted that this period of record is comparatively short, and higher temperatures may be expected. In making comparisons for localities other than those shown in Table 2, due consideration must be given to elevation.

Column 6 of Table 2 indicates the design dry-bulb temperature suggested by the ASHAE Technical Advisory Committee on Weather Design Conditions. This temperature is the maximum hourly outdoor temperature which has been equalled or exceeded 2½ percent of the total hours of June, July, August, and September for the period of record, in this case the 5-year period 1935–1939, and should not be confused with the period of record given in Column 4 which applies only to highest temperature ever recorded. Since all of these data (Column 4) are based on airport records, they are not necessarily applicable to cities.

The data given in Columns 7 and 8 were obtained from local ASHAE Chapter Secretaries, and represent the design temperatures in local use. Where such information was not available, it was taken from a publication of the ARI[1] and from various other sources.

The Technical Advisory Committee on Weather Design Conditions has suggested that wet-bulb design temperature be taken as that wet-bulb temperature which has been equalled or exceeded in 5 percent of the hours during months of the period of record. While not available for this edition of THE GUIDE, due to the tremendous task of compiling these data, it is hoped that they will be available for some stations for future editions.

The wind velocity to be used in design should be that wind velocity which accompanies the design temperatures in each instance, but since these data are not available, the average summer wind velocities for the period of record were taken from U. S. Weather Bureau records revised to 1948. It is pointed out that this is not necessarily the velocity which coincides with the design temperatures, but it may serve as a guide to the designer.

Other weather data, such as *daily range of temperature*, are useful particularly when making cooling load calculations for an early morning peak on an east exposure. Daily range of temperature is the difference between the average of the daily maximum dry-bulb temperatures and the average of the daily minimum temperatures. This range is highest in semi-arid or desert regions and at high elevations, and lowest near the oceans or very large lakes. The daily range of temperature (Fahrenheit degrees) in July for the principal areas of the United States can be approximated from the following tabulation:

East Sea Shore	12 to 18
East of Mississippi River	19 to 24
Gulf Sea Shore	12 to 18
Great Lakes Shore	18 to 21
Mississippi River to Rocky Mountains	24 to 33
Rocky Mountain Area	33 to 42
West Coastal States	20 to 35
West Sea Shore	15 to 20

Ventilation Rate

The introduction of outdoor air is necessary for the ventilation of conditioned spaces. Chapter 6 suggests *minimum* outdoor air requirements for representative applications; but it is to be emphasized that minimum requirements are not necessarily *adequate* requirements for all psychological attitudes and physiological responses. Where maximum economy in space and load are essential, as in submarines or other restricted spaces, as little as 1 cfm of outdoor air per person has been found to be sufficient provided that satisfactory

ventilation is simultaneously obtained by an adequate decontamination of recirculated air.[2]

Local codes and ordinances frequently specify ventilation requirements for public places and for industrial installations. For operating rooms, minimum requirements for safe practice are given in a National Board of Fire Underwriters' pamphlet.[3] This pamphlet does not require 100 percent outdoor air in operating rooms, although 100 percent outdoor air is normally used and recommended.

Recommended and minimum ventilation rates for the most common applications are summarized in Table 3. For further general applications, a basis of estimating the cfm per person may be taken as:

1. People not smoking	7½	*Recommended*	5 *Minimum*
2. People smoking	40	*Recommended*	25 *Minimum*

The cooling load due to the introduction of outdoor air for ventilation is determined once the indoor and outdoor design conditions are fixed. Calculations will be discussed subsequently.

INSTANTANEOUS HEAT LOAD

The total cooling load is frequently divided for convenience into two components, *sensible heat* and *latent heat*. While this subdivision is not imperative, past practice has found it convenient.

A gain of *sensible heat* is considered to occur when there is a direct addition of heat to the enclosure by any one or all of the mechanisms of conduction, convection, and radiation. A gain of *latent heat* is considered to occur when there is an addition of water vapor to the air of the enclosure. For example, when the humidity in an enclosure is increased by water vapor emitted by human occupants, or by water vapor resulting from a process such as cooking, the heat required to vaporize the water does *not* come from the air. Maintenance of a constant humidity ratio in a sealed enclosure requires the condensation of water vapor in the cooling apparatus at a rate equal to its rate of addition within the enclosure. The rate of heat removal from this condensing vapor would be substantially equal to the product of the rate of condensation and the latent heat of condensation; this product, expressed in Btu per hour, would be called a *latent heat* load.

As a further example, the infiltration of outdoor air with a high dry-bulb temperature and a high humidity ratio, and the corresponding escape of room air at a lower dry-bulb temperature and a lower humidity ratio, would increase both the *sensible heat* load and the *latent heat* load.

SOLAR RADIATION

Magnitude of Solar Radiation

If a plane surface were set perpendicular to the sun's rays (i.e., for *normal incidence*) outside the earth's atmosphere, it would receive solar radiation of about 420 Btu per (hr) (sq ft). A similarly oriented surface, at the surface of the earth, would receive considerably less solar energy because a large part of the radiation is scattered in passing through the air; moisture, smoke, and dust which comprise the earth's atmosphere, and also, because some of the atmospheric constituents, notably water vapor, ozone, and carbon dioxide, absorb solar radiation. The intensity of solar radiation varies with wavelength, reaching a peak at about 0.5 microns (a micron equals 1/1000 of a millimeter) and, for practical purposes, is confined to the radiation spectrum between 0.3 and 2.3 microns. The effects of scattering and absorption vary with the wavelength, but to make an exact analysis of these phenomena is impracticable in air-conditioning estimates. The important principle to remember is that the total radiation I_t, received by a surface at the earth, is the sum of I_D and I_d, where

$I_D = KI_{Dn}$ = the *direct* or *beamed* solar radiation, Btu per (hour) (square foot of receiving surface).

I_{Dn} = the direct solar radiation on a plane normal to the sun's rays, Btu per (hour) (square foot of receiving surface).

I_d = the *sky* or *diffuse* solar radiation, Btu per (hour) (square foot of receiving surface). This comes principally from the atmosphere itself as a consequence of scattering. Vertical surfaces also receive solar radiation by reflection of direct and diffuse radiation from the ground and other objects. Such radiation is usually diffuse. The diffuse radiation strikes at *all* angles.

I_t = total incident solar radiation, Btu per (hour) (square foot of receiving surface).

K = cosine of the angle of incidence, θ. For a vertical surface, θ is defined in Fig. 1.

Standardized, practical-purpose values of the *direct* solar radiation I_{Dn} incident upon a plane *perpendicular* to the sun's rays at the earth's surface, have been proposed by Moon.[4] Table 4 gives these values. They are representative of a *clear* summer day at sea level elevation, and are nearly identical with values derived from suggested design sol-air temperatures for Lincoln, Nebraska.[5] Values typical of a humid industrial area derived from sol-air data for New York City[6] are also given in Table 4. Day-to-day changes in the amount of dust and water vapor in the atmosphere cause large differences in solar intensity values observed on cloudless days at a given locality. For example, it has been observed in Cleveland that values of the order of those given for industrial atmospheres are usually associated with dry-bulb and wet-bulb temperatures near the design values of 95 F and 75 F (67 F dew point). On the other hand, values approaching or exceeding those for a clear atmosphere are often encountered during Cleveland summers, but with dew-point and maximum dry-bulb temperatures 10 to 15 deg lower. Considerable judgment, therefore, is required in selecting solar intensity values for design purposes.

Data regarding the irradiation of vertical and horizontal surfaces by diffuse or sky radiation are few. Suggested design values for a 40-deg latitude on August 1 (18-deg declination,

(*Continued on p. 176*)

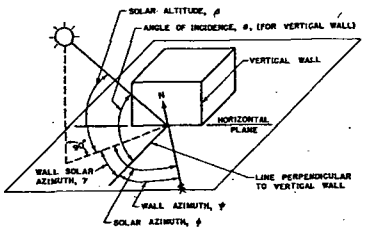


Fig. 1 .... Definition of Solar Angles

Table 2 .... Summer Climatic Conditions[a]

Suggested Design Wet-Bulb and Dry-Bulb Temperatures—F

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] Ft	Col. 4 Period of Record[d]	Col. 5 Highest Temp. Ever Recorded[d] F	Col. 6 Design Dry-Bulb Temp. on T.A.C. 2½% Basis[a]	Col. 7 Design Dry-Bulb Temp. in Common Use[f]	Col. 8 Design Wet-Bulb Temp. in Common Use[f]	Col. 9 Average Summer Wind Velocity[g] Mph
Ala.	Anniston CO	733	1893–1947	105	...	95	78	...
	Birmingham CO	711	1893–1945	107	...	95	78	5.4
	Birmingham AP	615	1939–1947	103	94	...	...	...
	Mobile CO	143	1872–1947	103	...	95	80	8.0
	Mobile AP	219	1940–1947	104	92	...	...	...
	Montgomery CO	293	1872–1947	107	...	95	78	...
	Montgomery AP	226	1938–1944[d]	103	...	...	...	...
Ariz.	Flagstaff CO	6957	1899–1947	93	...	90	65	...
	Kingman AP	3473	1935–1939	107	99	...	...	...
	Phoenix CO	1122	1895–1947	118	...	105	76	6.0
	Phoenix AP	1112	1933–1947	117	107	...	...	...
	Tucson AP	2561	1935–1939	112	102	105	72	...
	Winslow CO	4853	Up to 1946	107	...	100	70	...
	Winslow AP	4899	1937–1947	103	95	...	...	...
	Yuma CO	146	1876–1946	120	...	110	78	...
Ark.	Fort Smith CO	545	1882–1945	113	...	95	76	6.1
	Fort Smith AP	463	1945–1947	107	...	...	...	...
	Little Rock CO	451	1879–1942[d]	110	...	95	78	6.2
	Little Rock AP	282	1942–1947	107	97	...	...	...
Calif.	Bakersfield AP	499	1937–1946	113	104	105	70	...
	Burbank AP	740	1931–1947[d]	111	94	...	...	...
	Daggett AP	1925	1935–1939	113	104	...	...	...
	Eureka CO	132	1886–1947	85	...	90	65	...
	Fresno CO	387	1887–1939	115	...	105	74	7.9
	Fresno AP	281	1939–1947	111	103	...	...	...
	Los Angeles CO	534	1877–1947	109	...	90	70	5.8
	Oakland AP	21	1929–1947	102	80	85	65	...
	Red Bluff CO	305[e]	1877–1934	115	...	100	70	...
	Red Bluff AP	346	1944–1947	112	...	...	...	...
	Redding AP	579	1935–1939	112	101	...	...	...
	Sacramento CO	116	1877–1947	114	...	100	72	7.9
	Sacramento AP	22	1938–1947	108	...	...	...	...
	San Diego CO	90	1871–1940	110	...	85	68	...
	San Diego AP	34	1939–1947	106	79	...	...	...
	San Francisco CO	164	1875–1947	101	...	85	65	10.7
	San Jose CO	100	Up to 1946	106	...	91	70	...
	Williams AP	124	1935–1939	116	103	...	...	...
Colo.	Denver CO	5398	1871–1947	105	...	95	64	6.9
	Denver AP	5379	1934–1947	104	93	...	...	...
	Durango CO	6558	Up to 1946	99	...	95	65	...
	Grand Junction CO	4587[e]	Up to 1946	105	...	95	65	6.3
	Pueblo CO	4770	1889–1938	104	...	95	65	...
	Pueblo AP	4810	1939–1947	104	95	...	...	...
Conn.	Hartford CO	229	1905–1940[d]	101	...	93	75	...
	Hartford AP	20	1940–1947	98	88	...	...	...
	New Haven CO	180	1872–1947	101	...	95	75	7.4
	New Haven AP	17	1943–1947	94	84	...	...	...
D. C.	Washington CO	128	1871–1947	106	...	95	78	5.9
	Washington AP	20	1935–1939	106	92	...	...	...
Fla.	Apalachicola CO	23	1922–1947	102	...	95	80	...
	Jacksonville CO	104	1871–1947	104	...	95	78	8.4
	Jacksonville AP	29	1938–1947	105	94	...	...	...
	Key West CO	23	1871–1947	100	...	98	78	...
	Key West AP	48	1939–1947	95	...	...	...	...
	Miami CO	253	1896–1947	96	...	91	79	8.7
	Miami AP	13	1940–1947	100	89	...	...	...
	Pensacola CO	67	1879–1947	103	...	95	78	...
	Pensacola AP	113	1943–1947	105	...	...	...	...
	Tampa CO	111	1890–1940	98	...	95	78	7.4
	Tampa AP	12	1941–1946[d]	98	...	...	...	...
	Titusville AP	52	1935–1939	98	90	...	...	...
Ga.	Atlanta AP	1020	1935–1939	102	93	95	76	7.9
	Augusta CO	195	1871–1946	106	...	98	76	...
	Augusta AP	424	1939–1947	105	...	...	...	...

Table 2 .... Summer Climatic Conditions[a] (*Continued*)

Suggested Design Wet-Bulb and Dry-Bulb Temperatures—F

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] Ft	Co. 4 Period of Record[d]	Col. 5 Highest Temp. Ever Recorded[d] F	Col. 6 Design Dry-Bulb Temp. on T.A.C. 2½% Basis[a]	Col. 7 Design Dry-Bulb Temp. in Common Use[f]	Col. 8 Design Wet-Bulb Temp. in Common Use[f]	Col. 9 Average Summer Wind Velocity[g] Mph
Ga.	Macon CO	408	1899–1947	105	...	95	78	...
	Macon AP	432	1939–1947	102	...	...	...	...
	Savannah CO	115	1871–1945[d]	105	...	95	78	8.0
	Savannah AP	56	1939–1947	105	93	...	...	...
Idaho	Boise CO	2818	1864–1939	112	...	95	65	5.8
	Boise AP	2849	1939–1947	109	95	...	...	...
	Burley AP	4150	1935–1939	104	94	...	...	...
	Idaho Falls AP	4744	1935–1939	100	88	...	...	...
	Lewiston CO	763	1900–1944[d]	117	...	95	65	...
	Pocatello CO	4522	1899–1947	105	...	95	65	...
	Pocatello AP	4467	1938–1947	103	92	...	...	...
Ill.	Cairo CO	319	1872–1947	106	...	98	78	...
	Chicago CO	601	Up to 1946	105	...	95	75	9.5
	Chicago AP	615	1935–1939	107	92	...	...	...
	Moline AP	594	1932–1947	106	94	96	76	...
	Peoria AP	660	1935–1939	111	94	96	76	8.2
	Springfield CO	603	1879–1947	110	...	98	77	...
	Springfield AP	608	1930–1947	109	96	...	...	...
Ind.	Evansville CO	464	1897–1940	108	...	95	78	7.0
	Fort Wayne CO	885	1911–1941[d]	106	...	95	75	...
	Helmer AP	970	1935–1939	107	89	...	...	...
	Indianapolis CO	816	1871–1946	106	...	95	76	8.9
	Indianapolis AP	800	1932–1946	107	91	...	...	...
	Terre Haute CO	1146	1893–1946	110	...	95	78	...
	Terre Haute AP	589	1941–1946	103	...	...	...	...
Iowa	Davenport CO	648	1872–1947	111	...	95	78	...
	Des Moines AP	979	1935–1939	111	95	95	78	8.6
	Dubuque CO	740	1874–1947	110	...	95	78	...
	Keokuk CO	637	1872–1946	113	...	95	78	...
	Sioux City CO	1093[e]	1889–1944[d]	111	...	95	78	...
	Sioux City AP	1098	1940–1946	108	...	...	...	...
Kans.	Concordia CO	1425	1885–1947	116	...	95	78	...
	Dodge City CO	2515	1874–1942[d]	109	...	95	78	...
	Dodge City AP	2599	1942–1947	109	...	...	...	...
	Topeka CO	991	1887–1947	114	...	100	78	...
	Topeka AP	883	1946–1947	108	...	...	...	...
	Wichita CO	1497	1888–1939	114	...	100	75	11.8
	Wichita AP	1423	1939–1947	109	100	...	...	...
Ky.	Louisville CO	563	1871–1947	107	...	95	78	7.2
	Louisville AP	544	1937–1947	103	93	...	...	...
La.	New Orleans CO	85	1874–1947	102	...	95	80	6.9
	New Orleans AP	8	1937–1947	100	93	...	...	...
	Shreveport AP	179	1935–1939	109	98	100	78	7.0
Maine	Eastport CO	100	1873–1947	93	...	90	70	...
	Portland CO	185	1885–1940	103	...	90	73	8.7
	Portland AP	65	1940–1947	99	...	...	...	...
Md.	Baltimore CO	114	1871–1947	107	...	95	78	7.4
	Baltimore AP	43	1935–1939	105	91	...	...	...
Mass.	Boston CO	356	1870–1935	104	...	92	75	12.5
	Boston AP	45	1936–1947	101	87	...	...	...
	Nantucket CO	45	1886–1947	92	...	95	75	...
	Nantucket AP	48	1946–1947	82	...	...	...	...
Mich.	Alpena CO	615	1874–1946	104	...	95	75	...
	Detroit CO	1000	1873–1933	104	...	95	75	9.5
	Detroit AP	632	1934–1947	105	89	...	73[h]	...
	Lansing CO	861	1910–1947	102	...	95	75	...
	Lansing AP	863	1940–1947	98	...	...	...	...
	Marquette CO	721	1874–1947	108	...	93	73	...
Minn.	Duluth CO	1133	1874–1947	106	...	93	73	...
	Duluth AP	1413	1941–1947	95	...	...	...	...

Table 2 .... Summer Climatic Conditions[a] (*Continued*)

Suggested Design Wet-Bulb and Dry-Bulb Temperatures—F

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] Ft	Col. 4 Period of Record[d]	Col. 5 Highest Temp. Ever Recorded[d] F	Col. 6 Design Dry-Bulb Temp. on T.A.C. 2½% Basis[e]	Col. 7 Design Dry-Bulb Temp. in Common Use[f]	Col. 8 Design Wet-Bulb Temp. in Common Use[f]	Col. 9 Average Summer Wind Velocity[g] Mph
Minn.	Minneapolis CO	945	1890–1947	108	...	95	75	10.2
	Minneapolis AP	873	1938–1947	104	...	...	...	...
	St. Paul CO	951	1871–1933	104	...	95	75	...
	St. Paul AP	708	1937–1947	104	91	...	...	...
Miss.	Meridian CO	410	1889–1947	105	...	95	79	4.6
	Meridian AP	298	1939–1947	105	...	...	...	...
	Vicksburg CO	316	1874–1947	104	...	95	78	6.4
	Vicksburg AP	266	1941–1947	104	...	...	...	...
Mo.	Columbia CO	739	1889–1947	111	...	100	78	...
	Columbia AP	787	1939–1947	102	...	...	...	...
	Kansas City AP	780	1935–1939	112	101	100	76	9.1
	St. Louis CO	646	1871–1947	110	...	95	78	9.5
	St. Louis AP	597	1930–1947	111	97	...	...	...
	Springfield AP	1270	1935–1939	105	96	...	...	8.7
Mont.	Billings AP	3584	1935–1947	106	92	90	66	...
	Butte AP	5538	1931–1947	100	85	...	...	...
	Havre CO	2498	1880–1947	108	...	95	70	...
	Helena CO	4175	1880–1940	103	...	95	67	8.1
	Kalispell CO	3004	1897–1947	101	...	95	65	...
	Miles City AP	2629	1935–1939	108	97	...	...	...
Nebr.	Lincoln CO	1189	1887–1947	115	...	95	78	9.7
	Lincoln AP	1185	1933–1947	115	101	...	...	...
	North Platte CO	2815	1874–1947	109	...	95	78	8.1
	North Platte AP	2788	1935–1939	109	98	...	...	...
	Omaha CO	1219	1873–1935[d]	111	...	95	78	...
	Omaha AP	1009	1935–1947	114	98	...	...	...
	Valentine CO	2627	1889–1947	110	...	95	78	...
Nev.	Elko AP	5079	1935–1939	102	92	...	...	...
	Las Vegas AP	1882	1937–1947	117	108	115	75	...
	Reno CO	4588	1905–1942	106	...	95	65	7.2
	Reno AP	4417	1940–1947	105	93	...	...	...
	Winnemucca CO	4293	1871–1947	108	...	95	65	...
N. H.	Concord CO	343	1871–1941[d]	102	...	90	73	4.9
	Concord AP	359	1941–1947	99	...	...	...	...
N. J.	Atlantic City CO	45	1874–1947	104	...	95	78	...
	Camden AP	20	1935–1939	105	91	...	...	...
	Newark AP	15	1931–1947	104	89	95	75	...
	Trenton CO	144	1866–1946	106	...	95	78	8.8
N. M.	Albuquerque CO	5022	1931–1933[d]	99	...	95	70	7.8
	Albuquerque AP	5319	1933–1947	101	93	...	...	...
	El Morro AP	7120	1935–1939	92	84	...	...	...
	Rodeo AP	4116	1935–1939	104	97	...	...	...
	Roswell CO	3643	1905–1947[d]	107	...	95	70	...
	Tucumcari AP	4054	1935–1939	107	97	...	...	...
N. Y.	Albany CO	114	1874–1947	104	...	93	75	7.5
	Albany AP	280	1938–1947	99	88	...	...	...
	Binghamton CO	915	1891–1946	103	...	95	75	...
	Binghamton AP	836	1942–1947	97	...	...	...	...
	Buffalo AP	726	1935–1939	95	86	93	73	12.1
	Canton CO	458	1906–1947	99	...	90	73	8.2
	Elmira AP	948	1935–1939	96	88	...	...	...
	New York CO	425	1871–1947	102	...	95	75	12.5
	Oswego CO	363	1871–1947	100	...	93	73	...
	Rochester CO	609	1872–1947	102	...	95	75	...
	Rochester AP	560	1935–1939	98	89	...	...	...
	Syracuse CO	465	1902–1940	102	...	93	75	...
	Syracuse AP	404	1940–1947	97	88	...	...	...
N. C.	Asheville CO	2280	1902–1947	99	...	93	75	5.6
	Charlotte CO	809	1878–1947	103	...	95	78	...
	Charlotte AP	757	1939–1947	103	93	...	...	...
	Greensboro AP	896	1928–1947	101	91	95	78	...

Table 2 .... Summer Climatic Conditions[a] (*Continued*)

Suggested Design Wet-Bulb and Dry-Bulb Temperatures—F

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] Ft	Col. 4 Period of Record[d]	Col. 5 Highest Temp. Ever Recorded[d] F	Col. 6 Design Dry-Bulb Temp. on T.A.C. 2½% Basis[e]	Col. 7 Design Dry-Bulb Temp. in Common Use[f]	Col. 8 Design Wet-Bulb Temp. in Common Use[f]	Col. 9 Average Summer Wind Velocity[g] Mph
N. C.	Raleigh CO	405	1887–1947	104	...	95	78	6.3
	Raleigh AP	446	1944–1947	102	93	...	...	...
	Wilmington CO	78	1871–1947	103	...	95	78	8.4
N. D.	Bismarck CO	1675	1875–1940	114	...	95	73	9.5
	Bismarck AP	1655	1940–1947	109	96	...	...	...
	Devils Lake CO	1481	1904–1947	112	...	95	70	...
	Dickinson AP	2599	1935–1939	112	94	...	...	...
	Fargo AP	900	1935–1939	115	93	95	75	...
	Pembina AP	830	1935–1939	109	92	...	...	...
	Williston CO	1919	1879–1947	110	...	95	73	...
Ohio	Akron AP	104	1935–1939	101	88	95	75	...
	Cincinnati CO	772	1870–1947	108	...	95	78	5.6
	Cincinnati AP	488	1931–1947	108	94	...	...	...
	Cleveland CO	669	1871–1946	100	...	95	75	11.1
	Cleveland AP	813	1930–1946	107	90	...	...	...
	Columbus CO	812	1878–1946	106	...	95	76	...
	Columbus AP	820	1939–1947	100	90	...	...	...
	Dayton CO	1086	1883–1943[d]	108	...	95	78	...
	Dayton AP	1002	1940–1947	99	...	...	...	...
	Sandusky CO	608	1878–1946	105	...	95	75	...
	Toledo CO	668	1871–1947	105	...	95	75	...
	Toledo AP	626	1940–1947	100	91	...	...	...
Okla.	Ardmore AP	762	1935–1939	110	99	...	...	...
	Oklahoma City CO	1264	1890–1947	113	...	101	77	9.8
	Oklahoma City AP	1311	1939–1947	109	99	...	...	...
	Tulsa AP	686	1932–1947	109	100	101	77	...
	Waynoka AP	1529	1935–1939	115	103	...	...	...
Ore.	Arlington AP	881	1935–1939	111	95	...	...	...
	Baker CO	3501	1889–1947	104	...	90	66	...
	Baker AP	3374	1939–1947	103	90	...	...	...
	Eugene CO	366	1890–1942	104	...	90	68	...
	Eugene AP	368	1942–1947	105	88	...	...	...
	Medford CO	1428	1911–1929	110	...	95	70	...
	Medford AP	1343	1929–1947	115	95	...	...	...
	Portland CO	98	1874–1947	107	...	90	68	6.5
	Portland AP	25	1940–1947	105	87	...	...	...
	Roseburg CO	523	1877–1947	109	...	90	66	...
Pa.	Curwensville AP	2219	1943–1947	90	82	...	...	...
	Erie CO	771	1873–1946	98	...	93	75	...
	Erie AP	736	1935–1939	96	85	...	...	...
	Harrisburg AP	339	1935–1939	103	91	95	...	...
	Philadelphia CO	200	1871–1947	106	...	95	78	9.7
	Philadelphia AP	18	1940–1947	100	...	...	...	...
	Pittsburgh CO	929	1875–1947	103	...	95	75	8.9
	Pittsburgh AP	1284	1935–1947	102	88	...	...	...
	Reading CO	311	1913–1947	105	...	95	75	...
	Scranton CO	877	1901–1947	103	...	95	75	...
	Sunbury AP	448	1935–1939	101	89	...	...	...
R. I.	Block Island CO	46	1881–1947	93	...	95	75	...
	Providence CO	77	1904–1947	100	...	93	75	9.5
S. C.	Charleston CO	59	1871–1947	104	...	95	78	9.8
	Charleston AP	51	1940–1947	103	91	...	...	...
	Columbia CO	401	1887–1947	106	...	95	75	...
	Columbia AP	227	1939–1947[d]	104	...	...	...	...
S. D.	Huron CO	1342	1881–1938[d]	111	...	95	75	10.3
	Huron AP	1287	1938–1947	110	...	...	...	...
	Rapid City CO	3309	1888–1947	106	...	95	70	7.9
	Rapid City AP	3220	1939–1947	108	...	...	...	...
Tenn.	Chattanooga CO	952	1879–1947	103	...	95	76	5.6
	Chattanooga AP	675	1940–1947	105	94	...	...	...
	Knoxville CO	1024	1871–1942[d]	104	...	95	75	5.7
	Knoxville AP	1007	1942–1947	102	94	...	...	...

Table 2....Summer Climatic Conditions[a] (*Continued*)

Suggested Design Wet-Bulb and Dry-Bulb Temperatures—F

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[a] Ft	Col. 4 Period of Record[d]	Col. 5 Highest Temp. Ever Recorded[d] F	Col. 6 Design Dry-Bulb Temp. on T.A.C. 2½% Basis[e]	Col. 7 Design Dry-Bulb Temp. in Common Use[f]	Col. 8 Design Wet-Bulb Temp. in Common Use[f]	Col. 9 Average Summer Wind Velocity[g] Mph
Tenn.	Memphis CO	348	1872–1941[d]	106	...	95	78	7.3
	Memphis AP	267	1941–1947	105	96	...	...	...
	Nashville CO	714	1871–1947	106	...	95	78	...
	Nashville AP	610	1939–1947	104	...	...	...	...
Texas	Abilene CO	1748	1885–1944[d]	111	...	100	74	...
	Abilene AP	1756	1940–1947	109	97	...	...	...
	Amarillo CO	3686	1892–1941	107	...	100	72	11.8
	Amarillo AP	3595	1941–1947	106	96	...	...	...
	Austin CO	625	1897–1942	109	...	100	78	...
	Austin AP	625	1942–1947	104	...	...	...	...
	Brownsville CO	140	1922–1943[d]	102	...	100	80	7.0
	Brownsville AP	25	1943–1947	100	...	...	...	...
	Corpus Christi CO	21	1887–1942	105	...	95	80	...
	Corpus Christi AP	45	1943–1946	101	...	...	...	...
	Dallas CO	732	1913–1940	110	...	100	78	9.3
	Dallas AP	520	1940–1947	109	99	...	...	...
	Del Rio CO	1020	1905–1947	111	...	100	78	...
	El Paso CO	3792	1887–1942	106	...	100	69	8.4
	El Paso AP	3956	1939–1947	104	97	...	...	...
	Fort Worth CO	708	1898–1939	112	...	100	78	9.5
	Fort Worth AP	728	1940–1947	110	...	...	...	...
	Galveston CO	128	1871–1947	101	...	95	80	9.7
	Galveston AP	9	1939–1947	101	...	...	...	...
	Houston CO	198	1888–1947	108	...	95	80	8.8
	Houston AP	73	1932–1947	105	93	...	...	...
	Palestine CO	555	1881–1947	108	...	100	78	...
	Port Arthur CO	64	1917–1947	102	...	95	79	...
	Port Arthur AP	21	1944–1947	98	...	...	...	...
	San Antonio CO	770	1885–1941[d]	107	...	100	78	7.8
	San Antonio AP	800	1942–1947	104	98	...	...	...
	Waco AP	513	1931–1947	111	98	...	...	...
	Wink AP	2811	1935–1939	110	99	...	...	...
Utah	Milford AP	5095	1935–1939	103	94	...	...	...
	Modena CO	5472	1901–1947	101	...	95	65	...
	Salt Lake City CO	4346	1874–1947	105	...	95	65	9.8
	Salt Lake City AP	4254	1928–1947	106	95	...	...	...
Vt.	Burlington CO	409	1884–1943[d]	100	...	90	73	8.5
	Burlington AP	335	1943–1947	101	...	...	...	...
Va.	Cape Henry CO	24	1874–1947	104	...	95	78	...
	Lynchburg CO	644	1874–1944	106	...	95	75	...
	Lynchburg AP	*951*	*1944–1947*	*100*	...	...	...	...
	Norfolk CO	91	1871–1947	105	...	95	78	10.1
	Richmond CO	180	1897–1947	107	...	95	78	6.4
	Richmond AP	172	1929–1947	104	92	...	...	...
	Roanoke AP	1194	1935–1939	103	90	95	76	...
Wash.	Ellensburg AP	1731	1935–1939	105	90	...	...	...
	North Head CO	199	1884–1947	97	...	85	65	...
	Seattle CO	104	1890–1947	100	...	85	65	7.7
	Seattle AP	47	1928–1947	99	81	...	...	...
	Spokane CO	2030	1881–1941[d]	108	...	93	65	6.5
	Spokane AP	1974	1941–1947	104	92	...	...	...
	Tacoma CO	279	1897–1947	98	...	85	64	...
	Tatoosh Island CO	110	1883–1947	88	...	...	...	...
	Yakima CO	1160	1928–1946	110	...	95	65	...
	Yakima AP	1066	1944–1947	103	...	...	...	...
W. Va.	Parkersburg CO	685	1888–1947	106	...	95	75	5.2
Wisc.	Green Bay CO	598	1888–1947	104	...	95	75	9.2
	La Crosse CO	725	1872–1947	108	...	95	75	6.4
	La Crosse AP	677	1943–1947	96	91	...	...	...
	Madison CO	1008	1858–1947	107	...	95	75	7.9
	Madison AP	884	1935–1939	106	89	...	...	...
	Milwaukee CO	744	1870–1947	105	...	95	75	9.8
	Milwaukee AP	707	1927–1947	106	87	...	...	...

Table 2....Summer Climatic Conditions[a] (*Concluded*)

Suggested Design Wet-Bulb and Dry-Bulb Temperatures—F

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[a] Ft	Col. 4 Period of Record[d]	Col. 5 Highest Temp. Ever Recorded[d] F	Col. 6 Design Dry-Bulb Temp. on T.A.C. 2½% Basis[e]	Col. 7 Design Dry-Bulb Temp. in Common Use[f]	Col. 8 Design Wet-Bulb Temp. in Common Use[f]	Col. 9 Average Summer Wind Velocity[g] Mph
Wyo.	Cheyenne CO	6144	1873–1935	100	...	95	65	9.2
	Cheyenne AP	*6161*	*1935–1947*	*100*	89	...	...	...
	Lander CO	5448	1891–1946	102	...	95	65	...
	Lander AP	5568	1936–1947	97	...	...	...	...
	Rock Springs AP	6746	1932–1942	98	87	...	...	...

Summer Climatic Conditions for Canada[j]

Col. 1 Prov. of Canada[j]	Col. 2 Station[b]	Col. 3 Elevation[a] Ft	Col. 4 Period of Record[k]	Col. 5 Highest Temp. on Record[k] F	Col. 6 Summer Design Temp.[l]—F 1%	Col. 7 2%	Col. 8 5%	Col. 9 10%	Col. 10 Design DB Temp. in Common Use[f]	Col. 11 Design WB Temp. in Common Use[f]	Col. 12 Avg. Summer Wind Velocity[m] Mph
Alta.	Calgary AP	3540[c]	1921–1950	97	86	83	80	77	90	66	9.7
	Edmonton AP	2219[c]	1921–1950	99	87	83	80	77	90	68	8.9
	Medicine Hat AP	2365[c]	1921–1950	106	92	89	86	82	90	65	9.1
B.C.	Vancouver AP	22[c]	1938–1950	87	80	77	75	72	80	67	7.8
Man.	Winnipeg AP	786[c]	1921–1950	108	89	86	84	81	90	71	11.5
N.B.	Fredericton CO	164[c]	1921–1950	102	89	87	84	81	90	75	
	Saint John CO	119[c]	1921–1950	93	80	77	75	72			7.9
N.S.	Halifax CO	83[c]	1921–1950	94	80	78	76	73	90	75	6.6
	Sydney AP	197[c]	1921–1950	95	85	83	80	75			9.9
Ont.	Fort William AP	644[c]	1921–1950	104	85	82	79	76			8.4
	North Bay AP	1210[c]	1925–1950	99	83	80	77	74			9.6
	Ottawa AP	339[c]	1921–1950	102	89	86	84	81	90	75	8.9
	Sault Ste. Marie CO	635[c]	1921–1950	99	83	80	77	74	93	75	
	Toronto CO	379[c]	1921–1950	105	88	86	83	81	93	75	8.1
	Toronto AP	578[c]	1938–1950	101	90	87	84	81			8.7
P.E.I.	Charlottetown CO	74[c]	1921–1950	98	82	79	77	74			8.7
Que.	Montreal CO	187[c]	1921–1950	97	87	84	82	79	90	75	9.9
	Montreal AP	98[c]	1942–1950	96	88	85	83	80			9.5
	Quebec CO	296[c]	1921–1950	96	85	82	80	77	90	75	9.0
Sask.	Regina AP	1884[c]	1921–1950	111	91	88	85	81	90	71	12.4
	Saskatoon AP	1645[c]	1921–1950	104	92	89	85	81	90	70	10.7

Notes for Table 2

[a] Data compiled from U. S. Weather Bureau Data and various other sources.

[b] Column 2. The station designation AP or CO indicates airport or city office station, respectively.

[c] Column 3. The elevations marked c are ground elevations of the station. All other elevations given are the actual elevations of the thermometer bulb above mean sea level, corrected to 1948.

[d] The periods of record indicated apply only to the highest temperature ever recorded shown in Column 5, and do not necessarily include all of the summer months of the first year indicated. The last year indicated includes July or August, except those marked d which terminate prior to July of that year.

[e] It should be noted that Column 6 in the United States section of the table applies only to airports, as these data for city stations are not available at this time. The temperature shown is the maximum hourly outdoor temperature which has been equalled or exceeded 2½ percent of the total hours of June, July, August, and September for the 8-year period 1935–1939 inclusive. It is pointed out that in most cases the airport stations are outside of the city, and that these data would apply primarily to rural areas.

[f] Columns 7 and 8 in the United States section and Columns 9 and 10 in the Canadian section of the table record wet- and dry-bulb temperatures in use by ASHAE members as reported by Chapter secretaries for various stations. Where such values were not available, the design temperatures published by API or obtained from various other sources have been inserted.

[g] The average wind velocities in Column 9 in the United States section were furnished by the U. S. Weather Bureau, corrected to 1948. In general these velocities are averages for the months of June through September.

[h] The bulletin published by the ASHAE for the annual weather data of Detroit indicates 73 F as the design wet-bulb temperature which has been equalled or exceeded 5 percent of the hours for period 1935–39.

[i] Blank spaces indicate data not available.

[j] The data for Canada were compiled by D. W. Boyd and Morley Thomas from the records of the Meteorological Division, National Research Council of Canada.

[k] The highest temperature ever recorded in the Canadian section of the table is for the 30-year period, 1921–1950, where available. In some cases the stations have been relocated and in a few instances, some years are missing.

[l] In the Canadian section of the table, the summer design dry-bulb temperatures for the basis of *1*, 2½, *5*, and *10* percent are the Fahrenheit temperature values at or above which these percentages of all the July hourly outdoor temperatures occur, for the years 1941–1950 inclusive.

[m] Most of the wind speeds are based on the periods ending in 1947 which are somewhat shorter than the periods for the highest temperatures. The three months June, July, and August were used.

Table 3 .... Outdoor Air Requirements[a]

Application	Smoking	Cfm per Person[b]		Cfm per Sq[b] Ft of Floor
		Recommended	Minimum[c]	Minimum[c]
Apartment				
Average	Some	20	10	....
DeLuxe	Some	20	10	....
Banking space	Occasional	10	7½	....
Barber shops	Considerable	15	10	....
Beauty parlors	Occasional	10	7½	....
Brokers' board rooms	Very heavy	50	20	....
Cocktail bars		40	25	....
Corridors (supply or exhaust)		...	...	0.25
Department stores	None	7½	5	0.05
Directors' rooms	Extreme	50	30	....
Drug stores[e]	Considerable	10	7½	....
Factories[d, f]	None	10	7½	0.10
Five and Ten Cent stores	None	7½	5	....
Funeral parlors	None	10	7½	....
Garages[d]		...	...	1.0
Hospitals				
Operating rooms[f, g]	None	...	...	2.0
Private rooms	None	30	25	0.33
Wards	None	20	10	....
Hotel rooms	Heavy	30	25	0.33
Kitchens				
Restaurant		...	...	4.0
Residence		...	...	2.0
Laboratories[e]	Some	20	15	....
Meeting rooms	Very heavy	50	30	1.25
Offices				
General	Some	15	10	....
Private	None	25	15	0.25
Private	Considerable	30	25	0.25
Restaurants				
Cafeteria[e]	Considerable	12	10	....
Dining room[e]	Considerable	15	12	....
Schoolrooms[d]	None	...	...	....
Shop, retail	None	10	7½	....
Theater[d]	None	7½	5	....
Theater	Some	15	10	....
Toilets[d] (exhaust)		...	...	2.0

[a] Taken from present-day practice.
[b] This is contaminant-free air.
[c] When minimum is used, take the larger of the two.
[d] See local codes which may govern.
[e] May be governed by exhaust.
[f] May be governed by special sources of contamination or local codes.
[g] All outside air recommended to overcome explosion hazard of anesthetics. See *National Board of Fire Underwriters' Pamphlet* No. 56.

(*Continued from p. 169*)

north) are given in Table 4 for the two types of atmospheres. These are based upon observations made on cloudless days in Cleveland over a period of several summers. Since less extensive data were available for industrial atmospheres, there is more uncertainty regarding these values. In both instances, the values include an unknown amount of ground reflection, which may be expected to vary with location. It should be noted that clouds which do not obscure the sun tend to *increase* diffuse radiation values. Nearby buildings may *reduce* diffuse irradiation by partial shading.

Calculation Tables

The irradiation of a surface by the sun is the product of I_{Dn}, the direct normal radiation (see Table 4), and the cosine K of the incident angle θ. For horizontal surfaces, the cosine K equals the sine of the solar altitude. For vertical walls, K is a function of the solar altitude β and the wall solar azimuth γ, thus

$$K = \cos\theta = \cos\beta\cos\gamma \qquad (1)$$

These three angles are defined in Fig. 1. Values of K are given in Table 5 and values of β and γ are given in Table 6 for 18-deg north declination (August 1).

To compute K values for orientations other than those given in Table 5, third angle ϕ, the solar azimuth, is required. In this discussion, ϕ will be measured east from south in the morning, and west from south in the afternoon. Hence, ϕ values are equal to 90 deg minus the γ values for an east or west facing wall, except when Table 6 shows the south walls to be in the shade. In this case ϕ equals $90 + \gamma$, that is, ϕ is greater than 90 deg.

The wall azimuth ψ is the angle, measured east from south to the perpendicular to the wall for walls which have an easterly component, and west from south for those having a westerly component. For example, ψ for a wall facing northeast is 135 deg.

The wall solar azimuth γ may be found according to the following schedule:

For walls facing east of south:	For walls facing west of south:
$\gamma = \phi - \psi$ a.m.	$\gamma = \phi + \psi$ a.m.
$\gamma = \phi + \psi$ p.m.	$\gamma = \phi - \psi$ p.m.

Treat negative values of γ as if they were positive. If γ is greater than 90 deg, the wall is in the shade.

Values of K for other seasons and latitudes may be found in the literature,[7] or may be computed from data given in *Hydrographic Office Bulletin* No. 214, *Tables of Computed Altitude and Azimuth*[8] and the Ephemeris of the Sun.[9] Table 7 shows the variation of solar declination during the months ordinarily requiring cooling.

Example 1: Find the solar azimuth ϕ at 6:30 p.m. at 40-deg north latitude on August 1.

Solution: From Table 6 in the column of γ for a wall facing west ϕ for 6:00 p.m. is $90 + 14 = 104$ deg, and at 7:00 p.m. is $90 + 24 = 114$ deg. By interpolation, ϕ for 6:30 p.m. is 109 deg west of south (at 5:30 a.m. ϕ would be 109 deg east of south).

Example 2: Find K for a wall facing 18 deg east of south at 10:00 a.m. on August 1 at 50-deg north latitude.

Solution: The wall azimuth is 18 deg. The solar azimuth is 48 deg east (Table 6). The wall solar azimuth is $48 - 18$ or 30 deg. From Table 6, β is 50 deg. Then

$$K = \cos\beta\cos\gamma = \cos 50 \times \cos 30 = 0.643 \times 0.866 = 0.557.$$

Example 3: Find K for the wall in *Example 2* at 3:00 p.m.

Solution: The solar azimuth is 65 deg west. The wall solar azimuth is therefore $65 + 18 = 83$ deg. The angle β is 42 deg.

$$K = \cos 42 \times \cos 83 = 0.743 \times 0.122 = 0.091.$$

Example 4: Find the total solar irradiation for the wall for the conditions of *Example 2*.

Solution: Use clear atmosphere solar intensities. At 50-deg altitude, the direct normal radiation is 273 Btu per (hr) (sq ft). Then,

$$I_D = K \times I_{Dn} = 0.557 \times 273 = 152.0 \text{ Btu per (hr) (sq ft).}$$

Table 4 .... Values of I_{Dn}, Direct Solar Radiation Received at Normal Incidence at the Earth's Surface, and Values of I_d, Diffuse or Sky Solar Radiation, Received by Variously Oriented Surfaces

Solar Altitude β, Degrees	Btu per (hour) (square foot)											
	For Clear Atmospheres						For Industrial Atmospheres					
	Direct[a] normal radiation	Diffuse or sky radiation[b, c]					Direct[d] normal radiation	Diffuse or sky radiation[b, c]				
AM → ↓		N	E	S	W	Horiz.		N	E	S	W	Horiz.
5	67	6	11	4	4	7	34	4	11	5	3	9
10	123	11	20	8	7	14	58	8	22	9	7	18
15	166	14	27	11	10	19	80	11	28	13	9	24
20	197	15	32	13	12	23	103	13	36	17	12	31
25	218	16	35	15	13	26	121	16	43	21	16	38
30	235	17	36	17	15	28	136	18	47	24	18	44
35	248	17	36	19	16	30	148	19	50	27	21	48
40	258	18	36	21	17	31	158	20	50	30	23	52
45	266	19	35	23	18	32	165	21	49	31	25	55
50	273	19	33	25	19	33	172	22	47	34	27	58
60	283	21	28	27	21	34	181	22	41	37	30	63
70	289	22	23	29	23	35	188	22	34	41	34	69
80	292	—	—	—	—	—	195	—	—	—	—	—
90	294	—	—	—	—	—	200	—	—	—	—	—
↑ PM →		N	W	S	E	Horiz.		N	W	S	E	Horiz.

[a] Moon's[a] proposed standard for sea level, 20 mm precipitable water vapor, 300 dust particles per cu cm, 2.8 mm Hg partial pressure of ozone.
[b] For 40 deg north latitude on about August 1.
[c] Based on observations by ASHAE Laboratory at Cleveland on cloudless days during which the observed normal incidence values closely approximated the normal incidence values tabulated.
[d] Derived from recommended design sol-air temperatures[a] for New York City for a horizontal surface with absorptivity of 1.0.

Table 5 .... Values of K, the Cosine of the Incident Angle, for Variously Oriented Walls and a Horizontal Surface

Computed for 18 Deg Declination, North (August 1)

Latitude	Sun Time	Cosine K of the Incident Angle						
	AM → ↓	N	NE	E	SE	S	SW	Horiz.
30 Deg north	6 a.m. 6 p.m.	0.267	0.862	0.952	0.484			0.156
	7 5	0.144	0.752	0.919	0.548			0.367
	8 4	0.030	0.604	0.824	0.561			0.566
	9 3		0.427	0.672	0.524	0.068		0.737
	10 2		0.234	0.476	0.438	0.144		0.866
	11 1		0.039	0.246	0.310	0.192		0.951
	12			0.000	0.147	0.208	0.147	0.978
40 Deg north	5 a.m. 7 p.m.	0.406	0.934	0.914	0.358			0.009
	6 6	0.237	0.840	0.951	0.505			0.199
	7 5	0.079	0.705	0.919	0.594			0.391
	8 4		0.533	0.824	0.631	0.069		0.566
	9 3		0.337	0.673	0.614	0.196		0.713
	10 2		0.129	0.475	0.542	0.292		0.829
	11 1			0.246	0.424	0.354	0.076	0.903
	12			0.000	0.265	0.375	0.265	0.927
50 Deg north	5 a.m. 7 p.m.	0.385	0.922	0.920	0.378			0.078
	6 6	0.199	0.813	0.951	0.532			0.233
	7 5	0.010	0.656	0.918	0.643			0.399
	8 4		0.465	0.824	0.700	0.166		0.545
	9 3		0.252	0.673	0.699	0.316		0.669
	10 2		0.030	0.475	0.642	0.433		0.766
	11 1			0.247	0.532	0.505	0.183	0.829
	12			0.000	0.375	0.530	0.375	0.848
	↑ PM →	N	NW	W	SW	S	SE	Horiz.

Table 6 .... Values of the Wall Solar Azimuth, γ, for Variously Oriented Walls and Solar Altitude

Computed for 18 Deg Declination, North (August 1)

Latitude	Sun Time	Solar Altitude β Degrees	Azimuth Angle γ, Degrees					
	AM → ↓		N	NE	E	SE	S	SW
30 Deg north	6 a.m. 6 p.m.	9.0	74	29	16	61		
	7 5	21.5	81	36	9	54		
	8 4	34.5	88	43	2	47	Shade	
	9 3	47.5	Shade	51	6	39	84	
	10 2	60.0		62	17	28	73	
	11 1	72.0		83	38	7	52	Shade
	12	78.0		Shade	90	45	0	45
40 Deg north	5 a.m. 7 p.m.	0.5	66	21	24	69		
	6 6	11.5	76	31	14	59		
	7 5	23.0	85	40	5	50	Shade	
	8 4	34.5	Shade	50	5	40	85	
	9 3	45.5		61	16	29	74	
	10 2	56.0		76	31	14	59	Shade
	11 1	64.5		Shade	55	10	35	80
	12	68.0			90	45	0	45
50 Deg north	5 a.m. 7 p.m.	4.5	67	22	23	68		
	6 6	13.5	78	33	12	57		
	7 5	23.5	90	45	0	45	90	
	8 4	33.0	Shade	57	12	33	78	
	9 3	42.0		70	25	20	65	
	10 2	50.0		87	42	3	48	Shade
	11 1	56.0		Shade	64	19	26	71
	12	58.0			90	45	0	45
	↑ PM →		N	NW	W	SW	S	SE

By linear interpolation, the diffuse irradiation is

$$I_d = 25 + \tfrac{15}{60}(33 - 25) = 26.6 \text{ Btu per (hr) (sq ft)}.$$

The total solar irradiation is

$$I_t = 152.0 + 26.6 = 178.6 \text{ Btu per (hr) (sq ft)}.$$

PERIODIC HEAT FLOW THROUGH WALLS AND ROOFS

The calculation of heat flow, through a structural section of a building exposed to the weather, requires consideration of the diurnal cycles of solar irradiation and air temperature. These cycles and other factors lead to a periodic variation in the instantaneous rate of heat flow into the weather surface, and a related periodic variation in the rate of heat flow into the air-conditioned space. Because of heat capacity and other factors, these heat-flow cycles are, in general, out of time phase and unequal in amplitude.

In order to calculate the rate of heat entry into the weather surface of a building, it is necessary to know:

1. The intensity of direct solar radiation striking the surface.
2. The absorptivity (or reflectivity) of the surface for direct solar radiation.
3. The intensity of diffuse or sky solar radiation striking the surface.
4. The absorptivity (or reflectivity) of the surface for diffuse or sky solar radiation.
5. The rate at which the surface emits radiation to the sky and other surroundings.
6. The rate at which the surface absorbs the low temperature radiation emitted by the sky and other surroundings by virtue of their temperatures and radiating characteristics.
7. The temperature of the surrounding air.
8. The temperature of the outer building surface.
9. The unit convective conductance for heat transfer between the air and the building surface.

The Sol-Air Temperature

The complex interrelationship of the above factors can be considerably simplified through the use of the sol-air temperature concept. The sol-air temperature t_e is the temperature of the outdoor air, which, in the absence of all radiation exchanges, would give the same rate of heat entry into the surface as would exist with the actual combination of incident solar radiation, radiant energy exchange with the sky and other outdoor surroundings, and convective heat exchange with the outdoor air.

Table 7 .... Approximate Solar Declinations in Degrees

Date	Declination	Date	Declination
April 1	4.5	July 1	23.0
April 15	10.0	July 15	21.5
May 1	15.0	Aug. 1	18.0
May 15	19.0	Aug. 15	14.0
June 1	22.0	Sept. 1	8.5
June 15	23.5	Sept. 15	3.0

Table 8 .... Summer Design Sol-Air Temperatures Used for Tables 9 and 10

Mean Sun Time	Sol-Air Temperature t_e, Fahrenheit Degrees								
	Any Surface[b]	Horiz.	North	East		South		West	
Ratio* $\frac{\alpha}{f_o}$	0	0.225	0	0.225	0.125	0.225	0.125	0.225	0.125
12 Midnight	77	77	77	77	77	77	77	77	77
1 AM	76	76	76	76	76	76	76	76	76
2	76	76	76	76	76	76	76	76	76
3	75	75	75	75	75	75	75	75	75
4	74	74	74	74	74	74	74	74	74
5	74	74	74	75	80	74	74	74	74
6	74	76	74	110	93	74	74	74	74
7	75	91	75	123	100	75	75	75	75
8	77	106	77	126	103	82	78	77	77
9	80	119	80	125	104	93	86	80	80
10	83	129	83	117	100	102	93	83	83
11	87	137	87	108	96	110	99	89	87
12 Noon	90	142	90	92	92	114	104	96	92
1 PM	93	144	93	93	93	115	105	110	102
2	94	140	94	95	94	111	104	124	111
3	95	132	95	95	95	104	100	135	119
4	94	120	94	94	94	99	96	141	120
5	93	107	93	93	93	95	94	139	118
6	91	96	91	91	91	91	91	125	111
7	87	90	87	87	87	88	87	103	94
8	85	85	85	85	85	85	85	85	85
9	83	83	83	83	83	83	83	83	83
10	81	81	81	81	81	81	81	81	81
11	79	79	79	79	79	79	79	79	79
24 Hr Avg. t_m	83.1	100.5	83.1	93.0	88.4	89.0	86.2	93.0	88.4

* α = surface absorptivity, dimensionless, roof = 0.9; dark walls = 0.9, and light walls = 0.5. f_o = unit convective conductance = 4.0 Btu per (hr) (F deg).
[b] Values in this column are magnitudes of t_o, the outdoor air temperature.

The sol-air temperature data[4, 9, 10] as developed by Mackey and Wright for an industrial atmosphere were used as a basis for preparing Table 8 showing summer design sol-air temperatures. Sol-air temperatures may also be estimated from experimental observation of surface temperatures of walls and roofs which appear in the literature.[11, 13] Both analytical and experimental studies have been made on the problem[10] of heat flow through walls and roofs. Those concerned with a further study of the details of cooling load estimates in particular relation to periodic heat flow will find much of value and interest in the reports of experimental studies of these problems.[10, 11, 12, 13, 14, 15] The reader may also refer to the Cooling Load chapter of THE GUIDE 1952 for the theory of heat flow through walls and roofs.

PRACTICAL TABLES FOR CALCULATING SOLAR HEAT GAIN THROUGH WALLS AND ROOFS

The analytical[10] method reported by Mackey and Wright was used by Stewart[16] to obtain temperature differentials based on Table 8 and shown in Tables 9 and 10. These analytical procedures, as well as those using Tables 9 and 10, presented here, yield generally higher rates of heat gain than reported for Pittsburgh in early ASHAE experimental studies. Current authoritative opinion indicates a preference for analytical calculations. Thermal and physical properties of materials used in these tables are given in a paper.[16] The rate of heat flow is obtained by multiplying the overall heat transmission coefficient of the structure by the equivalent temperature differential obtained from the tables.

Tables 9 and 10 were developed by using an outside surface conductance of 4.0 and an inside film conductance of 1.65 Btu (hr) (sq ft) (F deg). A reduction was made in the temperature differentials for roofs amounting to some 20 percent of solar radiation as explained by Stewart.[16] This was to compensate for several factors, one of which is the radiant heat lost to the sky which is not included in the Mackey and Wright method. Experimental work by Parmelee[17] and previous work by Brunt[18] give data showing the magnitude of this radiant heat loss from a roof or wall to the sky. Temperature differentials for roofs probably would be reduced below those shown in Table 9 whenever the radiant heat lost to the sky is included in calculation of sol-air temperature. The *temperature differentials* for roofs were based on an inside surface conductance of 1.65 because the charts prepared by Mackey and Wright[10] used this value, and it was not practicable to repeat their work using a different film coefficient. An examination of the values given in their paper indicates that the *temperature differential* would be changed very little even if a value 1.20 were used instead of 1.65. To obtain the heat-flow rates through roofs, more accurate

Table 9.... Total Equivalent Temperature Differentials for Calculating Heat Gain Through Sunlit and Shaded Roofs

Description of Roof Construction[a]	Sun Time								
	A.M.			P.M.					
	8	10	12	2	4	6	8	10	12
Light Construction Roofs—Exposed to Sun									
1" Wood[b] or 1" Wood[b] + 1" or 2" insulation	12	38	54	62	50	26	10	4	0
Medium Construction Roofs—Exposed to Sun									
2" Concrete or 2" Concrete + 1" or 2" insulation or 2" Wood[b]	6	30	48	58	50	32	14	6	2
2" Gypsum or 2" Gypsum + 1" insulation 1" Wood[b] or 2" Wood[b] or 2" Concrete or 2" Gypsum + 4" rock wool in furred ceiling	0	20	40	52	54	42	20	10	6
4" Concrete or 4" Concrete with 2" insulation	0	20	38	50	52	40	22	12	6
Heavy Construction Roofs—Exposed to Sun									
6" Concrete	4	6	24	38	46	44	32	18	12
6" Concrete + 2" insulation	6	6	20	34	42	44	34	20	14
Roofs Covered with Water—Exposed to Sun									
Light construction roof with 1" water	0	4	16	22	18	14	10	2	0
Heavy construction roof with 1" water	−2	−2	−4	10	14	16	14	10	6
Any roof with 6" water	−2	0	0	6	10	10	8	4	0
Roofs with Roof Sprays—Exposed to Sun									
Light construction	0	4	12	18	16	14	10	2	0
Heavy construction	−2	−2	2	8	12	14	12	10	6
Roofs in Shade									
Light construction	−4	0	6	12	14	12	8	2	0
Medium construction	−4	−2	2	8	12	12	10	6	2
Heavy construction	−2	−2	0	4	8	10	10	8	4

[a] Includes ⅜ in. felt roofing with or without slag. May also be used for shingle roof.
[b] Nominal thickness of the wood.

Notes for Table 9

Explanation: {Total heat transmission from solar radiation and temperature difference between outdoor and room air. Btu per (hr) (sq ft) of roof area} = {Equivalent temperature differential from above table} × {Heat transmission coefficient for summer Btu per (hr) (sq ft) (F deg)}

1. *Source.* Calculated by Mackey and Wright method (see reference list) and adjusted after studying ASHAE original test data. Estimated for about August 1 in 40 deg north latitude. (For sol-air temperatures used in calculations see Table 8.) For typical design day where the maximum outdoor temperature is 95 F and minimum temperature at night is approximately 75 F (daily range of temperature, 20 F) mean 24 hr temperature 84 F for a room temperature of 80 F. All roofs have been assumed a dark color which absorbs 90 percent of solar radiation, and reflects only 10 percent.

2. *Application.* These values may be used for all normal air conditioning estimates; usually without correction, in latitude 0 deg to 50 deg north or south when the load is calculated for the hottest weather. Note 5 explains how to adjust the temperature differential for other room and outdoor temperatures.

3. *Peaked Roofs.* If the roof is peaked and the heat gain is primarily due to solar radiation, use for the area of the roof, the area projected on a horizontal plane.

4. *Attics.* If the ceiling is insulated and if a fan is used in the attic for positive ventilation, the total temperature differential for a roof exposed to the sun may be decreased 25 percent.

5. *Corrections. For temperature difference when outdoor maximum design temperature minus room is different from 15 deg.* If the outdoor design temperature minus room temperature is different from the base of 15 deg, correct as follows: When the difference is greater (or less) than 15 deg add the excess to (or subtract the deficiency from) the above differentials.

For outdoor daily range of temperature other than 20 deg. If the daily range of temperature is less than 20 deg, add 1 deg for every 2 deg lower daily range; if the daily range is greater than 20 deg, subtract 1 deg for every 2 deg higher daily range. For example, the daily range in Miami, Florida is 12 deg or 8 deg less than 20 deg, therefore, the correction is +4 deg at all hours of the day.

Notes for Table 9.... Total Equivalent Temperature Differentials for Calculating Heat Gain Through Sunlit and Shaded Roofs (Concluded)

Light Colors. Credit should not be taken for light colored roofs except where the permanence of the light color is established by experience, as in rural areas or where there is little smoke. When the exterior surface of roof exposed to the sun is a light color, such as white or aluminum (which absorb approximately 50 percent and reflect 50 percent of the solar radiation) add to the temperature differential for roof in shade 55 percent of the difference between the roof in sun and roof in shade. When the roof exposed to the sun is a medium color such as light grey, blue or green, or bright red, add 80 percent of this difference.

For solar transmission in latitudes other than 40 deg north, and in other months. The table values of temperature differentials will be approximately correct for a roof in the following months:

North Latitude		South Latitude	
Latitude (deg)	Months	Latitude (deg)	Months
0	All months	0	All months
10	All months	10	All months
20	All months except Nov, Dec, Jan	20	All months except May, June, July
30	Mar, Apr, May, June, July, Aug, Sept	30	Sept, Oct, Nov, Dec, Jan, Feb, Mar
40	April, May, June, July, Aug	40	Oct, Nov, Dec, Jan, Feb
50	May, June, July	50	Nov, Dec, Jan

For other months, the total temperature differential (t_d) may be approximated by the use of the following formula:

$$t_d = t_s + \frac{I_s}{I_y}(t_e - t_s)$$

where

t_s = temperature differential for the same roof in shade for desired time of day; obtained from Table 9.

I_y = maximum solar transmission through glass, Btu per (hr) (sq ft) for flat skylight in Aug., 40 deg north latitude (*Note:* this is maximum value irrespective of time).

I_s = same as I_y except use the maximum value for flat skylight, for month, and latitude desired for t_e.

t_e = temperature differential for particular roof exposed to sun for the desired time of day from Table 9.

(*Note that this makes adjustment only for solar radiation and that there may be additional correction for outdoor temperature.*)

values will be obtained if the *overall heat transmission coefficient is calculated using* 1.2 as the inside film conductance for summer.

The roof coefficients of transmission for summer shown in Table 11 are based on surface conductances f_{so} of 4.0 for an outside roof surface and 1.20 for an inside ceiling surface. The outside conductance 4.0 is used for summer because it corresponds to a wind velocity of approximately 7.5 mph averaged for rough and smooth surfaces, and is more representative of summer wind velocities. Also, the lower wind velocity should be used in order to be on the safe side in determining the sol-air temperature. The inside conductance 1.20 is used because the convective portion of the film conductance for downward heat flow from a horizontal surface is appreciably less than the winter conductance for heat flowing upward.

Since there is little difference in *wall transmission coefficients* for summer, based on the conductances of 4.0 and 1.65, and the winter coefficients, based on 6.0 and 1.65, it is recommended that the overall coefficient *U, for walls*, be taken directly from the tables in Chapter 9 in which they are based on an outside film conductance of 6.0, corresponding to a 15 mph wind velocity.

Advantages of Equivalent Temperature Differential Method

The advantages of the *equivalent* temperature differential method of determining the total heat transmission are given in following paragraphs, and are apparent from *Examples 5* to 7.

1. The total sensible heat flow is obtained by multiplying the overall heat transmission coefficient U, and the equivalent temperature differential indicated in Tables 9 and 10.

2. The temperature differentials listed for a few representative types of construction may be used on all classes of walls and roofs, even though the overall heat transmission coefficient is different, provided the structure has thermal and physical properties similar to one of those listed in Tables 9 and 10.

3. Adjustments can be made, according to instructions given in the footnotes, for room and outdoor conditions different from those on which the tables are based.

Examples of Use of Equivalent Temperature Tables

Example 5: Given: A roof is constructed of 6 in. of stone concrete with 2 in. of insulating board and tar felt roofing ⅜ in. thick, and is exposed to the sun. The location is the central part of the United States. Find the rate of heat flow into building at 2:00 p.m. during July for an outdoor design temperature 95 F, and an indoor temperature 80 F.

Solution: From Table 9 in 2 p.m. column for 6-in. concrete plus 2-in. insulation find the total equivalent temperature differential 34 deg. The overall heat transmission coefficient for summer is taken from Table 11 and is found to be 0.13. The heat-flow rate equals 34 × 0.13 = 4.42 Btu per (hr) (sq ft).

Example 6: For the conditions of *Example 5*, find the rate of heat flow into building at 2:00 p.m. during July for design temperatures of 105 F (outdoor) and 78 F (indoor). Daily range of temperature 30 deg, i.e., outdoor temperature minimum of 75 F which occurs at 4:00 or 5:00 a.m.; this being 30 deg less than the maximum.

Solution: Make correction in equivalent temperature differential in accordance with Note 5 in Table 9 as follows:

The correction for 27-deg design temperature difference is (27 − 15) = +12.

The correction for 30-deg daily range is $\left(-\frac{30-20}{2}\right) = -5$.

Net total correction is +12 − 5 = +7.

The heat-flow rate at 2:00 p.m. is (34 + 7) × 0.13 = 5.32 Btu per (hr) (sq ft).

A method of determining *heat-flow rates*, when structure is not given in Tables 9 or 10, is illustrated in *Example 7*.

Example 7: A 4-in. stone concrete roof covered with an average depth of 4 in. of cinder concrete ($k = 4.9$), on which is placed a ⅜-in. thick felt roof with ½-in. pitch and slag

Table 10. . . . Total Equivalent Temperature Differentials for Calculating Heat Gain Through Sunlit and Shaded Walls

North Latitude Wall Facing	Sun Time: A.M. 8		A.M. 10		A.M. 12		P.M. 2		P.M. 4		P.M. 6		P.M. 8		P.M. 10		P.M. 12		South Latitude Wall Facing
	Exterior color of wall—D = dark, L = light																		
	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	
	Frame																		
NE	22	10	24	12	14	10	12	10	14	14	14	14	10	10	6	4	2	2	SE
E	30	14	36	18	32	16	12	12	14	14	14	14	10	10	6	6	2	2	E
SE	13	6	26	16	28	18	24	16	16	14	14	14	10	10	6	4	2	2	NE
S	−4	−4	4	0	22	12	30	20	26	20	16	14	10	10	6	6	2	2	N
SW	−4	−4	0	−2	6	4	26	22	40	28	42	28	24	20	6	4	2	2	NW
W	−4	−4	0	0	6	6	20	12	40	28	48	34	22	22	8	8	2	2	W
NW	−4	−4	0	−2	6	4	12	10	24	20	40	26	34	24	6	4	2	2	SW
N (Shade)	−4	−4	−2	−2	4	4	10	10	14	14	12	12	8	8	4	4	0	0	S (Shade)
	4 in. Brick or Stone Veneer + Frame																		
NE	−2	−4	24	12	20	10	10	6	12	10	14	14	12	12	10	10	6	4	SE
E	2	0	30	14	31	17	14	14	12	12	14	14	12	12	10	8	6	6	E
SE	2	−2	20	10	28	16	26	16	18	14	14	14	12	12	10	8	6	6	NE
S	−4	−4	−2	−2	12	6	24	16	26	18	20	16	12	12	8	8	4	4	N
SW	0	−2	0	−2	2	2	12	8	32	22	36	26	34	24	10	8	6	6	NW
W	0	−2	0	0	4	2	10	8	26	18	40	28	42	28	16	14	6	6	W
NW	−4	−4	−2	−2	2	2	8	6	12	12	30	22	34	24	12	10	6	6	SW
N (Shade)	−4	−4	−2	−2	0	0	6	6	10	10	12	12	12	12	8	8	4	4	S (Shade)
	8 in. Hollow Tile or 8 in. Cinder Block																		
NE	0	0	0	0	20	10	16	10	10	6	12	10	14	12	12	10	8	8	SE
E	4	2	12	4	24	12	26	14	20	12	12	10	14	12	14	10	10	8	E
SE	2	0	2	0	16	8	20	12	20	14	14	12	14	12	12	10	8	6	NE
S	0	0	0	0	2	0	12	6	24	*14	26	16	20	14	12	10	8	6	N
SW	2	0	2	0	2	0	6	4	12	10	26	18	30	20	26	18	8	6	NW
W	4	2	4	2	4	2	6	4	10	8	18	14	30	22	32	22	18	14	W
NW	0	0	0	0	2	0	4	2	8	6	12	10	22	18	30	22	10	8	SW
N (Shade)	−2	−2	−2	−2	−2	−2	0	0	6	6	10	10	10	10	10	10	6	6	S (Shade)
	8 in. Brick or 12 in. Hollow Tile or 12 in. Cinder Block																		
NE	2	2	2	2	10	2	16	8	14	8	10	6	10	8	10	10	10	8	SE
E	8	6	8	6	14	8	18	10	18	10	14	8	14	10	14	10	12	10	E
SE	8	4	6	4	6	4	14	10	18	12	16	12	12	10	12	10	12	10	NE
S	4	2	4	2	4	2	4	2	10	6	16	10	16	12	12	10	10	8	N
SW	8	4	6	4	6	4	8	4	10	6	12	8	20	12	24	16	20	14	NW
W	8	4	6	4	6	6	8	6	10	6	14	8	20	16	24	16	24	16	W
NW	2	2	2	2	2	2	4	2	6	4	8	6	10	8	16	14	18	14	SW
N (Shade)	0	0	0	0	0	0	0	0	2	2	6	6	8	8	8	8	6	6	S (Shade)
	12 in. Brick																		
NE	8	6	8	6	8	4	8	4	10	4	12	6	12	6	10	6	10	6	SE
E	12	8	12	8	12	8	10	6	12	8	14	10	14	10	14	8	14	8	E
SE	10	6	10	6	10	6	10	6	10	6	12	8	14	10	14	10	12	8	NE
S	8	6	8	6	6	4	6	4	6	4	8	4	10	6	12	8	12	8	N
SW	10	6	10	6	10	6	10	6	10	6	10	8	10	8	12	8	14	10	NW
W	12	8	12	8	12	8	10	6	10	6	10	6	10	6	12	8	16	10	W
NW	8	6	8	6	8	4	8	4	8	4	8	4	8	6	10	6	10	6	SW
N (Shade)	4	4	2	2	2	2	2	2	2	2	2	2	2	2	4	4	6	6	S (Shade)

Table 10 . . . Total Equivalent Temperature Differentials for Calculating Heat Gain Through Sunlit and Shaded Walls (*Concluded*)

North Latitude Wall Facing	Sun Time: A.M. 8		A.M. 10		A.M. 12		P.M. 2		P.M. 4		P.M. 6		P.M. 8		P.M. 10		P.M. 12		South Latitude Wall Facing
	Exterior color of wall—D = dark, L = light																		
	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	
	8 in. Concrete or Stone or 6 in. or 8 in. Concrete Block																		
NE	4	2	4	0	16	8	14	8	10	6	12	8	12	10	10	8	8	6	SE
E	6	4	14	8	24	12	24	12	18	10	14	10	14	10	12	10	10	8	E
SE	6	2	6	4	16	10	18	12	18	12	14	12	12	10	12	10	10	8	N
S	2	1	2	1	4	1	12	6	16	12	18	12	14	12	10	8	8	6	NE
SW	6	2	4	2	6	2	8	4	14	10	22	16	24	16	22	16	10	8	NW
W	6	4	6	4	6	4	8	6	12	8	20	14	26	18	26	18	14	10	W
NW	4	2	4	0	4	2	4	4	6	6	12	10	20	14	22	16	8	6	SW
N (Shade)	0	0	0	0	0	0	2	2	4	4	6	6	8	8	6	6	4	4	S (Shade)
	12 in. Concrete or Stone																		
NE	6	4	6	2	6	2	14	8	14	8	10	8	10	8	12	10	10	8	SE
E	10	6	8	6	10	6	18	10	18	12	16	10	12	10	14	10	14	10	E
SE	8	4	8	4	6	4	14	8	16	10	16	10	14	10	12	10	12	10	NE
S	6	4	4	2	4	2	4	2	10	6	14	10	16	12	14	10	10	8	N
SW	8	4	8	4	6	4	6	4	8	6	10	8	18	14	20	14	18	12	NW
W	10	6	8	6	8	6	10	6	10	6	12	8	16	10	24	14	22	14	W
NW	6	4	6	2	6	2	6	4	6	4	8	6	10	8	18	12	20	14	SW
N (Shade)	0	0	0	0	0	0	0	0	2	2	4	4	6	6	8	8	6	6	S (Shade)

Notes for Table 10

Explanation: {Total heat transmission from solar radiation and temperature difference between outside and room air, Btu per (hr) (sq ft wall area)} = {Equivalent temperature differential from above table} × {Heat transmission coefficient for wall, Btu per (hr) (sq ft) (F deg)}

1. *Source.* Same as Table 9. A north wall has been assumed to be a wall in the shade; this is practically true. Dark colors on exterior surface of walls have been assumed to absorb 90 percent of solar radiation and reflect 10 percent; white colors absorb 50 percent and reflect 50 percent. This includes some allowance for dust and dirt since clean, fresh white paint normally absorbs only 40 percent of solar radiation.

2. *Application.* These values may be used for all normal air conditioning estimates, usually without corrections, when the load is calculated for the hottest weather. Correction for latitude (Note 3) is necessary only where extreme accuracy is required. There may be jobs where the indoor room temperature is considerably above or below 80 F, or where the outdoor design temperature is considerably above 95 F, in which case it may be desirable to make correction to the temperature differentials shown. The solar intensity on all walls other than east and west varies considerably with time of year.

3. *Corrections. Outdoor minus room temperature.* If the outdoor maximum design temperature minus room temperature is different from the base of 15 deg, correct as follows: When the difference is greater (or less) than 15 deg, add the excess to (or subtract the deficiency from) the above differentials.

Outdoor daily range temperature. If the daily range of temperature is less than 20 deg, add 1 deg to every 2 deg lower daily range; if the daily range is greater than 20 deg, subtract 1 deg for every 2 deg higher daily range. For example, the daily range in Miami, Florida is 12 deg, or 8 deg less than 20 deg; therefore, the correction is +4 deg.

Color of exterior surface of wall. Use temperature differentials for light walls only where the permanence of the light wall is established by experience. For cream colors use the values for light walls. For medium colors interpolate half way between the dark and light values. Medium colors are medium blue, medium green, bright red, light brown, unpainted wood, natural color concrete, etc. Dark blue, red, brown, green, etc., are considered dark colors.

For latitudes other than 40 deg north; and in other months. These table values will be approximately correct for the east or west wall in any latitude (0 deg to 80 deg North or South) during the hottest weather. In the lower latitudes when the maximum solar altitude is approximately 80 deg to 90 deg (the maximum occurs at noon) the temperature differential for either a south or north wall will be approximately the same as a north, or shade wall. The temperature differential (t_s) for any wall facing, and for any latitude for any month may be approximated as follows:

$$t_s = t_g + \frac{I_2}{I_1} \times (t_w - t_g)$$

where

t_g = temperature differential for the same wall in shade for desired time of day; obtained from Table 10.

I_1 = maximum solar radiation intensity transmitted through glass, Btu per (hr) (sq ft) for particular wall facing, in July, 40 deg north latitude (*note:* this is maximum value irrespective of time).

I_2 = same as I_1 except use the maximum value for wall facing, for month, and latitude desired for t_g.

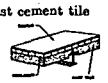
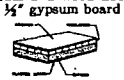
t_w = temperature differential for particular wall facing, for the desired time of day from above table.

(*Note that this makes adjustment only for solar radiation, and that there may be additional correction for outdoor temperature.*)

4. *For insulated walls* use same temperature differentials as used for uninsulated walls.

Table 11 .... Summer Coefficients of Heat Transmission *U* of Flat Roofs Covered with Built-Up Roofing[a]

Btu per (hour) (square foot) (F deg difference between the air on the two sides)

Type of Roof Deck (Ceiling not shown)	Thickness of Roof Deck (Inches)	Insulation on Top of Deck (Covered With Built Up Roofing)									
		No Ceiling—Underside of Roof Exposed					Furred Ceiling with Air Space, Metal Lath and Plaster				
		No Insulation	Insulating board[d] thickness, in.				No Insulation	Insulating board[d] thickness, in.			
			½	1	1½	2		½	1	1½	2
Flat metal roof deck 	4 Ply Felt Roof	0.73	0.35	0.23	0.17	0.13	0.40	0.25	0.18	0.14	0.12
	Ditto + ½ in. Slag	0.54	0.30	0.20	0.16	0.13	0.34	0.22	0.16	0.13	0.11
Precast cement tile 	4 Ply Felt Roof 1⅝	0.67	0.33	0.22	0.17	0.13	0.38	0.24	0.18	0.14	0.12
	Ditto + ½ in. Slag 1⅝	0.50	0.28	0.20	0.15	0.12	0.32	0.21	0.17	0.13	0.11
Concrete 	4 Ply 2	0.65	0.33	0.22	0.16	0.13	0.37	0.24	0.18	0.14	0.12
	Felt 4	0.59	0.31	0.21	0.16	0.13	0.36	0.23	0.17	0.13	0.12
	Roof 6	0.54	0.30	0.20	0.16	0.13	0.33	0.22	0.17	0.13	0.11
	Ditto 2	0.49	0.28	0.20	0.15	0.12	0.31	0.21	0.16	0.13	0.11
	4	0.46	0.27	0.19	0.15	0.12	0.30	0.21	0.16	0.13	0.11
	+ ½ in. Slag 6	0.42	0.26	0.19	0.14	0.12	0.29	0.20	0.16	0.13	0.10
Gypsum and wood fiber[b] on ½" gypsum board 	4 Ply Felt 2½	0.34	0.23	0.17	0.13	0.12	0.25	0.18	0.14	0.12	0.097
	Roof 3½	0.28	0.20	0.15	0.12	0.11	0.21	0.16	0.13	0.11	0.094
	Ditto 2½	0.29	0.20	0.16	0.13	0.11	0.22	0.16	0.13	0.11	0.093
	+ ½ in. Slag 3½	0.25	0.18	0.14	0.12	0.10	0.19	0.15	0.13	0.10	0.090
Wood[c] 	4 Ply 1	0.43	0.26	0.19	0.15	0.12	0.29	0.20	0.15	0.13	0.11
	Felt 1½	0.33	0.22	0.17	0.13	0.11	0.24	0.18	0.14	0.12	0.097
	Roof 2	0.29	0.20	0.16	0.13	0.11	0.22	0.16	0.13	0.11	0.094
	3	0.22	0.16	0.13	0.11	0.09	0.17	0.13	0.12	0.10	0.085
	1	0.35	0.23	0.17	0.14	0.11	0.25	0.18	0.14	0.12	0.10
	Ditto 1½	0.29	0.20	0.15	0.12	0.10	0.21	0.17	0.13	0.11	0.093
	+ ½ in. Slag 2	0.26	0.19	0.14	0.12	0.10	0.20	0.15	0.13	0.10	0.090
	3	0.20	0.15	0.12	0.10	0.09	0.16	0.13	0.11	0.09	0.081

[a] The summer coefficients are considered temporary, and have been calculated with an outdoor wind velocity of 8 mph. For summer an inside surface conductance of 1.2 has been used instead of the regular 1.65 value. In all of these roofs a 4 ply felt roof has been assumed ⅜ in. thick, thermal conductivity = 1.33. Pitch and slag have been assumed as an additional thickness of ½ in. which has been assigned thermal conductivity = 1.0. In both cases thermal conductivity refers to one inch thickness.

[b] 87½ percent gypsum, 12½ percent wood fiber. Thickness indicated includes ½ in. gypsum board. This is a poured roof.

[c] Nominal thickness of wood is specified, but actual thickness was used in calculations.

[d] If corkboard insulation is used, the coefficient *U* may be decreased 10 percent.

surface, is exposed to the sun. The location is the central part of the United States. Design temperatures are: outdoor 95 F; daily range 20 deg; indoor temperature 80 F. Find the heat-flow rate at 2:00 p.m. for a day in July.

Solution: For the purpose of selecting the *equivalent temperature differential*, this construction is assumed to be equal approximately to an uninsulated 6-in. concrete roof, for which the equivalent temperature is found to be 38 deg in the 2:00 p.m. column of Table 9. Calculate the overall heat transmission coefficient *U* (see Equation 3 of Chapter 9) of the roof as follows:

$$U = \frac{1}{\frac{1}{1.2} + \frac{4}{12} + \frac{4}{4.9} + \frac{0.375}{1.33} + \frac{0.50}{1.00} + \frac{1}{4.0}} = 0.33$$

The heat-flow rate is then 38 × 0.33 equals 12.5 Btu per (hr) (sq ft).

TABLES FOR CALCULATING SOLAR HEAT GAIN THROUGH GLASS AREAS

Basic Principles

In order to set forth the principles involved in calculating heat flow through glass areas, the general instantaneous heat balance relation will be presented. It will be shown schematically in Fig. 2. The net heat gain for the indoor space is the result of several contributing factors.

The following observations concerning the behavior of

glass with respect to radiant energy will lead to a better understanding of the heat balance relation.

1. Glass transmits, in varying degrees, radiation having wavelengths between 0.29 and 4.75 microns. The percentage of each wavelength transmitted is dependent upon the chemical and physical characteristics of the glass, and upon the angle of incidence of the radiant energy. Of the energy not transmitted, part is absorbed and part reflected.

2. Glass is opaque to radiant energy emitted from sources below 450 F.

3. Because of the above principles, it is convenient to group radiant energy into two classifications, solar radiant energy and low temperature radiant energy.

The complete heat balance for a glass section can be expressed for a unit time interval as follows:

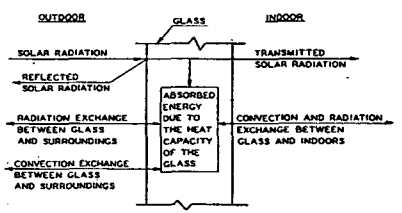


Fig. 2 .... Instantaneous Heat Balance for a Glass or Glass Block Section

$$\begin{bmatrix}\text{Total heat flow,}\\ \text{through glass section}\end{bmatrix} = \begin{bmatrix}\text{Transmitted,}\\ \text{solar radiation}\end{bmatrix} \pm \begin{bmatrix}\text{Heat flow by convective}\\ \text{and radiative exchanges at}\\ \text{the indoor surface}\end{bmatrix} \quad (2a)$$

The second term of the right side of Equation 2a can also be expressed by a heat balance equation as follows:

$$\begin{bmatrix}\text{Heat flow by convective}\\ \text{and radiative exchanges}\\ \text{at the indoor surface}\end{bmatrix} = \begin{bmatrix}\text{Absorbed}\\ \text{solar}\\ \text{radiation}\end{bmatrix} \pm \begin{bmatrix}\text{Radiative exchanges be}\\ \text{-tween outer surface of glass}\\ \text{and outdoor surroundings}\end{bmatrix} \quad (2b)$$

$$\pm \begin{bmatrix}\text{Convective exchanges}\\ \text{between outer surface of}\\ \text{glass and outdoor air}\end{bmatrix} \pm \begin{bmatrix}\text{Heat storage}\\ \text{within the}\\ \text{glass section}\end{bmatrix}$$

Equations 2a and 2b can be combined and expressed in symbolic terms by Equation 2c. Tabular values of the two bracketed terms of Equation 2a are presented later in this section for various types of glass for specific design conditions.

$$(q/A) = [\tau_D I_D + \tau_d I_d] + [\alpha_D I_D + \alpha_d I_d + \epsilon_{go} R_s - \epsilon_{go} R_{go} - f_{co}(t_{go} - t_o) - S] \quad (2c)$$

where

(q/A) = instantaneous rate of heat flow, Btu per (hour) (square foot).

τ_D, τ_d = transmittance of glass for direct and diffuse solar radiation, respectively.

I_D, I_d = incident direct and diffuse solar radiation, respectively, Btu per (hour) (square foot).

α_D, α_d = absorptance of glass for direct and diffuse solar radiation, respectively.

ϵ_{go} = emissivity of glass at temperature t_{go}.

R_s = low temperature radiant energy falling on glass from outdoor surroundings, Btu per (hour) (square foot).

R_{go} = low temperature radiant energy emitted by a surface with emissivity equal to 1.0 at temperature t_{go}.

f_{co} = outdoor convective conductance, Btu per (hour) (square foot) (Fahrenheit degree).

t_{go} = temperature of outdoor surface of glass, Fahrenheit.

t_o = temperature of outdoor air, Fahrenheit.

S = rate at which glass stores energy, Btu per (hour) (square foot).

Transmissivity and absorptivity vary with both wavelength of the incident radiation and incident angle. Normal incidence transmittance values for some commonly used types and combination are given in Table 14. Some variation in these values can be expected in practice due to variations in manufacture and in solar energy distribution. However, a change in transmissivity causes a compensating change in absorptivity. Generally, the *total heat flow* is not greatly altered. Transmittance data for a number of types of glass and various patterns of 8-in. glass block are given in ASHAE research papers.[19, 20, 21, 22, 23, 24]

As stated earlier in this chapter, present data as to the value of R_s are inadequate, so for the present it is suggested that f_{co} be increased to include radiation, and the term $\epsilon_{go}R_s - \epsilon_{go}R_{go}$ be disregarded. It is not practicable to give values of S in this chapter. However, for ordinary glass, the value of S is small.

Fig. 3 is a graphical solution, for single glass, of Equation 2b. Only absorbed solar radiation is considered, although low temperature radiation exchange and heat storage can be added algebraically to $\alpha_i I_i$ if such data are available. The small thermal resistance of the glass has been neglected. The heat-flow rates are for 80 F indoor temperature, an in-

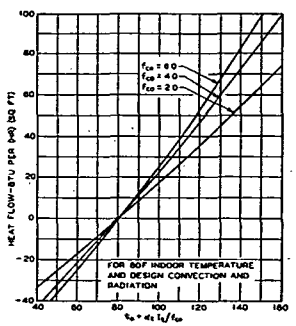


Fig. 3 .... Convection and Radiation Heat Flow for Vertical Single Glass

Table 12 .... Instantaneous Rates of Heat Gain Due to Transmitted Direct and Diffuse or Sky Solar Radiation by a Single Sheet of Unshaded Common Window Glass

For Clear Atmospheres and 18 Deg Declination, North (August 1)

Note: For total instantaneous heat gain, add these values to the Table 13 values

Latitude	Sun Time	Instantaneous Heat Gain in Btu per (hr) (sq ft)								
	AM → ↓	N	NE	E	SE	S	SW	W	NW	Horiz.
30 Deg north	6 a.m. 6 p.m.	25	98	108	52	*5*	*5*	*5*	*5*	17
	7 *5*	23	155	190	110	*10*	*10*	*10*	*10*	71
	8 4	16	148	205	136	*14*	*13*	*13*	*13*	137
	9 3	*16*[a]	106	180	136	21	*15*	*15*	*15*	195
	10 2	*17*	54	128	116	34	*17*	*16*	*16*	241
	11 1	*18*	20	59	78	45	*19*	*18*	*18*	267
	12	*18*	*19*	*19*	35	49	35	*19*	*19*	276
40 Deg north	5 a.m. 7 p.m.	3	7	6	2	*0*	*0*	*0*	0	1
	6 6	26	116	131	67	7	*6*	*6*	6	25
	7 5	16	149	195	124	*11*	*10*	*10*	10	77
	8 4	*14*	129	205	156	18	*13*	*13*	12	137
	9 3	*15*	79	180	162	42	*14*	*14*	14	188
	10 2	*16*	31	127	148	69	*16*	*16*	16	229
	11 1	*17*	*18*	58	113	90	23	*17*	17	252
	12	*17*	*17*	19	64	98	64	*19*	17	259
50 Deg north	5 a.m. 7 p.m.	20	54	54	20	*3*	*3*	*3*	*3*	6
	6 6	25	128	149	81	*8*	7	7	7	34
	7 5	*12*	139	197	136	*12*	*10*	*10*	*10*	80
	8 4	*13*	107	202	171	32	*12*	*12*	*12*	129
	9 3	*14*	54	176	183	72	*14*	*14*	*14*	173
	10 2	*15*	18	124	174	110	*16*	*15*	*15*	206
	11 1	*16*	*16*	57	143	136	42	*16*	*16*	227
	12	*16*	*16*	*18*	96	144	96	*18*	*16*	234
	↑ PM →	N	NW	W	SW	S	SE	E	NE	Horiz.

[a] Italics indicate shaded glass.

Table 13 .... Instantaneous Rates of Heat Gain by Convection and Radiation from a Single Sheet of Unshaded Common Window Glass

For Clear Atmospheres and 18 Deg Declination, North (August 1)

For 30, 40, 50 Deg North Latitude

For 80 F Indoor Temperature

Note: For total instantaneous heat gain, add these values to the Table 12 values

Sun Time	Dry-Bulb F Deg	Instantaneous Heat Gain in Btu per (hr) (sq ft)								
		N	NE	E	SE	S	SW	W	NW	Horiz.
5 a.m.	74	−6	−6	−6	−6	−6	−6	−6	−6	−6
6	74	−5	−4	−4	−5	−5	−6	−6	−6	−5
7	75	−5	−2	−2	−3	−5	−5	−5	−5	−3
8	77	−3	0	1	0	−2	−3	−3	−3	0
9	80	0	2	4	3	1	0	0	0	3
10	83	3	4	6	6	5	3	3	3	8
11	87	8	8	10	11	10	9	8	8	13
12	90	12	12	12	13	14	13	12	12	16
1 p.m.	93	15	15	15	16	17	17	17	15	20
2	94	16	16	16	16	18	19	19	17	21
3	95	17	17	17	17	19	21	21	19	21
4	94	16	16	16	16	17	20	20	19	19
5	93	15	15	15	15	15	18	19	18	17
6	91	13	13	13	13	13	14	15	15	13
7	87	8	8	8	8	8	8	8	8	8
8	85	6	6	6	6	6	6	6	6	6
9	83	3	3	3	3	3	3	3	3	3

Table 14 .... Application Factors to Apply to Tables 12, 13 and 15 to Obtain Instantaneous Rates of Heat Gain for Various Types of Single Flat Glass and Combinations of Two Sheets of Flat Glass Spaced at ¼ In.

Glass[a]	Normal Incidence Transmittance	Factor to Apply to Table 12	Factor to Apply to Table 13
Single common window	0.87	1.00	$1.0(X)^c + 0.0(Y)^d$
Single regular plate	0.77	0.87	$1.0(X) + 0.25(Y)$
Single heat absorbing plate	0.41	0.46[b]	$1.0(X) + 1.00(Y)$
Double common window	0.76	0.85	$0.6(X) + 0.10(Y)$
Double regular plate	0.60	0.66[b]	$0.6(X) + 0.55(Y)$
Heat absorbing plate outdoors, Regular plate indoors	0.35	0.37[b]	$0.6(X) + 0.75(Y)$

[a] Common window glass ⅛ in. thick. Plate glass ¼ in. thick.

[b] For better precision, increase factors 10 percent when glass is in the shade.

[c] X values are Table 13 values.

[d] Y values are Table 15 values.

side surface conductance for convection f_{ci} as given by Equation 3, and an equivalent surface conductance for radiation f_{ri} as given by Equation 4. Inside surfaces *seen* by the glass are assumed to radiate as a black body at room air temperature.

$$f_{ci} = 0.27\ (t_{gi} - t_i)^{0.25} \qquad (3)$$

$$f_{ri} = 0.162\left[\left(\frac{t_{gi} + 460}{100}\right)^4 - \left(\frac{t_i + 460}{100}\right)^4\right] \Big/ (t_{gi} - t_i) \qquad (4)$$

where

t_{gi} = temperature of inside surface of glass, Fahrenheit.

t_i = temperature of indoor air, Fahrenheit.

A more complete treatment of the problem is given in an ASHAE research paper.[22]

Design Tables for Flat Glass

Tables 12 and 13 give design values of instantaneous rates of heat gain for *single unshaded common window glass* for a solar declination of 18 deg. This corresponds to a nominal August 1 day. The tables are based upon the solar intensity values for a clear atmosphere as given in Table 4. Table 12 values represent the first bracketed term of Equation 2a; therefore, the values are dependent only upon values of I and τ. Table 13 values represent the second term of Equation 2a, and are based upon a 80 F indoor temperature and a dry-bulb temperature cycle, with a 95 F maximum as tabulated. *The total heat gain is the sum* of the Table 12 and Table 13 values. In preparing Table 13, convection and radiation heat exchange were combined, and a combined surface conductance of 4.0 used. Corrections to be applied for other design temperatures are given in Table 24 in a later section Effect of Deviation from Design Conditions.

Tables 12 and 13 may be used for other types of glass with good accuracy, by using the factors given in Table 14. Table 12 values are multiplied by the appropriate factor given in Table 14 to obtain heat gain due to transmitted solar radiation. For glasses having a transmittance for normally incident radiation differing from the table values, factors may be found by linear interpolation. To obtain instantaneous rates of heat gain by convection and radiation, two steps are required. *First*, Table 13 values are multiplied by the appropriate coefficient of X listed in Table 14. *Second*, Table 15 values are multiplied by the appropriate coefficient of Y listed in Table 14, and added to the first value. All convection and radiation gain values for double glass were computed for a ¼-in. air space. No great error is involved in cooling load estimates if these are used for double glass with other air spaces.

Example 8: Find the total instantaneous heat gain through a single sheet of regular plate glass in a southwest wall at 2 p.m. sun time and 40-deg north latitude on August 1. The maximum dry-bulb temperature for design is 98 F; the atmosphere is clear. The indoor temperature is 80 F.

Solution: From Table 12 the heat gain due to transmitted radiation is 148 Btu per (hr) (sq ft) for common window glass; from Table 14, the factor for regular plate glass is 0.87. The coefficient of X in Table 14 is 1.0, while X is found from Table 13 for common window glass for the same hour, orientation and latitude. The coefficient of Y in Table 14 is 0.25, while the Y value is found from Table 15 for a southwest wall at 2:00 p.m. and 40-deg north latitude. The correction for design dry-bulb temperature is found from Table 24 to be 1.0 Btu per (hr) (sq ft) per degree difference from 95 F design temperature. The total instantaneous heat gain is, from Equation 2a,

$$q = 0.87 \times 148 + 1.0 \times 19 + 0.25 \times 27 + 1.0\ (98 - 95)$$
$$= 157.5 \text{ Btu per (hr) (sq ft)}.$$

Design Tables for Rolled Figured Glass

Tables 16 and 17 give design values of instantaneous rates of heat gain for a number of common patterns of single vertical sheets of rolled figured glass. The tables are for a solar declination of 18 deg, which corresponds to a nominal August 1 day, and are based upon the solar intensity values for a clear atmosphere as given in Table 4. The values are given in terms of corrections to apply to Tables 12 and 13. The heat gain due to transmitted solar radiation is found by multiplying the Table 12 values by the approximate percentages given in Table 17. To obtain instantaneous rates of heat gain by convection and radiation, Table 15 values are multiplied by the appropriate value of Y from Table 16 and then added to the corresponding Table 13 values. The total instantaneous heat gain is the sum of the gain due to transmitted solar radiation and the gain by convection and radiation.

The values given in Tables 16 and 17 are based upon an ASHAE research paper[23] to which the reader is directed for additional data. The values in Tables 16 and 17 may be used with fair precision for other patterns of similar transmittance and surface characteristics. For example, the data for hammered glass may be used for glass having shallow, closely spaced ribs or for glass having small, closely spaced circular indentations. Because some patterns have distinct orienta-

Table 15. . . . Heat Absorbed in Glass. Values of Y to be Used with Factors in Table 14 and Table 16 in the Determination of Instantaneous Rates of Heat Gain Due to Convection and Radiation for Various Types of Single Glass and Combinations of Two Sheets of Glass Spaced at ¼ in.

For Clear Atmospheres and 18 Deg Declination, North (August 1)

Sun Time	Latitude	Values of Y in Btu per (hr) (sq ft)[a]								
		N	NE	E	SE	S	SW	W	NW	Horiz.
5 a.m.		0	0	1	0	0	0	0	0	0
6		4	16	18	9	1	1	1	1	3
7		2	24	30	20	2	2	2	2	11
8		2	22	33	25	2	2	2	2	21
9		2	16	30	29	8	3	3	3	32
10		3	5	25	27	14	3	3	3	37
11	40	3	3	12	21	18	3	3	3	42
12	Degrees	3	3	3	15	19	12	3	3	45
1 p.m.	North	3	3	3	3	19	22	10	3	44
2		3	3	3	3	16	27	24	4	41
3		3	3	3	3	10	30	31	15	35
4		3	3	3	3	4	29	36	23	26
5		2	2	2	2	2	23	34	27	17
6		4	1	1	1	1	14	24	21	6
7		0	0	0	0	0	2	3	3	1

Sun Time	Latitude	SE	S	SW	Sun Time	Latitude	SE	S	SW
5 a.m.		0	0	0	5 a.m.		2	0	0
6		7	1	1	6		13	1	1
7		18	2	2	7		22	2	2
8		22	2	2	8		28	3	2
9		24	3	3	9		30	13	3
10		22	5	3	10		31	20	3
11	30[b]	16	7	3	11	50[b]	27	25	5
12	Degrees	6	9	4	12	Degrees	20	27	17
1 p.m.	North	3	9	14	1 p.m.	North	9	25	26
		3	6	21	2		3	22	32
3		3	5	27	3		3	16	33
4		3	3	26	4		2	7	31
5		2	2	21	5		2	2	26
6		1	1	11	6		1	1	17
7		0	0	0	7		0	0	7

[a] Values of Y for 8 and 9 p.m. are zero.
[b] For N, NE, E, W, NW, and horizontal use 40 deg North Latitude values.

tion properties, no attempt has been made to give values for nonvertical glass.

Design Tables for Glass Block Walls

Table 18 describes the glass block patterns discussed in following text. Table 19 gives design values for instantaneous heat gain due to *transmitted direct and diffuse solar radiation* for 8-in. hollow glass block of Type I pattern. For glass blocks of Types II to V the corresponding instantaneous heat gain is found by multiplying the value found for Type I block from Table 19 by the appropriate factor from Table 20.

The instantaneous rate of heat gain due to *convection and radiation* for unshaded 8-in. hollow glass blocks of Types I to V are given in Table 21. The *convection and radiation* gain values for all blocks are so nearly the same that a single table suffices. Note, however, that corrections must be made for certain hours for east and west facing walls of some patterns.

Tables 22 and 23 give instantaneous heat gain values for two types of glass block design for use in skylights. The Type VI block is 10⅝ x 10⅝ in. and the Type VII block 12 x 12 in.

It should be noted that the data for Type VI glass block in Tables 22 and 23 are for panels as assembled. Since the manner of mounting this type of block is standard the ratio of glass area to *opaque* area is virtually constant from one skylight to another. The Type VII glass block may be mounted on a wide range of center-to-center distances. The *opaque* portion of the assembly should be given separate consideration independent of the glass block. See Reference 24 for more complete information.

These tables are based upon the solar intensity values for a clear atmosphere as given in Table 4. For solar energy transmittance data refer to References 21 and 24.

The values of convection and radiation gain are based on values of 80 F for indoor air temperature, a maximum outdoor air temperature of 95 F and upon experimentally determined values of solar energy and temperature difference between the two faces. Indoor and outdoor convection and radiation heat-transfer values are the same as those used for common

Table 16 . . . . Application Factors to Apply to Tables 13 and 15 to Obtain Instantaneous Rates of Heat Gain for Vertical Single Sheets of Rolled Figured Glass Having Normal Incidence Transmittances and Listed Thicknesses (Smooth Side Indoor, Figured Side Outdoor)

(See Table 17 for Factors to Apply to Table 12 Values)

Glass Pattern	Normal Incidence Transmittance	Thickness, Inches	Factor to Apply to Table 13
Hammered	0.75	7/32	1.0(X)[a] + 0.50(Y)[b, e]
Hammered, etched both sides	0.67	7/32	1.0(X) + 0.65(Y)[d]
Deep ribs on ⅛ in. centers	0.77 approx.	7/32	1.0(X) + 0.50(Y)[e]
Hammered heat absorbing	0.21	¼	1.0(X) + 1.15(Y)[f]
Hammered heat absorbing, etched both sides	0.14	¼	1.0(X) + 1.40(Y)

[a] X values are Table 13 values.
[b] Y values are Table 15 values.
[c] Use 0.40(Y) for east and west glass.
[d] Use 0.60(Y) for east and west glass.
[e] Use 0.35(Y) for east and west glass.
[f] Use 0.95(Y) for south glass.

Table 17 . . . . Instantaneous Rates of Heat Gain Due to Transmitted Direct and Diffuse Solar Radiation by Unshaded Rolled Figured Glass

Multiply the Table 12 Values by These Percentage Factors

For Clear Atmospheres and 18 Degrees Declination, North (August 1) For 30, 40, and 50 Degrees North Latitude

Note: To obtain total instantaneous heat gain add adjusted Table 12 values to adjusted Table 13 values

Instantaneous Heat Gain Due to Transmitted Solar Radiation as a Percentage of Single Sheet of Common Window Glass

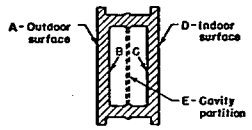
Sun Time	Hammered					Hammered and Etched				
AM → ↓	N, NW W, SW	NE	E	SE	S	N, NW W, SW	NE	E	SE	S
5 a.m. 7 p.m.	80	80	85	70	80	60	65	75	55	60
6 6	80	85	85	75	80	60	65	75	50	60
7 5	80	80	85	80	80	60	60	70	55	60
8 4	80	75	85	80	75	60	55	65	55	60
9 3	80	60	80	80	60	60	50	55	55	50
10 2	80	65	75	75	65	60	55	50	55	50
11 1	80	80	60	70	65[a]	60	60	50	50	50
12	80	80	80	60	65[a]	60	60	60	50	50

Sun Time	Ribs on ⅛ in. Centers					Hammered Heat Absorbing				
5 a.m. 7 p.m.	60	75	85	25	60	25	20	25	20	25
6 6	60	80	85	30	60	25	20	25	20	25
7 5	60	65	85	40	60	25	20	25	20	25
8 4	60	40	80	50	55	25	20	20	20	25
9 3	60	30	70	65	35	25	20	20	20	20
10 2	60	40	40	55	35	25	20	20	20	20
11 1	60	60	35	40	35[b]	25	25	20	20	20
12	60	60	60	35	45[b]	25	25	25	20	20
↑ PM →	N, NE E, SE	NW	W	SW	S	N, NE E, SE	NW	W	SW	S

Sun Time	Hammered and Etched Heat Absorbing				
AM → ↓	N, NW W, SW	NE	E	SE	S
5 a.m. 7 p.m.	20	15	15	10	20
6 6	20	15	15	10	20
7 5	20	10	15	10	20
8 4	20	10	15	10	20
9 3	20	10	10	10	10
10 2	20	15	10	10	10
11 1	20	20	10	10	10
12	20	20	20	10	10
↑ PM →	N, NE E, SE	NW	W	SW	S

[a] Decrease values 10 percent for 30 deg latitude; increase 15 percent for 50 deg latitude.
[b] Increase values 30 percent for 50 deg latitude.

Table 18 .... Description of Glass Block Patterns



Hollow Glass Block Used for Vertical Walls

Type I —Smooth Face
A, D: Smooth
B: Wide vertical ribs or flutes
C: Wide horizontal ribs or flutes
E: None

Type II —Semi-Light Diffusing
A, D: Narrow vertical ribs or flutes
B, C: Etched or stippled
E: None

Type III—Light Diffusing
A, D: Narrow vertical ribs or flutes
B, C: Etched or stippled
E: Glass fiber screen

Type IV —Light Diffusing
A, D: Close pitch deep horizontal corrugations
B, C: Vertical light diffusing prisms
E: None

Type IVA—Light Diffusing
Same as IV except corrugations vertical

Type V —Light Directing
A, D: Close pitch deep vertical corrugations
B, C: Horizontal light directing prisms
E: None

Hollow Glass Block Used for Horizontal Skylights

Type VI —Light Directing (skylight)
A: Smooth surface
B: Light-directing prisms
C, D: Light-diffusing surface
E: Clear glass fiber screen

Type VII—Light-diffusing (skylight)
A: Smooth surface
B, C: Stippled surface
D: Narrow-ribbed surface
E: Green-tinted glass fiber screen

window glass. Table 24 gives corrections to be applied for other design temperatures.

Example 9: Find the total instantaneous heat gain through an east wall of 8-in. hollow glass block of Type V pattern at 8 a.m. and 50-deg north latitude. The design temperatures are 80 F indoors and 95 F maximum outdoor dry-bulb, with a clear atmosphere.

Solution: The gain due to transmitted solar radiation is found from Table 19 for Type I pattern. The factor for Type V is found from Table 20. The convection and radiation gain is found from Table 21 (note the footnote).

The total instantaneous heat gain is, from Equation 2a,

$$q = 86 \times 0.65 + 26 \times 1.4 = 92.3 \text{ Btu per (hr) (sq ft)}.$$

Effect of Deviations from Design Conditions

If the indoor temperature differs from 80 F, or the design outdoor dry-bulb temperature differs from 95 F, corrections can be made to the convection and radiation gain values for flat glass, rolled figured glass, and glass block according to the schedule in Table 24.

The effect of the humid industrial-type atmosphere is to cause a considerable reduction in heat gain, if all factors except solar intensity remain the same. Reference 22 gives heat gain values for four orientations at 40-deg north latitude on August 1 for several types of flat glass and glass block, and solar intensities typical of humid industrial atmospheres. These data show that the following approximate reductions, based on total gain for the day, can be expected: 20 percent for all types of glass and glass block in east or west facing walls; 10 percent for south facing flat glass; 5 percent for south facing glass block walls.

Shading of Glass Areas—Shade Factors

The shade factor is defined as the total gain from a shade-glass combination minus the convection and radiation gain from single unshaded common window glass divided by the total incident solar radiation transmitted by single unshaded common window glass. In equation form,

$$\text{Shade factor} = \frac{\left(\begin{matrix}\text{Total Gain from}\\ \text{Shade Glass}\\ \text{Combination}\end{matrix}\right) - \left(\begin{matrix}\text{Convection \& Radiation}\\ \text{Gain from Single}\\ \text{Unshaded Common Glass}\end{matrix}\right)}{\begin{matrix}\text{Total Solar Energy Transmitted by}\\ \text{Single Unshaded Common Glass}\end{matrix}}$$

Design shade factors for a number of combinations are listed in Table 25. The total gain for a shaded window is found by multiplying the values of Table 12 by the appropriate shade factor and adding to these values the corresponding values from Table 13.

The values for sunlit windows are average values based upon calculations for several orientations. Actually the shade factor varies somewhat with orientation and time of day, because the proportion of direct to diffuse incident solar radiation as well as the profile angle varies. Generally these variations may be neglected.

There are a number of variables affecting these ratios such as color, fit, solar altitude, and angle of incidence of the solar radiation. These values, therefore, must be considered as approximate, only, and will have to be used with considerable judgment. An inside shade is effective to the extent of its reflectivity, since the portion of the solar radiation directly transmitted by the glass that is absorbed by the shade is transferred by convection to the room air, and by radiation to the solid room surfaces.

A more complete analysis of shade factors may be found in Reference 25.

Roof Overhang, Canopies, and Other Horizontal Projections

In some cases, a considerable reduction in heat transfer through windows can be attained by the use of a suitable horizontal projection. This is particularly true for S, SE, and SW orientations. In Table 26 are tabulated the required projections to produce a shadow height of 10 ft on a window or wall. The projection necessary for other shadow heights can be found by direct ratio. For example, a shadow height of 5 ft would require one-half the projection needed for a shadow height of 10 ft. Note that the table is for the period from April 11 through September 1.

The choice of projection should not necessarily be based on providing complete shading during all daylight hours. Reference should be made to Tables 12 through 15 to determine during which hours of the day the solar gain is sufficiently large to indicate the desirability of shading. This and the architectural practicability of the required projection should be deciding factors.

Table 19 .... Instantaneous Rate of Heat Gain Due to Transmitted Direct and Diffuse Solar Radiation by Unshaded Walls of 8-in. Hollow Glass Block of Type I Pattern

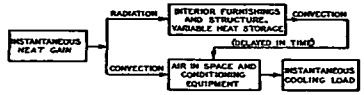
For Clear Atmospheres and 78 Deg Declination, North (August 1)

Note: For total instantaneous heat gain add these values to Table 21 values

Latitude	Sun Time	Instantaneous Heat Gain in Btu per (hr) (sq ft)							
	AM → ↓	N	NE	E	SE	S	SW	W	NW
30 Deg north	6 a.m. 6 p.m.	4	45	55	12	2	2	2	2
	7 5	5	59	94	29	4	3	3	3
	8 4	5	42	94	38	5	4	4	4
	9 3	5	25	59	34	6	5	5	5
	10 2	6	12	27	24	9	6	6	6
	11 1	6	8	12	13	10	7	6	6
	12	6	6	7	8	9	8	7	6
40 Deg north	5 a.m. 7 p.m.	1	3	3	0	0	0	0	0
	6 6	5	50	67	17	2	2	2	2
	7 5	4	54	98	36	4	4	4	4
	8 4	5	34	90	47	5	4	4	4
	9 3	5	18	59	47	10	5	5	5
	10 2	6	8	29	35	15	5	6	6
	11 1	6	6	13	22	18	7	6	6
	12	6	6	6	13	17	13	6	6
50 Deg north	5 a.m. 7 p.m.	4	28	27	4	1	1	1	1
	6 6	5	53	77	22	3	2	2	2
	7 5	4	44	101	44	4	4	4	4
	8 4	4	26	86	57	7	4	4	4
	9 3	5	12	57	60	16	5	5	5
	10 2	5	6	29	54	25	6	5	5
	11 1	6	6	14	34	32	10	6	6
	12	6	6	6	20	34	20	6	6
	↑ PM →	N	NW	W	SW	S	SE	E	NE

INSTANTANEOUS HEAT GAINS VS. INSTANTANEOUS COOLING LOADS

The difference between instantaneous heat gain and instantaneous cooling load has been mentioned previously; its practical importance is sufficient to warrant further consideration. Fig. 4 offers a simplified schematic illustration showing how the *radiative* part of the instantaneous heat gain is first absorbed by solid objects, and is not encountered by the conditioning equipment as a cooling load until some later time, when it finally appears in the air stream entering the equipment. While it is true that some lag also is inherent in convective heat transfer and the time required to change the air in the conditioned space, this is usually of the order of a few minutes to perhaps half an hour. Heat storage in the interior furnishings and structure increases according to the proportion of the instantaneous heat gain which is in the form of radiation, and also increases as the thermal capacitance of the objects and materials involved is increased.



The radiation absorbed by the interior furnishings and structure reaches the conditioning equipment after a considerable delay in time.

Fig. 4 .... Origin of the Difference Between the Magnitudes of the Instantaneous Heat Gain and Instantaneous Cooling Load

Constituents of the total instantaneous heat gain which have appreciable radiation components include those due to glass areas, exposed walls and roofs, lighting, appliances, and people.

A large difference in the time-incidence of the peaks between various spaces or parts of the same space indicates the necessity for zoning. In a building having an east and west exposure, where solar heat gains form a fair share of the cooling load, the times of individual zone peaks are apt to be some hours apart, and the peak load of one plus the off-peak load of the other will be substantially less than their combined peak loads. Proper zoning will permit operation to take full advantage of this condition or of similar conditions of nonsimultaneous peaks, and will result in a lower total load and in savings in equipment.

A factor similar in effect and closely related to the non-

Table 20 .... Instantaneous Rates of Heat Gain Due to Transmitted Direct and Diffuse Solar Radiation by Unshaded Walls of 8-in. Hollow Glass Block of Types II, III, IV, IVA and V Patterns

Multiply the Table 19 Values for Type I by These Percentage Factors

For Clear Atmospheres and 18 Deg Declination, North (August 1) For 30, 40, and 50 Deg North Latitude

Note: To obtain total instantaneous heat gain add adjusted Table 19 values to the Table 21 values

Instantaneous Heat Gain Due to Transmitted Solar Radiation as a Percentage of Type I Pattern

Sun Time	Type II Pattern					Type III Pattern				
AM → ↓	N, NW, W, SW	NE	E	SE	S	N, NW, W, SW	NE	E	SE	S
5 a.m. 7 p.m.	100	100	100	100	100	70	70	70	70	70
6 6	100	95	95	90	100	70	65	65	70	70
7 5	100	90	90	90	100	70	65	60	65	70
8 4	100	95	90	90	100	70	65	60	65	70
9 3	100	95	85	90	90	70	65	65	70	65
10 2	100	95	95	95	95	70	70	70	75	65
11 1	100	100	95	95	100	70	70	70	80	75
12	100	100	100	100	100	70	70	70	75	90
↑ PM →	N, NE, E, SE	NW	W	SW	S	N, NE, E, SE	NW	W	SW	S

Sun Time	Type IV Pattern					Type IVA Pattern				
AM → ↓	N, NW, W, SW	NE	E	SE	S	N, NW, W, SW	NE	E	SE	S
5 a.m. 7 p.m.	55	45	35	75	55	55	45	35	30	55
6 6	55	45	35	80	55	55	35	35	30	55
7 5	55	45	35	60	55	55	35	40	35	55
8 4	55	45	30	40	55	55	45	45	45	55
9 3	55	50	25	30	60	55	50	65	65	50
10 2	55	55	35	30	50	55	55	85	90	60
11 1	55	55	45	35	40	55	55	70	105	95
12	55	55	55	50	45	55	55	55	100	115
↑ PM →	N, NE, E, SE	NW	W	SW	S	N, NE, E, SE	NW	W	SW	S

Sun Time	Type V Pattern				
AM → ↓	N, NW, W, SW	NE	E	SE	S
5 a.m. 7 p.m.	60	35	35	30	60
6 6	60	35	35	30	60
7 5	60	35	40	35	60
8 4	60	50	65	50	60
9 3	60	80	90	90	60
10 2	60	70	105	105	90[a]
11 1	60	60	80	110	105[a]
12	60	60	60	85	115[a]
↑ PM →	N, NE, E, SE	NW	W	SW	S

Designation of Block Type
II—Semi-Light Diffusing
III—Light Diffusing
IV—Light Diffusing
IVA—Light Diffusing
V—Light Directing

[a] Reduce by 30% for 30 deg N latitude only.

simultaneous occurrence of peak loads, is diversity. Typical of this is the case of a large department store where the air-handling equipment serving a certain space must be sufficient to handle the load created by the throngs of people attending sales in that space. Under such a condition the number of people in other spaces is usually normal or below. While this means that the air-handling equipment for certain departments must be large enough to cope with the situation, the refrigeration equipment need be only large enough to handle the average maximum. If a system employing zone recirculating fans and a single central fan and dehumidifier were used, the saving would be reflected in the capacity of the central fan and dehumidifier. Another example of this diversity is found in an office building having restaurants and stores of certain types in the first story and basement.

Table 21 .... Instantaneous Rates of Heat Gain by Convection and Radiation from Unshaded Walls of 8-in. Hollow Glass Block of Patterns Type I, II, III, IV, IVA and V[a]

For Clear Atmospheres and 18 Deg Declination, North (August 1) For 80 F Indoor Temperature

Note: For total instantaneous heat gain add these values to values in Table 17, or Table 19 adjusted by Table 20 factors

Sun Time	Dry-Bulb Fahr.	Latitude	Instantaneous Heat Gain in Btu per (hr) (sq ft)							
			N	NE	E	SE	S	SW	W	NW
5 a.m.	74		−3	−3	−3	−3	−3	−3	−3	−3
6	74		0	3	4[b]	3	−2	−2	−2	−2
7	75		1	15	17[b]	13	−1	−1	−1	−1
8	77		0	21	26[b]	21	1	0	0	0
9	80		0	20	33	27	5	1	2	1
10	83		2	12	36	32	12	3	3	3
11	87		4	6	29	32	19	5	6	5
12	90	40	6	8	16	26	23	9	8	7
1 p.m.	93	Degrees	8	10	14	14	26	25	12	9
2	94	North	9	10	15	11	25	37	28	11
3	95		11	11	16	11	21	44	41	17
4	94		11	11	15	11	16	45	44	30
5	93		11	10	13	10	13	43	42[b]	35
6	91		13	9	11	9	11	35	35[b]	29
7	87		11	7	8	7	8	19	21[b]	15
8	85		7	4	4	4	4	7	7	4
9	83		4	3	2	3	2	3	4	3

Sun Time	Dry-Bulb	Latitude	SE	S	SW	Latitude	SE	S	SW
5 a.m.	74		−3	−3	−3		−3	−3	−3
6	74		1	−2	−2		5	−2	−2
7	75		11	−1	−1		13	−1	−1
8	77		20	0	0		21	0	0
9	80		26	1	2		27	6	1
10	83		29	4	4		32	15	2
11	87		26	8	5		35	22	4
12	90	30[c]	18	12	7	50[c]	33	28	17
1 p.m.	93	Degrees	10	15	16	Degrees	23	31	31
2	94	North	11	16	30	North	12	32	39
3	95		12	14	39		11	28	43
4	94		12	13	43		11	20	44
5	93		11	11	41		10	12	43
6	91		10	9	32		9	10	36
7	87		6	6	15		7	7	23
8	85		4	4	6		4	4	10
9	83		3	3	3		3	3	3

[a] For types III, IV, IVA and V patterns and 30, 40 and 50 deg latitudes, multiply east wall values for 5, 7 and 8 a.m. by 1.40, and west wall values for 5, 6 and 7 p.m. by 1.25.

[b] For N, NE, E, W and NW use 40 deg North Latitude values.

At noon, when the restaurants and stores are crowded, the offices are below normal occupancy.

Heat lag should be carefully considered in the cooling load calculations. In certain types of buildings the effect of solar radiation is still apparent several hours after the sun has shifted from that exposure. In other types having a much lighter construction, the heat gain due to solar radiation decreases markedly with the passing of the sun. Some walls, warmed by the sun, may radiate heat long after the passing of the sun, thus requiring lower indoor temperatures to offset the radiant energy.

Buildings have considerable heat storage capacity which can often be utilized to great advantage, and which has provided an unexpected safety factor in many installations. If a space is kept below the design indoor temperature for some time, the interior walls, floors, furniture, and fixtures begin to assume the temperature of the space. Where the time is sufficient the entire mass, rather than merely its surface, may reach the room temperature. Thus, when a space has been precooled below the design maximum temperature for a period of time prior to the advent of the peak load, and the heat gain begins to increase to peak conditions, some of the increase is used in raising the temperature of the furniture, fixtures, etc., to the design conditions and the cooling load can be reduced accordingly. However, unless very accurate data with regard to the mass, surface, specific heat, etc., of the items within the space are available, due caution must be used in discounting the cooling load for this storage effect. In the absence of reliable data this allowance is often a matter of experience rather than calculation.

Table 22 .... Instantaneous Rates of Heat Gain Due to Transmitted Direct and Diffuse Solar Radiation by Unshaded Skylights of Hollow Glass Block of Types VI and VII Patterns

For Clear Atmosphere and 18 Deg Declination, North (August 1)

Note: For total instantaneous heat gain add these values to Table 23 values

Instantaneous Heat Gain in Btu per (hr) (sq ft)[a]

Sun Time	Type VI, North Latitude			Type VII, North Latitude			Sun Time
	30	40	50	30	40	50	
5 a.m. 7 p.m.	0	0	1	0	0	1	5 a.m. 7 p.m.
6 6	3	4	4	3	4	5	6 6
7 5	7	8	7	10	11	11	7 5
8 4	11	13	14	19	19	18	8 4
9 3	19	19	20	31	29	26	9 3
10 2	29	23	22	41	38	33	10 2
11 1	38	26	23	49	44	38	11 1
12	37	27	22	52	46	39	12

[a] Values for Type VI are for panel as assembled. Values for Type VII are for glass block only. See text.

Table 23 .... Instantaneous Rates of Heat Gain by Convection and Radiation from Unshaded Skylights of Hollow Glass Block of Types VI and VII Patterns

For Clear Atmosphere and 18 Deg Declination, North (August 1) For 80 F Indoor Temperature

Note: For total instantaneous heat gain, add these values to Table 22 values

Instantaneous Heat Gain in Btu per (hr) (sq ft)[a]

Sun Time	Type VI, North Latitude			Type VII, North Latitude			Sun Time
	30	40	50	30	40	50	
5 a.m.	−9	−9	−9	−9	−9	−9	5 a.m.
6	−9	−9	−9	−9	−9	−9	6
7	−9	−9	−8	−9	−9	−9	7
8	−7	−5	−4	−7	−7	−5	8
9	4	5	5	2	4	4	9
10	14	14	13	11	11	11	10
11	21	21	19	16	16	15	11
12	29	27	25	20	20	17	12
1 p.m.	34	32	29	25	23	22	1 p.m.
2	37	34	31	27	26	24	2
3	37	35	32	28	26	25	3
4	34	32	29	26	25	22	4
5	29	28	26	24	23	22	5
6	22	22	22	20	20	19	6
7	14	14	14	13	13	13	7

[a] Values for Type VI are for panel as assembled. Values for Type VII are for glass block only. See text.

Table 24 .... Approximate Corrections to Tables 13 and 21 for Deviations from Indoor and Outdoor Design Temperatures

For each degree the design room temperature exceeds 80 F, subtract correction.[a] For each degree the design outdoor dry-bulb temperature exceeds 95 F, add correction.[b] Apply these corrections to each value in Table 13 or Table 21.

Glass Type	Correction Btu per (hr) (sq ft)
Single flat or rolled figured glass	1.0
Double flat glass and glass block	0.5

[a] If temperature is less than 80 F, add correction.
[b] If temperature is less than 95 F, subtract correction.

Where air-conditioning supply and return ducts pass through unconditioned spaces, there will be a transfer of heat from these spaces to the air in the ducts, even though these ducts are well insulated. An allowance should be made for this heat gain and included in the heat estimate so that air can be supplied at a temperature low enough to offset the rise caused by this heat gain (see Chapter 21). There will also be some heat gain to the air in ducts passing through conditioned spaces, but since a cooling effect is produced in *the space through which the duct passes, this is not a loss* and usually can be compensated for by adjustment of air quantities between the various spaces.

No comprehensive data are presently available for use in design load estimates to evaluate the interior load-lag effect, but several investigators[12, 14, 26, 27, 28] have made a study of the problem and have presented many useful data. Tables 9, 10, 12, 13, 17, 19, 20, 21, 22, and 23 are all based on instantaneous rates of heat transfer. Hence, practical judgment and experience offer the only basis of procedure. Until the needed data become available, it is recommended that the noncontinuous load be averaged over two or three hours during the time of maximum load, when determining the total instantaneous cooling load where a large portion of the heat gain is radiant. This suggestion applies only to conditions near the time of maximum heat gain, as the heat stored within the structure would necessarily appear in the cooling load eventually; but if it appears at a time when the gain from outdoors is relatively low, the equipment will be able to maintain satisfactory conditions within the range of maximum capacity.

LOAD FROM INTERIOR PARTITIONS, CEILINGS, AND FLOORS

Whenever a conditioned space is adjacent to another space in which a different temperature prevails, the transfer of heat through the separating structural section must be considered. Calculations are made according to the relation:

$$q = U_i A_i (t_b - t_i) \text{ Btu per hour.} \quad (6)$$

where

U_i = coefficient of overall heat transfer between the adjacent and the conditioned space, Btu per (hour) (square foot) (Fahrenheit degree).

Table 25 .... Shade Factors for Various Types of Shading

Type of Shading	Finish on Side Exposed to Sun	Shade Factor
Canvas awning sides open	Dark or medium	0.25
Canvas awning top and sides tight against building	Dark or medium	0.35
Inside roller shade, fully drawn[a]	White, cream	0.41[f]
Inside roller shade, fully drawn[a]	Medium	0.62
Inside roller shade, fully drawn[a]	Dark	0.81
Inside roller shade, half drawn[a]	White, cream	0.71
Inside roller shade, half drawn[a]	Medium	0.81
Inside roller shade, half drawn[a]	Dark	0.91
Inside venetian blind, slats set at 45 deg[b]	White, cream	0.56[f]
Inside venetian blind, slats set at 45 deg[b]	Diffuse reflecting aluminum metal	0.45[f]
Inside venetian blind, slats set at 45 deg[b]	Medium	0.65[f]
Inside venetian blind, slats set at 45 deg[b]	Dark	0.75[f]
Outside venetian blind, slats set at 45 deg[b]	White, cream	0.15[f]
Outside venetian blind, slats set at 45 deg[b, c] extended as awning fully covering window	White, cream	0.15[f]
Outside venetian blind, slats set at 45 deg, extended as awning covering ⅔ of window[c]	White, cream	0.43
		Dark[d, f] / Green tint[e, f]
Outside shading screen, solar altitude 10 deg		0.52 / 0.46
Outside shading screen, solar altitude 20 deg		0.40 / 0.35
Outside shading screen, solar altitude 30 deg		0.25 / 0.24
Outside shading screen, solar altitude, above 40 deg		0.15 / 0.22

[a] Roller shades are assumed to be opaque. Some white shades may transmit considerable solar radiation. For white translucent shades fully drawn use 0.55 and for half drawn use 0.77.
[b] Venetian blinds are fully drawn and cover window. It is assumed that the occupant will adjust slats to prevent direct rays from passing between slats. If slats are fully closed (slats set at 90 deg) use same factors as used for roller shade fully drawn.
[c] Commercial shade with wide slats. The sun may shine on window through sides of shade. Estimate the exposed portion of glass as unshaded.
[d] Commercial shade, bronze. Metal slats 0.05 inches wide 17 per inch and set at 17 deg angle with horizontal. At solar altitudes below 40 deg some direct solar rays are allowed to pass between slats, and this amount becomes progressively greater at low solar altitudes.
[e] Commercial aluminum shade. Slats 0.057 inches wide, 17.5 per inch, set at 17 deg angle with horizontal. At solar altitudes below 40 deg some direct solar rays are allowed to pass between slats and this amount becomes progressively greater at low solar altitude.
[f] From first paper in Reference 25.

Table 26 .... Length of Horizontal Projection Required for Shading Windows and Walls

For Shading 10 Ft Down From Projection For April 11 Through September 1

Latitude	Sun Time (AM →)		Projection in Feet N	NE	E	SE	S	SW
30 Deg north	6 a.m.	6 p.m.	17.4	*	*	*	—	—
	7	5	5.6	*	*	*	0.9	—
	8	4	2.0	11.2	17.4	13.8	2.5	—
	9	3	0.6	6.3	10.8	9.7	3.5	—
	10	2	—	3.3	6.3	7.1	3.9	—
	11	1	—	1.1	3.1	4.7	4.1	0.9
	12		—	—	—	2.9	4.1	2.9
40 Deg north	5 a.m.	7 p.m.	*	*	*	*	—	—
	6	6	12.0	*	*	*	—	—
	7	5	3.5	*	*	*	1.8	—
	8	4	—	9.7	18.9	16.1	4.3	—
	9	3	—	4.7	11.2	11.6	5.4	—
	10	2	—	1.3	6.5	9.1	5.8	—
	11	1	—	—	3.1	6.5	6.1	2.2
	12		—	—	—	4.3	6.3	4.3
50 Deg north	5 a.m.	7 p.m.	*	*	*	*	—	—
	6	6	8.4	*	*	*	—	—
	7	5	1.6	*	*	*	3.7	—
	8	4	—	9.4	19.7	18.9	6.5	—
	9	3	—	3.1	12.8	14.3	7.9	—
	10	2	—	0.6	7.6	11.6	8.4	0.6
	11	1	—	—	3.7	8.7	8.7	3.9
	12		—	—	—	6.5	9.1	6.5
		(PM →)	N	NW	W	SW	S	SE

[a] Projection greater than 20 ft required.

A_i = area of separating section concerned, square feet.

t_b = air temperature in adjacent space, Fahrenheit.

t_i = air temperature in conditioned space, Fahrenheit.

Magnitudes of U_i may be obtained from Chapter 9. The temperature t_b may have any value over a considerable range, according to conditions in the adjacent space. The temperature in a kitchen or boiler room may be as much as 15 to 50 deg above the outdoor air temperature. It is recommended that actual temperatures be measured in adjoining spaces wherever practicable. Where nothing is known, except that the adjacent space is of conventional construction and contains no heat sources, it is recommended that the difference $(t_b - t_i)$ be taken as the difference between the outdoor air and conditioned-space design dry-bulb temperatures *minus* 5 deg. In some cases it may be that the air temperature in the adjacent space will correspond closely to the outdoor air temperature at all times. Under these latter conditions, the heat gain through the partition will be periodic in nature, and the value of a shaded wall should be used from Table 10.

For floors directly in contact with the ground, or over an underground basement that is neither ventilated nor warmed, the heat transfer may be neglected for cooling-load estimates.

LOAD FROM OUTDOOR AIR, VENTILATION, AND INFILTRATION

Ventilation. Data for determining the necessary ventilation rate have been presented previously in this chapter. Ventilation required is primarily dependent upon the number of occupants and upon the materials and apparatus within the space which may give off odors. For spaces having ceiling heights 10 ft or less, the total requirement should be checked against the volume, and in no case should the ventilation air rate be less than one air change per hour. In spaces having ceilings higher than 10 ft where the occupant load is low, a check calculation can be made against the volume of the space below an assumed 10-ft ceiling.

Infiltration must never be counted upon to provide ventilation, because on still days there will be little or no infiltration.

Infiltration. The principles of infiltration calculations have been discussed in Chapters 11 and 12, with emphasis on the heating season. For the cooling season, infiltration calculations are usually limited to doors and windows.

To compute cooling-load infiltration for windows by the crack method, use the data of Table 2, Chapter 11, for a wind velocity of 10 mph. Note that for double-hung windows the length of crack is three times the width plus twice the height; while for metal-sash windows the crack length is the total perimeter of the movable or ventilating sections. In calculating window infiltration for an entire structure, it is not necessary to consider the total crack length on all sides of the building, for the wind would not act simultaneously on all sides at once. In no case, however, should less than half of the total crack length be figured. A knowledge of the prevailing wind direction will aid judgment in this consideration.

Cooling-load infiltration for doors[28] may be obtained from Table 3, Chapter 11. For conditions other than those covered, the notes appended to the table will provide a basis for estimates. The tabulated data may also be used as the basis of estimates for interior doors between an air-conditioned and a non-air-conditioned space.

Infiltration load must be included whenever the new air introduced through the system is not sufficient to maintain sufficient pressure within the enclosure to prevent the infiltration. Whenever economically feasible, it is desirable to introduce sufficient outdoor air through the air-conditioning equipment to maintain a constant outward escape of air, and thus eliminate the infiltration portion of the load. The pressure maintained must be sufficient to overcome wind pressure through cracks and door openings. When this condition prevails it is not necessary to include any infiltration load. When the quantity of new air introduced through the cooling equipment is not sufficient to build up the required pressure to offset infiltration, the entire infiltration load should be included in the cooling load calculations.

Total Outdoor Air Load. Many cooling coil manufacturers publish tables giving psychrometric data based on the *average* conditions of the leaving air for various coil temperatures, air velocities, and entering dry-bulb and wet-bulb conditions. When these tables are used, it is necessary to calculate the mixed-air condition entering the coil, and determine from the tables what coil and air velocity will produce the desired leaving-air conditions as required for the space to be conditioned. When cooling coils are listed as 80 to 95 percent efficient, the manufacturer indicates that 20 to 5 percent of the air passes through the coil without being cooled. If data of this nature are used, the uncooled portion of the air must be added to the space load before determining the effective air quantity. See later section on Apparatus Dew Point.

To determine the design cooling load caused by the introduction of outdoor air, the maximum rate of outdoor-air entry is first established. In some applications the use of special exhausters from the conditioned space may add to the outdoor-air requirements in determining the maximum rate. When this design quantity is established, and the design indoor- and outdoor-air states are known, the cooling load may be computed. There are several methods in use, the more accurate of which require rather detailed calculations. Refer to Chapter 3, and also section on Apparatus Dew Point in this chapter. The following equations are considered to be of sufficient precision for use at usual design conditions, as their accuracy is within 1 percent.

Sensible Load $q_s = Q \times 60 \times 0.244 \times 0.075 \left(\frac{1 - 0.00923}{0.62}\right) (t_o - t_i)$

$$= Q \times 1.08\ (t_o - t_i), \text{ Btu per hour} \quad (7)$$

Latent Load $q_s = Q \times 60 \times 0.075 \times 1076\ (W_o - W_i)$

$$= Q \times 4840\ (W_o - W_i), \text{ Btu per hour} \quad (8)$$

Total Load $q_t = q_s + q_s$ (9)

where

Q = rate of entry of outdoor air, cubic feet per minute.

t_o = outdoor dry-bulb temperature, Fahrenheit.

t_i = indoor dry-bulb temperature, Fahrenheit.

W_o = outdoor humidity ratio, pounds moisture per pound of dry air.

W_i = indoor humidity ratio, pounds moisture per pound of dry air.

0.075 = standard air density, pounds per cubic foot.

0.244 = a constant approximating the specific heat of dry air corrected for moisture, Btu per (pound) (Fahrenheit degree).

1076 = a factor approximating the average Btu released in condensing one pound of water vapor from air.

As explained later in the section on Apparatus Dew Point, some methods of load calculations break down the ventilation air into two parts: one portion which does not contact the coil surfaces (i.e., bypasses the coil) in passing through the coil and thus becomes a part of the room load; and a second portion, the remainder of the air which contacts the coil surfaces and is cooled down to the apparatus dew point. This detailed method explained in the literature by Ashley,[29] is believed to be very easy and accurate to use. Similar equations, therefore, can be written:

Space Sensible Ventilation Load,

$$q_{si} = Q \times 1.08\ (t_o - t_i) \times b \quad (10)$$

Space Latent Ventilation Load,

$$q_{si} = Q \times 4840\ (W_o - W_i) \times b \quad (11)$$

Remaining Sensible Ventilation Load,

$$q_{ss} = Q \times 1.08\ (t_o - t_i)\ (1 - b) \quad (12)$$

Remaining Latent Ventilation Load,

$$q_{ss} = Q \times 4840\ (W_o - W_i)\ (1 - b) \quad (13)$$

$$q_t = q_{si} + q_{ss} + q_{si} + q_{ss} \quad (14)$$

where

b = fraction of air passing through coil which does not contact surfaces, coil bypass factor.

Standard air weight (0.075 lb per cu ft) is recommended for use in all calculations, as this is the basis for rating fans and its consistent use keeps all parts of the calculations in conformity.

HOW OUTDOOR AIR LOAD AFFECTS ROOM LOAD

Actually, the outdoor air used for ventilation would pass through the conditioning equipment, and be cooled and dehumidified to a lower temperature and humidity ratio than room conditions before entering the room; but for heat-balance purposes the cooling load chargeable to the outdoor air is that corresponding to the difference between the outdoor and indoor air conditions.

One important purpose of the cooling-load estimate is to determine the conditions and quantity of air supplied to the space. All the various sensible and latent heat loads within the space must be included. Infiltration must be included in the space load since this air enters the doors and windows, and its heat and moisture load must be offset by the introduction of cooler, dryer air to the space. However, since outdoor air is taken through the conditioning equipment and cooled, this portion does not become a part of the space load, except a small portion which passes through the coil untreated. To determine the total load on the refrigeration machine, the remaining outdoor-air load must be included in the grand total load.

Example 10: For outdoor design conditions of 95 F dry-bulb and 75 F wet-bulb, and indoor design conditions of 80 F dry-bulb and 67 F wet-bulb, and for the supply of outdoor air at the rate of 1000 cfm and the exhaust of room air at the corresponding rate, calculate the total, sensible, and latent heat gains.

Solution: Substituting in Equation 7:

$$q_s = 1000 \times 1.08\ (95 - 80) = 16{,}200 \text{ Btu per hr.}$$

From psychrometric data $W_o = 0.01413$, $W_i = 0.01122$. Substituting in Equations 8 and 9:

$$q_s = 1000 \times 4840\ (0.01413 - 0.01122) = 14{,}100 \text{ Btu per hr.}$$

$$q_t = q_s + q_s = 30{,}300 \text{ Btu per hr.}$$

HEAT SOURCES WITHIN THE CONDITIONED SPACE

People. The rates at which heat and moisture are given off by human beings under different states of activity are given in Table 27. In many applications these sensible and latent heat gains become a large fraction of the total load. Appreciable variations in heat-emission rates must be recognized according to the age and sex of the individual, state of activity, environmental influences, and duration of occupancy (since for short occupancy the extra heat and moisture brought in by people may be a significant factor).

While Chapter 6 should be referred to for detailed information, Table 27 in this chapter summarizes practical data representing conditions commonly encountered.

Lighting. In general, the instantaneous rate of heat gain from electric lighting[21] may be calculated from the following relation:

$$q_{sl} = \begin{Bmatrix}\text{total light}\\ \text{wattage}\end{Bmatrix} \times \begin{Bmatrix}\text{use}\\ \text{factor}\end{Bmatrix} \times \begin{Bmatrix}\text{special allow-}\\ \text{ance factor}\end{Bmatrix} \times 3.41, \text{ Btu per hr.} \quad (15)$$

The *total light wattage* is obtained from the ratings of all fixtures installed, both for general illumination and for display use.

The *use factor* is the ratio of the wattage in use, for the conditions under which the load estimate is being made, to the total installed wattage. For commercial applications such as stores, the use factor would be unity.

The *special allowance factor* is introduced to care for fluorescent fixtures, and for fixtures which are either ventilated or installed so that only part of their heat goes to the conditioned space. For *fluorescent fixtures*, the special allowance factor is recommended to be taken as 1.20 in order to allow for power consumed in the *ballast*. For *ventilated fixtures, recessed fixtures*, and the like, manufacturers' or other data[21] must be sought to establish the fraction of the total wattage which may be expected to enter the conditioned space.

Power. When equipment of any sort is operated within the conditioned space by electric motors, the heat equivalent of this operation must be considered in the cooling load. The general equation for calculating this load is:

$$q_m = \left(\frac{\text{Horsepower Rating}}{\text{Motor Efficiency}}\right) \times \begin{Bmatrix}\text{Load}\\ \text{Factor}\end{Bmatrix} \times 2544, \text{ Btuh} \quad (16)$$

It is assumed that both the motor and the driven equipment are within the conditioned space. If the motor is outside the space, then *do not divide* by the motor efficiency in Equation 16. The *load factor* is merely the fraction of the rated load which is being delivered under the conditions of the cooling-load estimate. *Motor efficiencies* may be approximated as follows: about 50 to 60 percent at ½ hp rating, increasing to 80 percent at 1 hp, and to 88 percent at 10 hp and above.

Appliances. Care must be taken in a cooling-load estimate to take into account the heat gain from all appliances, electrical, gas, or steam. Table 28 presents recommended data.[21] Note that the *maintaining rate* in Table 28 is the heat input required to maintain the appliance at the normal operating temperature even though it is not being used, i.e., no coffee is being made, no toast is being made, no food is being cooked in the fry kettle, etc. The maintaining rate is useful in setting

Table 27 .... Rates of Heat Gain from Occupants of Conditioned Spaces[a]

Degree of Activity	Typical Application	Total Heat Adults, Male Btu/Hr	Total Heat Adjusted[b] Btu/Hr	Sensible Heat Btu/Hr	Latent Heat Btu/Hr
Seated at rest	Theater—Matinee	390	330	180	150
	Theater—Evening	390	350	195	155
Seated, very light work	Offices, hotels, apartments	450	400	195	205
Moderately active office work	Offices, hotels, apartments	475	450	200	250
Standing, light work; or walking slowly	Department store, retail store, dime store	550	450	200	250
Walking; seated Standing; walking slowly	Drugstore, Bank	550	500	200	300
Sedentary work	Restaurant[c]	490	550	220	330
Light bench work	Factory	800	750	220	530
Moderate dancing	Dance hall	900	850	245	605
Walking 3 mph; moderately heavy work	Factory	1000	1000	300	700
Bowling[d] Heavy work	Bowling alley Factory	1500	1450	465	985

[a] *Note:* Tabulated values are based on 80 F room dry-bulb temperature. For 78 F room dry-bulb, the total heat remains the same, but the sensible heat values should be increased by approximately 10 percent, and the latent heat values decreased accordingly.

[b] *Adjusted total heat gain* is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is 85 percent of that for an adult male, and that the gain from a child is 75 percent of that for an adult male.

[c] *Adjusted total heat value for sedentary work, restaurant,* includes 60 Btu per hour for food per individual (30 Btu sensible and 30 Btu latent).

[d] For *bowling* figure one person per alley actually bowling, and all others as sitting (400 Btu per hour) or standing (550 Btu per hour).

up a lower limit to the heat gain to a room from the appliance when in operation.

Experienced judgment must be used in the application of data given in Table 28. Consideration must be given to the heat contributed by appliances which are in use at the time of peak load. The quantity of heat will depend upon whether products of combustion are vented to a flue, whether they escape into the space to be conditioned, or whether appliances are hooded to allow part of the heat to escape through a stack. There are no generally accepted data available on the effects of venting and shielding heating appliances, but it is believed that when they are properly hooded with a positive fan-exhaust system through the hood, 50 percent of the heat will be carried away and 50 percent dissipated in the space to be conditioned. In the case of gas-burning appliances where the heat of combustion adds considerably to the total heat output, it is believed that a well-designed hood will remove more than 50 percent of the heat generated at the appliance. The same effectiveness of the hood should be figured for both latent and sensible heat.

For each meal served the heat transferred to the dining space is approximately 50 Btu per hr, of which 75 percent is sensible and 25 percent latent.

Heat gains from cooking appliances, motors, lights, and people are not considered in cooling load calculations if the appliances are located in a separate kitchen that is not part of the conditioned area.

LOAD FROM MOISTURE TRANSFER THROUGH PERMEABLE BUILDING MATERIALS

The diffusion of moisture through all common building materials is a natural phenomenon which is always present to a greater or lesser degree.

The permeability and permeance values for various building materials are given in Table 2 of Chapter 10, together with an explanation of moisture transmission through these materials.

In the usual comfort air-conditioning application, it is common practice to neglect moisture transfer through walls, because the actual rate is quite small and the corresponding latent-heat load is hardly significant. *Vapor barriers* are frequently employed in modern construction for the purpose of keeping moisture transfer to a minimum, and reducing the deteriorating and insulation-destroying effects of moisture.

Industrial jobs, on the other hand, frequently call for a low moisture content to be maintained in a conditioned space. Here the matter of moisture transfer cannot be neglected; indeed, it is quite possible to have the latent-heat load accompanying this transfer be of greater magnitude than any other latent-heat load. The equation for computing this load is:

$$\left(\frac{q_m}{A}\right) = \frac{M}{7000} \times \left(\begin{matrix}\text{Vapor-Pressure}\\ \text{Difference, In. Hg}\end{matrix}\right) \times 1076, \text{ Btuh/sq ft} \quad (17)$$

where

M = permeance of the specimen in perms, or grains per (square foot) (hour) (inch of mercury vapor-pressure *difference*).

7000 = grains per pound.

The factor 1076 is defined in list of symbols at Equation 9. (Sensible cooling of the water vapor is included in the factor 1076.)

The only means of preventing moisture transfer is to use a vapor proof wall, or to apply a special lining, which is vapor proof. All openings in moisture proof construction must be equipped with special gaskets to prevent entrance of moisture.

When moisture transfer contributes an appreciable part of the latent-heat load, it is recommended that estimates should be made intentionally liberal in order to avoid later difficulties with insufficient dehumidifying capacity. Storage spaces, for example, would require sufficient dehumidifying capacity to handle the moisture brought in with goods to be stored, in addition to moisture leaking in subsequently.

MISCELLANEOUS HEAT LOADS

This designation is intended to cover the various small heat gains from exposed piping, ducts, work done by circulating fan, and unforeseen contingencies. Where sufficient data are available, these various heat gains may be estimated individually. In the majority of cases, however, common practice is to lump these factors together and combine them with a *safety factor* according to the experience and judgment of the estimator. On this basis, a small safety factor is added to the calculated cooling load to compensate for miscellaneous effects. *No rules* can be given for this procedure, as experience in air conditioning is indispensable for application of suitable safety factors.

APPARATUS DEW POINT AND REQUIRED AIR QUANTITY THROUGH CONDITIONING EQUIPMENT

In ordinary practice, with commercial apparatus, complete saturation of the air is seldom obtained. Four-row finned cooling coils contact approximately 80 percent of the air, whereas six-row finned coils contact approximately 95 percent of the air. In spray-type dehumidifiers of good design the air leaves the dehumidifier at 1 to 2 deg higher wet-bulb temperature than the spray water leaving the dehumidifier, and the difference between the dry-bulb and wet-bulb temperatures leaving the dehumidifier may be as low as 1 deg. A spray-type dehumidifier having sufficient length of spray chamber and density of spray, together with proper arrangement of nozzles, may approach saturation very closely.

As explained in Chapter 3, and shown in Fig. 5, the slope of the line on the psychrometric chart connecting the room condition with the apparatus dew point on the saturation line, determines the ratio of sensible heat absorbing capacity to the moisture absorbing capacity of the supply air. Therefore the room condition can be maintained as long as the supply-air temperature lies on this line, but a greater volume of supply air must be used to satisfy the room load if the cooling coil does not contact 100 percent of the air. For a given room load, the same apparatus dew point will be required whether the cooling appliance contacts all the air or only part of the air.

From the point of view of satisfying the given cooling load requirements, the air passing through the apparatus without being cooled below the dew-point temperature produces two effects:

1. The air quantity which must be passed through the dehumidifier must be increased. Thus, if 20 percent of the air passing is not contacted, then $(20 \div 80) \times 100 = 25$ percent more air must be used than would be necessary if all of it were contacted.

2. Passing untreated air may change the room cooling load, which in turn may change the enthalpy-humidity difference ratio (sometimes called the sensible-heat factor). If return air only is passed through the dehumidifier or if room air only is bypassed, the room load will not change, but if some outdoor air is passed through, the room sensible-heat gain and room latent heat gain will be changed due to the addition of untreated outdoor air which changes the enthalpy-humidity difference ratio. When a load calculation is made, it is necessary to know the percentage of air treated in the dehumidifier, and calculation must be made accordingly.

If the ventilation air is drawn through the dehumidifier before it goes into the room, only that portion of the air not saturated must be included in the room load for the purpose of determining the apparatus dew point and supply-air quantity. It should be noted when evaluating the load added by untreated outdoor air that the temperature difference between room air and outdoor air, and the moisture content difference between room air and outdoor air, should be used, rather than the difference between outdoor air and apparatus dew point, since the rise from the apparatus dew point to room condition is charged against the dehumidifier as the cooling and dehumidifying load.

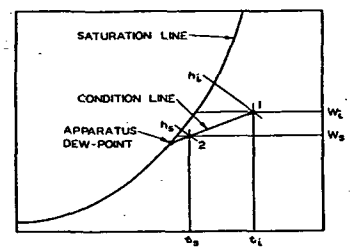


Fig. 5 .... Apparatus Dew-Point Shown on ASHAE PSYCHROMETRIC CHART

The procedure for determining the required air quantity is based upon the thermodynamic principles of Chapter 3 and the use of the ASHAE PSYCHROMETRIC CHART. Readers are advised to review these principles, paying particular attention to the illustrative examples.

Calculation of the cooling load for a conditioned space is equivalent to making, for the space, a heat balance in which all heat, moisture, and infiltration are treated as directly entering the space. As explained in the section, Load from Outdoor Air, Ventilation, and Infiltration, the outdoor-air load normally does not become a part of the space load, because heat and moisture are removed in the air conditioner before this air gets into the conditioned space. The desired conditions are maintained by considering a certain quantity of air to be withdrawn from the space, passed through the conditioning equipment, and returned to the space with such a temperature and humidity ratio that its net effect will be to counterbalance or remove the given entering amounts of heat and water vapor. This quantity of *indoor air*, which is considered to be circulated in this manner, is called the *required air quantity* and its determination is normally part of every cooling-load estimate. The procedure is as follows:

1. Determine the total sensible and latent heat loads in Btu per hour for the space.

2. Compute the quantity called the enthalpy-humidity difference ratio (also referred to as heat-moisture ratio) of

Table 28 .... Rate of Heat Gain From Appliances WITHOUT HOODS[a, b]

Appliance	Capacity	Overall Dimensions (Less Legs and Handles; Last Dimension is Height) Inches	Control A—Automatic M—Manual	Miscellaneous Data	Manufacturer's Rating Watts	Btu/Hr	Maintaining Rate Btu per Hour	Recommended Rate of Heat Gain Btu per Hour: Sensible	Latent	Total
Restaurant Electrical Appliances										
Coffee, brewer and warmer	½ gal		M	Brewer 660 w	600	2000		900	220	1120
			M	Warmer 90 w	90	300	306	230	60	290
Coffee brewer unit with tank	½ gal	20 x 30 x 26		2000 w Water heater, 2960 w brewer	4960	17000		4800	1200	6000
Coffee urn	3 gal	12 x 23 x 21	A	Nickel plated	4500	15000	2600	2200	1500	3700
	5 gal	18 (Diam.) x 37	A	Nickel plated	5000	17000	3600	3400	2300	5700
Doughnut machine		22 x 22 x 57	A	Exhaust system	4700	16000		5000	0	5000
Egg boiler	2 cups	10 x 13 x 25	M		1100	3750		1200	800	2000
Food warmer, with plate warmer, per sq ft of top surface			A	Insulated, separate heat unit for each pot; plate warmer in base	400	1350	500	350	350	700
Food warmer, alone, per sq ft of top surface			A		300	1000	400	200	350	550
Fry kettle	11½ lb fat	12 (Diam.) x 14	A		2600	8900	1100	1600	2400	4000
Fry kettle	25 lb fat	16 x 18 x 12	A	Area 12 x 14 in.	7000	24000	2000	3800	5700	9500
Griddle, frying		18 x 18 x 8	A	Area 18 x 14 in.	2350	8000	2800	3100	1700	4800
Griddle, frying		24 x 20 x 10	A	Area 23 x 18 in.	4000	13500	5000	5300	2900	8200
Grill, meat		14 x 14 x 10	A	Area 10 x 12 in.	3000	10250	1900	3900	2100	6000
Grill, sandwich		13 x 14 x 10	A	Area 12 x 12 in.	1650	5600	1900	2700	700	3400
Roll warmer		23 x 23 x 29	A	Three drawers	1000	3400	900	2400	300	2700
Toaster, continuous	360 slices/hr	15 x 15 x 28	A	2 slices wide	2200	7500	5000	5100	1300	6400
Toaster, continuous	720 slices/hr	20 x 15 x 28	A	4 slices wide	3000	10250	6000	6100	2600	8700
Toaster, pop-up	216 slices/hr	12 x 11 x 9	A	4 slice	2450	8400	2000	4900	900	5800
Waffle iron	20 waffles/hr	12 x 13 x 10	A	7 in. diam. waffle	750	2500	600	1100	750	1850
Restaurant Gas-Burning Appliances										
Coffee brewer and warmer	½ gal		M	Brewer		3400		1350	350	1700
			M	Warmer		500	500	400	100	500
Coffee brewer unit with tank	4½ gal Tank	19 x 30 x 26		4 Brewers and tank				7200	1800	9000
Coffee urn	3 gal	12 x 23 x 21	A	Nickel plated			3400	2500	2500	5000
	5 gal	18 (Diam.) x 37	A	Nickel plated			4700	3900	3900	7800
Food Warmer, per sq ft of top surface			M	Water bath		2000	900	850	430	1280
Fry kettle	15 lb fat	12 x 20 x 18	A	Area 10 x 10		14250	3000	4200	2800	7000
Fry kettle	28 lb fat	15 x 35 x 11	A	Area 11 x 16		24000	4500	7200	4800	12000
Grill		22 x 14 x 17	M	Insulated, grill surface of 1.4 sq ft Top burner 22,000 Btu/hr Bottom burner 15,000 Btu/hr		37000		14400	3600	18000

Table 28 .... Rate of Heat Gain From Appliances WITHOUT HOODS[a, b] (*Concluded*)

Appliance	Capacity	Overall Dimensions (Less Legs and Handles; Last Dimension is Height) Inches	Control A—Automatic M—Manual	Miscellaneous Data	Manufacturer's Rating Watts	Btu/Hr	Maintaining Rate Btu per Hour	Recommended Rate of Heat Gain Btu per Hour: Sensible	Latent	Total
Restaurant Gas-Burning Appliances—Continued										
Stoves, short order										
Open top, per sq ft top			M	Ring type burners		14000		4200	4200	8400
Closed top, per sq ft top			M	Ring type burners		11000		3300	3300	6600
Fry top, per sq ft top			M	Tubular type burners		12000		3600	3600	7200
Toaster, continuous	360 slices/hr	15 x 15 x 28	A	2 slices wide		12000	10000	7700	3300	11000
Toaster, continuous	640 slices/hr	20 x 15 x 28	A	4 slices wide		20000	14000	12000	5000	17000
Restaurant Steam-Heated Appliances										
Coffee urn	3 gal	12 x 23 x 21	T	Nickel plated				2400	1600	4000
	5 gal	18 (Diam.) x 37	T	Nickel plated				3400	2300	5700
Coffee urn	3 gal	12 x 23 x 21	M	Nickel plated				2600	2600	5200
	5 gal	18 (Diam.) x 37	M	Nickel plated				3700	3700	7400
Food warmer, per sq ft of top surface			T					400	500	900
Food warmer, per sq ft of top surface			M					450	1150	1600
Miscellaneous Electrical Appliances										
Hair dryer, blower type			M	Fan, 165 w; Low, 915 w; High, 1580 w	1580	5400		2300	400	2700
Hair dryer, helmet type			M	Fan, 80 w; Low, 300 w; High, 710 w	705	2400		1870	330	2200
Permanent wave machine			M	60 heaters at 25 w each, 36 in normal use	1500	5000		850	150	1000
Neon sign, per linear ft of tube				½ in. outside diam.				30		30
				⅝ in. outside diam.				60		60
Sterilizer, instrument			A	For physicians; thermostat cuts off 550 w before boiling	1100	3750		650	1200	1850
Miscellaneous Gas-Burning Appliances										
Burners, laboratory										
Small Bunsen		7/16 in. Barrel	M	Manufactured Gas		1800		960	240	1200
Small Bunsen		7/16 in. Barrel	M	Natural Gas		3000		1680	420	2100
Fishtail		7/16 in. Barrel	M	Manufactured Gas		3500		1960	490	2450
Fishtail		7/16 in. Barrel	M	Natural Gas		5500		3080	770	3850
Large Bunsen		1½ in. Mouth	M	Adjustable orifice		6000		3350	850	4200
Cigar lighter			M	Continuous Flame		2500		900	100	1000
Hair dryer, 5 helmets			A	Heater and fan blowing air to helmets		33000		15000	4000	19000
Stoves, oven				Insulated, modern		25000	6000	7200	1800	9000
				Not insulated		25000	8500	9200	2300	11500

[a] For restaurant appliances, miscellaneous electrical and miscellaneous gas burning appliances.
[b] When these appliances are hooded and provided with adequate exhaust, use 50 percent of recommended rate of heat gain from unhooded appliances.

the room load, $\frac{h_i - h_s}{W_i - W_s}$. Use the equation

$$\frac{h_i - h_s}{W_i - W_s} = \frac{\text{(Space sensible load + space latent load)}}{\text{Space latent load/1076}} \quad (18)$$

where

h_s = enthalpy of moist air supplied to the space, Btu per pound of dry air.

h_i = enthalpy of moist air at room design conditions, Btu per pound of dry air.

W_s = humidity ratio of moist air supplied to the space, pounds of vapor per pound of dry air.

W_i = humidity ratio of moist air at room design conditions, pounds of vapor per pount of dry air.

Note that the ratio (space latent load/1076) is the equivalent of the required rate of water vapor removal in pounds per hour. If the rate of water removed is known, it may be used directly in Equation 18.

3. Draw a line through the reference point on the ASHAE PSYCHROMETRIC CHART and the value of $(h_i - h_s)/(W_i - W_s)$ determined above. Draw a second line through the state point of the room air (design wet-bulb and dry-bulb temperatures) parallel to this line. This is the *condition line* for the process.

4. Read the temperature where the condition line from step 3 intersects the saturation line. This is called the *apparatus dew point*.

See Fig. 5. Note that instead of using this graphical method the left hand side of Equation 18 may be solved by trial and error by substituting values of h_s and W_s corresponding to assumed apparatus dew-point temperatures.

5. Compute the *required air quantity* from the relation

$$Q_{ra} = \text{Space sensible load} \Big/ \left\{1.08 \left[\begin{pmatrix}\text{Space}\\ \text{dry-bulb}\end{pmatrix} - \begin{pmatrix}\text{Apparatus}\\ \text{dew-point}\end{pmatrix}\right] \times \left(1 - \begin{matrix}\text{Coil}\\ \text{bypass}\\ \text{factor}\end{matrix}\right)\right\} \quad (19)$$

The magnitude of Q_{ra} is substantially the quantity, cfm, of cooled and dehumidified air for which the distribution system must be designed.

The numerical factor 1.08 is derived from the product 1 cfm × 60 min × 0.244 × 0.075$\left(1 - \frac{0.00923}{0.62}\right)$ = 1.08, assuming an average supply air dew point of 55 F. Since standard air density (0.075) includes the weight of the water vapor, it is desirable to reduce it to the basis of dry air by the last factor where 0.00923 = humidity ratio of air at 55 F dew point, and 0.62 = ratio of density of water vapor to dry air at same temperature and pressure. Refer to Chapter 23 for coil selection.

Note that the product [(space dry-bulb) − (apparatus dew point)] × (1 − coil bypass factor) is equal to the dry-bulb range through which the conditioned air is cooled. Hence, in rare instances when the *condition line* of the process may not intersect the saturation line, any other convenient reference temperature on the *condition line* may be used instead, provided that the coil bypass factor is specified accordingly on the proper basis.

MINIMUM ENTERING AIR TEMPERATURE

Due consideration must be given to the temperature of the air entering the conditioned space in order to prevent objectionable drafts. With ceiling-type diffusers or wall grilles with a high aspect ratio (see Chapter 20), many engineers consider 20 deg as the maximum difference for good design under average conditions. This difference can only be exceeded with extremely high ceiling outlets or wall grilles. Thus, if 80 F dry-bulb is to be maintained in a space with average ceiling height, the minimum delivered air temperature would be limited to about 60 F dry-bulb temperature. If the latent heat load is relatively high, it is often necessary to circulate more air with a higher delivered dry-bulb temperature in order to produce a thermodynamic balance. If the dry-bulb temperature of the air supplied to the space is known, the required air quantity can be calculated from the formula,

$$Q_{ra} = \frac{q_s}{1.08\ (t_i - t_s)} \quad (20)$$

or the supply temperature t_s can be determined as follows,

$$t_s = t_i - \frac{q_s}{1.08 \times Q_{ra}} \quad (21)$$

EXAMPLE—COOLING-LOAD CALCULATION

Example 11: A one-story office building Fig. 6 is located in an eastern state near 40 deg latitude. The adjoining buildings on the north and west are not conditioned, and the air temperature within them is known to be substantially equal to the outdoor air temperature at any time of the day.

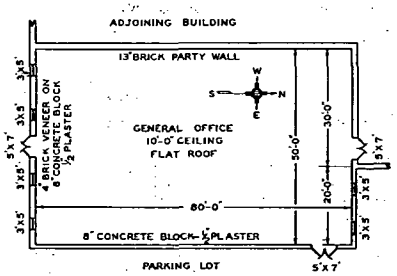


Fig. 6 .... Plan of One-Story Office Building

South wall construction: 8-in. concrete block, 4-in. brick veneer. (Table 7, Chapter 9, U = 0.41.)

East wall and outside north wall construction: 8-in. concrete block, no plaster on walls. (Table 6, Chapter 9, U = 0.52.)

West wall and adjoining north party wall construction: 13-in. solid brick, no plaster:

$$\frac{1}{U} = \frac{1}{1.65} + \frac{13}{5} + \frac{1}{1.65} \quad \text{or,} \quad U = 0.263. \quad \text{Use } U = 0.26.$$

Roof construction: 2½-in. flat roof deck of 2-in. gypsum slab on ¼-in. asbestos cement board surfaced with built-up roofing. (Table 13B, U = 0.34 *for summer*.)

Floor construction: 4-in. concrete on ground.

Window: 3 ft x 5 ft, nonopening type, with medium colored Venetian blinds for windows on south wall. Approximately 4-in. reveal on all windows.

Front doors: Two 2 ft-6 in. x 7 ft (glass panels).

Side doors: Two 2 ft-6 in. x 7 ft (½ glass panels).

Rear doors: Two 2 ft-6 in. x 7 ft (wood panels).

Outdoor design conditions: Maximum dry-bulb 95 F, wet-bulb 78 F; W_o = 0.0168 lbs vapor per lb dry air; h_o = 41.38 Btu per lb dry air.

Indoor design conditions: Dry-bulb 80 F, wet-bulb 65 F; W_i = 0.0098 lb vapor per lb dry air; h_i = 29.95 Btu per lb dry air.

Occupancy: 85 office workers.

Lights: 12,000 watts, fluorescent; 4000 watts, tungsten.

Fan motor: 7½ hp.

Assume that cooling coil has a bypass factor of 0.15, i.e., that 15 percent of the air passes through the coil without contacting the coil surface.

Conditioning equipment to be located in adjoining structure to north.

Find: Total, sensible, and latent maximum cooling loads and required air quantity through conditioning equipment.

Solution: From Table 3, the recommended ventilation rate is 15 cfm per person. Total necessary = 85 × 15 = 1275 cfm or 76,500 cu ft per hr.

As the room volume is 40,000 cu ft, the air changes per hour will be 76,500/40,000 = 1.91 which is more than one air change.

Estimated Time of Maximum Cooling Load:

For this job, judgment indicates that the roof will make the greatest single contribution to the cooling load. Hence, the time of maximum cooling load probably will be the time of maximum heat gain through the roof. From Table 9 the maximum temperature differential for a 2-in. gypsum roof of medium weight construction is 54 deg at 4:00 p.m., and 53 deg at 3:00 p.m. Examination of Table 12 (40 deg N Latitude) shows that solar heat gain through glass on the south wall is 18 Btu per (hr) (sq ft) at 4:00 p.m.; and 42 Btu at 3:00 p.m. This indicates that the maximum cooling load occurs at approximately 3:00 p.m. Therefore make load calculations at 3:00 p.m. sun time. (This may be slightly different from 3:00 p.m. local time.) In some cases, there would be no clear-cut evidence of this nature, and consequently, it would be necessary to estimate the load for several successive times, and then to select the maximum.

Heat Gain Through Outer Wall and Roof Areas:

From Table 10 the temperature differential for the south wall (8-in. concrete block with 4-in. brick veneer) may be about the same as a 12-in. brick which is 6 deg at 3:00 p.m. for a dark colored wall. From the same table, the temperature differential for the east wall (8-in. concrete block with plaster) will be 11 deg at 3:00 p.m. for a *light colored wall* (interpolating between 2:00 and 4:00 p.m.). Likewise, the temperature differential for the north exposed wall (8-in. concrete block plus plaster) will be 3 deg at 3:00 p.m. (by interpolation) for a light wall.

The party wall of 13-in. brick on the west side and part of the north side may be treated as if it were an outside wall in the shade which has a temperature differential (from Table 10) of 2 deg.

For the door in north wall, estimate U = 0.59 from Chapter 9. The outdoor temperature at 3:00 p.m. is 95 F. Neglect time lag and any decrement factor. The temperature differential is $(t_o - t_i)$ = 95 − 80 = 15 deg. The tabulation of the preceding values at 3:00 p.m. is given in the following table:

Section	Net Area Sq Ft	Temperature Differential F Deg	Heat Transmission Coefficient U	Heat Flow Rate per Hour Btu
Roof	4000	53	0.34	72,000
South wall	405[a]	6	0.41	995
East wall	765[a]	11	0.52	4,380
North exposed wall	170[a]	3	0.52	265
West & north party wall	1065[a]	2	0.26	550
Door in north wall	35	15	0.59	310
Total				78,500

[a] Calculated from gross wall area, less windows and doors.

Heat Gain Through Glass Areas:

In computing the load for 3:00 p.m., only the south windows and doors will be exposed to direct sunlight. Tables 12 and 13 will give the total heat gain from the glass areas. The window reveals will shade the south windows; the fraction of the window area receiving direct radiation is obtained from Equation 5 by substituting values as follows:

$r_1 = s/l = 4/60$; $r_2 = 4/36$; β = 45.5 deg, tan β = 1.02

γ = 74 deg, tan γ = 3.487, cos γ = 0.276

$$G_f = 1 - \frac{4}{60}\left(\frac{1.02}{0.276}\right) - \frac{4}{36}(3.487) + \left(\frac{4}{60}\right)\left(\frac{4}{36}\right)\frac{(1.02)(3.487)}{0.276} = 0.462.$$

The south doors will be considered entirely sunlit. The outdoor air temperature is 95 F at 3:00 p.m. From Table 25 the inside Venetian blind factor is 0.65. The instantaneous heat gains due to transmitted direct and diffuse solar radiation, and from convection and radiation gain, are found in Tables 12 and 13 as listed below for the south-facing doors and windows, the north-facing windows and the ½ glass doors in the east wall. The gain through the solid portion of the east doors may be approximated by use of Fig. 3, since the wood panels have little heat capacity. From Table 4 the diffuse radiation value is taken as 18 Btu per (hr) (sq ft) from which $t_o + \alpha_o I_o / f_{co}$ is found to be 98.2 for α = 0.7 and f_{co} = 4.0. From Fig. 3, q = 22.0 Btu per (hr) (sq ft). These heat gains are itemized in the following table.

Location	Area Sq Ft	Fraction Sunlit	Trans Solar Gain Btu/(hr) (sq ft)	Conv and Rad Gain Btu/(hr) (sq ft)	Total Gain Btu/(hr) (sq ft)	Total Gain Btu/hr
South windows*	60	0.462	13	19	32	1920
South doors	35	1.00	42	19	61	2135
East doors						
Glass	18	—	14	17	31	560
Wood	18	—	—	22	22	395
North windows	30	—	15	17	32	960
Total						5970

* Shade factor = 0.65.

In some jobs it would be desirable to increase (or decrease) the instantaneous radiation heat gain by a load-lag factor. The reason for not doing so in this case is that the solar gain is of a low magnitude, and reference to the table indicates that 0.8 of the previous hour would not affect the results materially.

Heat Gain from Ventilation and Infiltration:

Since the desired outdoor air rate 1275 cfm is greater than one air change per hour, it will be satisfactory for determining the ventilation component of the heat gain.

Window infiltration can be taken as negligible since the windows do not open.

Door infiltration requires some judgment. Assume that for each person passing through the double doors, the infiltration will be 100 cu ft of outdoor air, *see* Chapter 10, Table 3. Assume that the outside doors will be used at the rate of 10 persons per hour and the inside doors at the rate of 30 persons per hour. Total infiltration will then be 40 × 100 = 4000 cfh or 67 cfm.

The design rate of entry of outdoor air is then:

$$Q = 1275 + 67 = 1342 \text{ cfm.}$$

The sensible, latent, and total loads are determined from Equations 7, 8, and 9, respectively, at 3:00 p.m. (t_o = 95, t_i = 80, W_o = 0.0168, W_i = 0.0098). All the air entering the room as infiltration becomes a part of the space load.

Infiltration (see Equations 7, 8, and 9):

$q_s = 67 \times 1.08\ (95 - 80) = 1085$ Btuh, sensible.

$q_s = 67 \times 4840\ (0.0168 - 0.0098) = 2270$ Btuh, latent.

$q_t = q_s + q_s = 1085 + 2300 = 3385$ Btuh, total.

Ventilation Air Taken through Cooling Unit Which Becomes a Part of the Space Load (see Equations 10 and 11):

q_{si} = 1275 × 1.08 (95–80) (0.15) = 3,100 Btuh, sensible.

q_{ei} = 1275 × 4840 (0.0168–0.0098) (0.15) = 6,480 Btuh, latent.

Ventilation Air Taken through Cooling Unit Which Does Not Become a Part of the Space Load (see Equations 12 and 13):

q_{ss} = 1275 × 1.08 (95–80) (1–0.15) = 17,600 Btuh, sensible.

q_{es} = 1275 × 4840 (0.0168–0.0098) (1–0.15)
= 36,715 Btuh, latent.

$q_t = q_{si} + q_{ss} + q_{ei} + q_{es}$ = 3100 + 17,600 + 6480 + 36,715
= 63,895 Btuh total.

Heat Gain from Sources within the Conditioned Space:

For the occupants, use the data of Table 27 for moderately active office work.

Sensible heat gain = 85 × 200 = 17,000 Btu per hr.
Latent heat gain = 85 × 250 = 21,250 Btu per hr.
Total = 38,250 Btu per hr.

For the gain from *lighting*, use Equation 15 with a use factor of unity, and a special allowance factor of 1.20 for the fluorescents and of unity for the tungsten globes.

q_{el} = (12,000 × 1.20 + 4000) × 3.41 = 62,700 Btu per hr.

For the *fan motor*, use Equation 16 with a load factor of unity, and omit term *Motor Efficiency* because the motor is not within the space.

q_{em} = 7.5 × 2544 = 19,100 Btu per hr.

Moisture Permeation, Miscellaneous Allowance, and the Load-Lag Estimate:

Moisture permeation will be negligible, since this is a comfort job with a good building construction.

There would be some heat gain in the ductwork, but this would not be great because of the short run involved. Practical judgment for this job would suggest that no adjustment for load lag need be made to the load as computed. (Refer to Fig. 4). While it is true that inside radiation forms an important part of the total heat gain, it is advisable to be conservative in recognizing the effect of the large, flat, hot roof on the *comfort sensations of the occupants*. Radiation from the relatively low ceiling, augmented by heat absorption from the lighting fixtures, would produce a sensation of warmth in excess of the nominal effective temperature (see Chapter 6) established by the wet-bulb and dry-bulb temperatures. Hence, it is not desirable to take advantage of every small decrease possible in the peak design load, especially since the peak occurs in mid-afternoon when everything would be rather well warmed.

Total Loads and Required Air Quantity through Conditioning Equipment:

The total loads are summarized in the following table:

SUMMARY OF TOTAL LOADS—EXAMPLE 11

LOAD COMPONENT	SENSIBLE BTU/HR	LATENT BTU/HR
All walls, roof and doors	78,500	....
Glass areas	5,970	....
Infiltration 67 cfm	1,085	2,270
Ventilation air (1275 cfm × 0.15)	3,100	6,480
Occupants	17,000	21,250
Lighting	62,700	....
Motor, fan	19,100	....
Space load	187,455	30,000
Ventilation 1275 cfm × (1–0.15)	17,600	36,715
Totals	205,055	66,715
Grand total sensible and latent		271,770

Compute the enthalpy difference ratio from Equation 18.

$$\frac{h_i - h_s}{W_i - W_s} = \frac{(187{,}455 + 30{,}120)}{30{,}120} \times 1076 = 7770.$$

From the ASHAE PSYCHROMETRIC CHART, Chapter 3, determine that the apparatus dew point is 53.9 F.

Compute the effective air quantity (Equation 19). Then,

$$Q_{es} = \frac{187{,}455}{1.08(80 - 53.9) \times 0.85} = 7830 \text{ cfm.}$$

(Refer to Chapter 23 for coil selection.)

From note under Equation 19 the dry-bulb range will be (80 − 53.9) × 0.85 = 22.2 deg, and the dry-bulb temperature of air leaving the coil will be 80 − 22.2 = 57.8 F. The dry-bulb temperature leaving the fan (including the heat supplied by the fan motor) or, delivered into the room, will be (from Equation 21):

$$t_s = 80 - \frac{187{,}455 - 19{,}100}{1.08 \times 7830} = 80 - 19.9 = 60.1 \text{ F.}$$

With good distribution and diffusion, this temperature should not produce objectionable drafts.

The various calculations for the sensible, latent, and total heat loads for Example 11 may be summarized as follows:

EXAMPLE 11: SUMMARY

OUTDOOR CONDITIONS	95 DB	78 WB	0.0168 HUMIDITY RATIO
SPACE CONDITIONS	80 DB	65 WB	0.0098 HUMIDITY RATIO
DIFFERENCE	15		0.0070

	Btu/Hr.	
SENSIBLE LOAD		
Transmission		
Roof 4000 sq ft × 53° × 0.34 =	72,000	
S. wall 405 sq ft × 6° × 0.41 =	995	
E. wall 765 sq ft × 11° × 0.52 =	4,380	
N. wall ex. 170 sq ft × 3° × 0.52 =	265	
N. & W. party wall 1065 sq ft × 2° × 0.26 =	550	
Floor none		
Door 35 sq ft × 15 × 0.59 =	310	
All glass and rest of doors =	3,020	
Solar Radiation		
S. glass 60 sq ft × 13 =	780	
S. glass (doors) 35 sq ft × 42 =	1,470	
E. glass (doors) 18 sq ft × 14 =	250	
N. glass 30 sq ft × 15 =	450	
Internal Load		
Infiltration 67 cfm × 1.08 × 15° =	1,085	
Ventilation 1275 cfm × 1.08 × 15° × 0.15 =	3,100	
Lights (12,000 × 1.20 + 4000) 3.41 =	62,700	
People 85 × 200 =	17,000	
Motor, fan 7.5 hp × 2544 =	19,100	
TOTAL SENSIBLE SPACE LOAD		187,455
LATENT LOAD		
Infiltration 67 cfm × 4840 × 0.0071 =	2,270	
Ventilation 1275 cfm × 4840 × 0.0071 × 0.15	6,480	
People 85 × 250 =	21,250	
TOTAL LATENT SPACE LOAD		30,000
VENTILATION AIR WHICH DOES NOT BECOME PART OF SPACE LOAD		
Sensible 1275 cfm × 1.08 × 15° × (1-0.15) =		17,600
Latent 1275 cfm × 4840 × 0.0071 × (1–0.15) =		36,715
GRAND TOTAL LOAD		271,770

LETTER SYMBOLS USED IN CHAPTER 13

α = fraction of incident solar radiation absorbed, dimensionless; subscripts D, d, and t refer to direct, diffuse, and total, respectively.

β = solar altitude, degrees.

γ = wall solar azimuth, degrees.

ϵ = emissivity, dimensionless.

θ = incident angle, degrees.

τ = fraction of incident solar radiation transmitted, dimensionless.

Subscripts D, d, and t refer to direct, diffuse, and total, respectively.

ϕ = solar azimuth, degrees.

ψ = wall azimuth, degrees.

A = area across which heat is being transferred, square feet.

b = fraction of air passing through coil which does not contact surfaces, coil bypass factor.

f = unit surface conductance, Btu per (hour) (square foot) (Fahrenheit degree).

Subscripts c, r, o, and i refer to convection, radiation, outdoor, and indoor, respectively.

G_f = fraction of total window area receiving direct solar radiation when shaded by window reveal, dimensionless.

h = enthalpy of air per pound of dry air, Btu per pound. Subscripts i, o, and s refer to indoor, outdoor, and supply air, respectively.

I = incident solar radiation, Btu per (hour) (square foot). Subscripts D, d, Dn, and t refer to direct, diffuse, direct normal, and total solar radiation, respectively.

K = cosine of angle of incidence for direct solar radiation striking a surface, dimensionless.

k = thermal conductivity of building material, Btu per (square foot) (hour) (Fahrenheit degree per inch).

l = height of window, feet.

M = the permeance of the specimen in *perms* or grains per (square foot) (hour) (inch of mercury vapor pressure difference).

Q = rate of entry of outdoor air, cubic feet per minute.

Q_{es} = required air quantity through conditioning equipment, cubic feet per minute.

q = instantaneous rate of heat transfer, Btu per hour.

q_e = instantaneous latent heat load, Btu per hour.

q_{ei} = instantaneous space latent ventilation load, Btu per hour.

q_{es} = instantaneous latent ventilation load which does not become a part of space load, Btu.

q_m = latent heat load due to moisture transmission through materials, Btu per (hour) (square foot).

q_s = instantaneous sensible heat load, Btu per hour.

q_{si} = instantaneous space sensible ventilation load, Btu per hour.

q_{ss} = instantaneous sensible ventilation load which does not become a part of space load, Btu per hour.

q_t = $q_s + q_e$, also $q_{ei} + q_{es} + q_{si} + q_{ss}$, Btu per hour.

R_r = low temperature radiant energy received from outdoor surroundings (does not include solar radiation), Btu per (hour) (square foot of receiving surface).

R = radiant energy emitted by a black body, Btu per (hour) (square foot). Subscripts go and L refer to outdoor surfaces of glass and building, respectively.

S = rate of heat storage within a glass section, Btu per (hour) (square foot).

t_e = sol-air temperature, Fahrenheit.

t_{gi} = temperature of indoor glass surface, Fahrenheit.

t_{go} = temperature of outdoor glass surface, Fahrenheit.

t_i = indoor air temperature, Fahrenheit.

t_m = 24-hr cyclic average sol-air temperature, Fahrenheit.

t_o = outdoor air temperature, Fahrenheit.

t_s = room supply air dry-bulb temperature, Fahrenheit.

U = overall coefficient of heat transfer of a structural section, Btu per (square foot) (hour) (Fahrenheit degree).

v_o = volume of outdoor air per pound of dry air, cubic feet.

w = width of window, feet.

W = humidity ratio, pounds moisture per pound of dry air. Subscripts i, o, and s refer to indoor, outdoor, and supply air, respectively.

REFERENCES

[1] *Application Engineering Standards for Air Conditioning for Comfort* (Air Conditioning and Refrigerating Machinery Association, Inc., 1947, p. 4).

[2] W. V. Consolazio and L. J. Pecora: Minimal replenishment air required for living spaces (ASHVE TRANSACTIONS, Vol. 53, 1947, p. 127).

[3] *Recommended Safe Practice of the NBFU for Hospital Operating Rooms* (*National Board of Fire Underwriters Pamphlet* No. 56).

[4] P. Moon: Proposed standard solar radiation curves for engineering use (*Journal of the Franklin Institute*, Vol. 230, November 1940, p. 583).

[5] C. O. Mackey: ASHVE RESEARCH REPORT No. 1268—Summer weather data and sol-air temperature—Study of data for Lincoln, Nebr. (ASHVE TRANSACTIONS, Vol. 51, 1945, p. 93).

[6] C. O. Mackey and E. B. Watson: Summer weather data and sol-air temperature—Study of data for New York City (ASHVE TRANSACTIONS, Vol. 51, 1945, p. 75).

[7] G. A. Hendrikson and J. H. Walker: Summer cooling for comfort as affected by solar radiation (*Heating and Ventilating*, Vol. 29, November 1932, p. 14).

[8] *Tables of Computed Altitude and Azimuth* (U. S. Navy Dept. *Hydrographic Office Bulletin* No. 214, Vol. 1–9, 1940).

[9] *The American Nautical Almanac* (U. S. Naval Observatory, annual).

[10] F. C. Houghten, J. L. Blackshaw, E. M. Pugh, and Paul McDermott: ASHVE RESEARCH REPORT No. 923—Heat transmission as influenced by heat capacity and solar radiation (ASHVE TRANSACTIONS, Vol. 38, 1932, p. 231). J. S. Alford, J. E. Ryan, and F. O. Urban: Effect of heat storage and variation in outdoor temperature and solar intensity on heat transfer through walls (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 369). Victor Paschkis: Periodic heat flow in building walls determined by electrical analogy method (ASHVE TRANSACTIONS, Vol. 48, 1942, p. 75). C. O. Mackey and L. T. Wright, Jr.: Periodic heat flow—Homogeneous walls or roofs (ASHVE TRANSACTIONS, Vol. 50, 1944, p. 293). C. O. Mackey and L. T. Wright, Jr.: Periodic heat flow—Composite walls or roofs (ASHVE TRANSACTIONS, Vol. 52, 1946, p. 283). H. A. Johnson: Periodic heat transfer at the inner surface of a homogeneous wall (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 143).

[11] F. C. Houghten, Carl Gutberlet, and A. A. Rosenburg: The effect of solar radiation on the heat transmission through walls (*American Society of Testing Materials Symposium on Thermal Insulating Materials*, 1939).

[12] F. C. Houghten, H. T. Olson, and Carl Gutberlet: ASHVE RESEARCH REPORT No. 1157—Summer cooling load as affected

by heat gain through dry, sprinkled and water covered roofs (ASHVE TRANSACTIONS, Vol. 46, 1940, p. 237).

[13] F. C. Houghten, Carl Gutberlet, and A. J. Wahl: ASHVE RESEARCH REPORT No. 1002—Cooling requirements of single rooms in a modern office building (ASHVE TRANSACTIONS, Vol. 41, 1935, p. 53).

[14] J. N. Livermore: Study of actual vs. predicted cooling load on an air conditioning system (ASHVE TRANSACTIONS, Vol. 49, 1943, p. 287).

[15] F. C. Houghten, E. C. Hach, S. I. Taimuty, and Carl Gutberlet: ASHVE RESEARCH REPORT No. 1195—Heat gain through walls and roofs as affected by solar radiation (ASHVE TRANSACTIONS, Vol. 48, 1942, p. 91).

[16] J. P. Stewart: Solar heat gain through walls and roofs for cooling load calculations (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 361).

[17] G. V. Parmelee and W. W. Aubele: ASHVE RESEARCH REPORT No. 1442—Radiant energy emission of atmosphere and ground (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 85).

[18] D. Brunt: Radiation in the atmosphere (*Supplement to the Quarterly Journal of the Royal Meteorological Society*, Vol. 66, 1940).

[19] G. V. Parmelee, W. W. Aubele, and R. G. Huebscher: ASHVE RESEARCH REPORT No. 1333—Measurements of solar heat transmission through flat glass (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 165).

[20] G. V. Parmelee and W. W. Aubele: ASHVE RESEARCH REPORT No. 1348—Solar and total heat gain through double flat glass (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 407).

[21] G. V. Parmelee and W. W. Aubele: ASHVE RESEARCH REPORT No. 1374—Solar energy transmittance of eight-inch hollow glass block (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 435).

[22] G. V. Parmelee and W. W. Aubele: ASHVE RESEARCH REPORT No. 1399—Heat flow through unshaded glass: Design data for use in load calculations (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 371).

[23] G. V. Parmelee and W. W. Aubele: ASHVE RESEARCH REPORT No. 1417—Solar energy transmittance of figured rolled glass (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 209).

[24] D. J. Vild and G. V. Parmelee: ASHAE RESEARCH REPORT No. 1560—Heat gain through glass skylight fenestrations (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 89).

[25] G. V. Parmelee and D. J. Vild: ASHVE RESEARCH REPORT No. 1485—Design data for slat-type sun shades for use in load estimating (ASHVE TRANSACTIONS, Vol. 59, p. 403). G. V. Parmelee, W. W. Aubele, and D. J. Vild: ASHVE RESEARCH REPORT No. 1474—The shading of sunlit glass: An experimental study of slat-type sun shades (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 221). G. V. Parmelee and W. W. Aubele: ASHVE RESEARCH REPORT No. 1460—The shading of sunlit glass: An analysis of the effect of uniformly spaced flat opaque slats (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 337).

[26] C. S. Leopold: The mechanism of heat transfer, panel cooling, heat storage (*Refrigerating Engineering*, July 1947, p. 33). C. S. Leopold: Hydraulic analogue for the solution of problems of thermal storage, radiation, convection and conduction (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 389).

[27] C. O. Mackey and N. R. Gay: Heat gains are not cooling loads (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 413).

[28] C. O. Mackey and N. R. Gay: Cooling load from sunlit glass (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 321).

[29] See Reference 1, p. 8.

[30] C. M. Ashley: Psychrometric factors in the air conditioning estimate (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 91).

[31] W. G. Darley: Cooler footcandles for air conditioning (ASHVE TRANSACTIONS, Vol. 46, 1940, p. 367). IES-ASHVE Joint Committee on Lighting and Air Conditioning: Lighting and air conditioning design factors (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September 1941, p. 605). H. M. Sharp: Lighting and air conditioning (*Heating and Ventilating*, November 1942, p. 35).

[32] Compiled by J. P. Stewart from various sources.

CHAPTER 14

RADIATORS, CONVECTORS, BASEBOARD AND FINNED-TUBE UNITS

Definitions, Heat Emission, Radiators, Convectors, Baseboard, Finned-Tube, Ratings, Corrections for Non-Standard Conditions, Enclosed Radiators

RADIATORS, convectors, baseboard, and finned-tube are the types of heat-distributing units used in steam and hot water heating systems to supply heat to a room by radiation and convection. The function of these devices is the maintenance of the desired mean radiant and air temperatures in the area. Since heat losses through the various parts of the structure constantly tend to lower these temperatures, heat distributing units should be so placed and regulated that their output will replace the losses when and where they occur. If 80 percent of the room heat loss occurs through a cold wall or window area, then 80 percent of the input should be introduced in or directed toward that area.

The term *radiator* is generally confined to sectional cast-iron radiation. Cast-iron radiator types may be column, large-tube, small-tube, or wall.

The term *convector* refers to a heat-distributing unit that operates with gravity-recirculated room air, is surrounded on all sides by an enclosure having an air-inlet opening below the heating element, and an air-outlet opening above the heating element.

The terms *baseboard* and *baseboard radiation* refer to steam or water heat-distributing units designed for installation along the bottom of the walls replacing the conventional baseboard. They operate with gravity-recirculated room air, and have a substantial portion of their frontal face surface directly exposed to the room.

The term *finned-tube* refers to steam or water heat-distributing units fabricated from metallic tubing with metallic fins bonded to the tube. They operate with gravity-recirculated room air and are designed for installation without enclosure, or with open-type grilles or covers, or with enclosures having top, front, or inclined outlets.

HEAT EMISSION

Steam or water heat-distributing units emit heat by radiation and convection. These heat-transfer processes and the factors that affect them are discussed in detail in Chapter 5. In general, those units having a large portion of their heated surface exposed emit a larger portion of heat by radiation than do units having their heating surfaces completely or partly concealed from view.

The output of a heat-distributing unit is expressed in units of Btu per hr (Btuh), 1000 Btu per hr (Mbh), or in square feet equivalent direct radiation (EDR) (e.g., 240 Btuh for steam = 1 sq ft EDR).

RADIATORS

Column and large-tube radiators are no longer manufactured, but since many of these units are still in use, Tables 1 and 2 are included to provide principal dimensions and average ratings.

The *small-tube type radiators*, with a spacing of 1¾ in. per section, occupy less space than the older column and large-tube radiators, and are particularly suited for installation in recesses.

After a study of the demand for various sizes of radiators, the Institute of Boiler and Radiator Manufacturers, in cooperation with the Division of Simplified Practice, National Bureau of Standards, established *Simplified Practice Recommendation* R174-47, *Cast Iron Radiators*, for small-tube cast-iron radiators. Table 3 shows the size and dimensions now being manufactured.

Wall radiators are hung from wall brackets and are well adapted to use in factory buildings. Tests have shown that the heat emitted from a wall-type radiator may be reduced from 5 to 10 percent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. Installation of wall radiators at the ceiling is not recommended because the resulting large temperature gradients between floor and ceiling make it difficult to heat the living zone satisfactorily. Dimensions and heat emission rates of wall radiators for normal wall installations are given in Table 4.

Pipe Coils

Pipe coils are sometimes used in factory buildings and are usually placed under windows or along heavily-exposed walls. The heat emission of such pipe coils may be obtained from Table 5.

CONVECTORS

Convectors are made in a wide variety of depths, sizes, lengths, and enclosure or cabinet types. The basic sizes and types are listed in *Simplified Practice Recommendation* 238-50, *Convectors*. The heating elements are available in fabricated ferrous and non-ferrous metals as well as cast iron. The air enters the enclosure below the heating element, is heated in passing through the element, and leaves the enclosure through the outlet grille located above the heating element. Factory-assembled units comprised of a heating element and enclosure are widely used. These may be free standing, wall hung, or recessed, (Fig. 1) and may have outlet grilles and arched inlets or inlet grilles as desired.

In cases where cabinets or enclosures are to be used but are not furnished by the manufacturer, it is important that the proportions of the cabinets or enclosures and the grilles be so designed that they will not impair the performance of the assembled convector. It is desirable that the cabinet or

Table 1 . . . . Column-Type Cast-Iron Radiator

Height In.	Generally Accepted Rating per Section[a]					
	One Column		Two Column		Three Column	
	Sq Ft	Btuh	Sq Ft	Btuh	Sq Ft	Btuh
15			1½	360		
18					2¼	540
20	1½	360	2	480		
22			2¼	540	3	720
23	1⅝	400	2½	560		
26	2	480	2⅝	640	3⅜	900
32	2½	600	3½	800	4½	1080
38	3	720	4	960	5	1200
45			5	1200	6	1440
	Four Column		Five Column		Six Column	
	Sq Ft	Btuh	Sq Ft	Btuh	Sq Ft	Btuh
13					3	720
16					3¾	900
18	3	720	4⅜	1120	4½	1080
20					5	1200
22	4	960				
26	5	1200	7	1680		
32	6½	1560				
38	8	1920	10	2400		
45	10	2400				

[a] These ratings are based on steam at 215 F and air at 70 F. They apply only to installed radiators exposed in a normal manner; not to radiators installed behind enclosures, grilles, or under shelves. For Btu per hour ratings at other temperatures, divide table values by factors found in Table 7.

enclosure for the convector fit as snugly as possible so that the air passing through cannot bypass the heating element.

BASEBOARD UNITS

Baseboard heat-distributing units are divided into three types: (1) radiant, (2) radiant-convector, and (3) finned-tube.[1]

Radiant-type baseboard is made of cast iron or steel. This type of unit emits a large portion of its heat output by radiation. Units are sometimes suspended from the ceiling in rooms having little or no available wall space for floor mounted units. As the convected heat obtained from these units when mounted at the ceiling is less than when they are mounted at floor level additional length must be added as recommended by the manufacturer to allow for the reduction.

Radiant-convector type baseboard also is made of cast iron or steel. The units are provided with air openings at the top and bottom to permit circulation of room air over the wall side of the unit. The wall side of the unit has extended surface to provide increased heat output. A large portion of the heat emitted is transferred by convection. This type of baseboard having a greater output per linear foot than the radiant type is particularly adaptable where wall space is at a premium or the heat loss of the room is high.

Finned-tube type of baseboard has a finned-tube heating element that is concealed by a long low sheet-metal enclosure or cover. A major portion of the heat is transferred to the room by convection. The output varies over a wide range depending on the physical dimensions and the materials used. When selecting this type of baseboard the designer should avoid using a unit with too high an output per linear foot. Baseboard performs best when units are so selected that they are installed along as much of the exposed wall as possible.

The basic advantage of the baseboard heat-distributing unit is that its normal placement is along the cold walls and under areas where the greatest heat loss occurs. Other advantages claimed for the baseboard heat-distributing unit are: it is inconspicuous; it offers a minimum of interference with furniture placement; and it distributes the heat near the floor. This last characteristic reduces the floor-to-ceiling temperature gradient to about 2 to 4 F deg, and tends to produce uniform temperatures throughout the room. It also makes baseboard heat-distributing units adaptable to basementless homes, where cold floors are prevalent.[2]

Table 2 . . . . Large-Tube Cast-Iron Radiators

Sectional, cast-iron, tubular-type radiators of the large-tube pattern, that is, having tubes approximately 1⅝ in. in diameter, 2½ in. on centers.

Number of Tubes per Section	Catalog Rating per Section[a]		Height	Width	Section Center Spacing[b]	Leg Height[c] to Tapping
	Sq Ft	Btuh	In.	In.	In.	In.
3	1¾	420	20	4⅝	2½	4½
	2	480	23		2½	4½
	2½	560	26		2½	4½
	3	720	32		2½	4½
	3½	840	38		2½	4½
4	2¼	540	20	6¼–6 13/16	2½	4½
	2½	600	23		2½	4½
	2¾	660	26		2½	4½
	3½	840	32		2½	4½
	4¼	1020	38		2½	4½
5	2⅝	640	20	8–8 9/16	2½[d]	4½
	3	720	23		2½[d]	4½
	3½	840	26		2½[d]	4½
	4⅓	1040	32		2½[d]	4½
	5	1200	38		2½[d]	4½
6	3	720	20	9–10 3/8	2½	4½
	3½	840	23		2½	4½
	4	960	26		2½	4½
	5	1200	32		2½	4½
	6	1440	38		2½	4½
7	2½	600	14	11 3/8–12 13/16	2½	3
	3	720	17		2½	3
	3⅔	880	20		2½	3 or 4½

[a] These ratings are based on steam at 215 F and air at 70 F. They apply only to installed radiators exposed in a normal manner; not to radiators installed behind enclosures, grilles, or under shelves. For Btu per hour ratings at other temperatures, divide table values by factors found in Table 7.

[b] Maximum assembly 60 sections. Length equals number of sections times 2½ in.

[c] Where greater than standard leg heights are required, this dimension shall be 6 in., except for 7-tube sections, in heights from 13 to 20 in., inclusive, for which this dimension shall be 4½ in. Radiators may be furnished without legs.

[d] For 5-tube hospital-type radiation, this dimension is 3 in.

Table 3 . . . . Small-Tube Cast-Iron Radiators

Number of Tubes per Section	Catalog Rating per Section[a]		Section Dimensions				
			A Height[c]	B Width		C Spacing[b]	D Leg Height[c]
				Min	Max		
	Sq Ft	Btuh	In.	In.	In.	In.	In.
3[d]	1.6	384	25	3¼	3½	1⅜	2½
4[d]	1.6	384	19	4 7/16	4 13/16	1¾	2½
	1.8	432	22	4 7/16	4 13/16	1¾	2½
	2.0	480	25	4 7/16	4 13/16	1¾	2½
5[d]	2.1	504	22	5 5/8	6 9/16	1¾	2½
	2.4	576	25	5 5/8	6 9/16	1¾	2½
6[d]	2.3	552	19	6 13/16	8	1¾	2½
	3.0	720	25	6 13/16	8	1¾	2½
	3.7	888	32	6 13/16	8	1¾	2½



[a] These ratings are based on steam at 215 F and air at 70 F. They apply only to installed radiators exposed in a normal manner, not to radiators installed behind enclosures, grilles, or under shelves. For Btu per hour ratings at other temperatures, divide table values by factors found in Table 7.

[b] Length equals number of sections times 1¾ in.

[c] Overall height and leg height, as produced by some manufacturers, are one inch greater than shown in Columns *A* and *D*. Radiators may be furnished without legs. Where greater than standard leg heights are required this dimension shall be 4½ in.

[d] Or equal.

Table 4 . . . . Cast-Iron Wall Radiators

Approximate Dimensions—Inches			Heat Output[a]	
Height	Length or Width	Thickness	Sq Ft	Btuh
13½	16½	3	6½	1560
13¼	22	3	8	1920
22	13¼	3	8	1920
13¼	29	3	11	2640
29	13¼	3	11	2640

[a] These ratings are based on steam at 215 F and air at 70 F. They apply only to installed radiators exposed in a normal manner, not to radiators installed behind enclosures, grilles, or under shelves. For Btu per hour ratings at other temperatures divide table values by factors found in Table 7.

Table 5 . . . . Heat Emission of Pipe Coils Placed Vertically on a Wall (Pipes Horizontal) Containing Steam at 215 F and Surrounded with Air at 70 F

Btu per linear foot of coil per hour (not linear feet of pipe)

Size of Pipe	1 In.	1¼ In.	1½ In.
Single row	132	162	185
Two	252	312	348
Four	440	545	616
Six	567	702	793
Eight	651	796	907
Ten	732	907	1020
Twelve	812	1005	1135

Heat loss calculations for baseboard heating systems are the same as those used for other types of heat-distributing units. The procedure for designing baseboard heating systems is given in *I = B = R Installation Guide* No. 5.[1] Ratings for baseboard heat-distributing units are expressed in Btuh per linear foot.

FINNED-TUBE UNITS

A finned-tube heat-distributing unit is a room-air heater composed of a finned-tube element fabricated from metallic tube to which metallic fins have been bonded. It does not include a fin-tube type baseboard element provided with an enclosure for replacement of the conventional baseboard.

The finned-tube unit can be used with either steam or hot water systems. It has advantages for installation where it is desired to distribute the heat along the entire outside wall and thereby prevent down drafts along the walls in buildings such as schools, churches, hospitals, and factories.

Normal placement of finned tube is along the walls where the heat loss is greatest. If necessary, the units can be installed in two or three tiers along available wall space to meet the heating requirements. For hot water system installations where two or three tiers are required, a sinuous water flow through the heaters is recommended because a header connection with parallel flow may permit the water to short circuit along the path of least resistance.

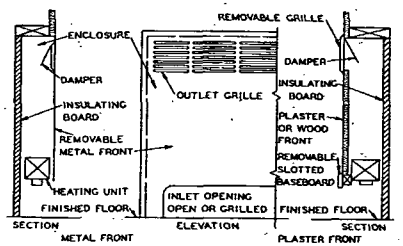


Fig. 1 . . . . Typical Recessed Convector

Protection of the heating element may be provided by open-type grilles fabricated of expanded metal or perforated materials covering the top and front of the finned-tube element. Covers and enclosures are also available to enhance the appearance and to increase heating efficiency.

A *cover* is a fabricated shield having at least a portion of the front skirt made of solid material. It can be mounted on the finned-tube element so that there is clearance between the wall and the cover, and the rear of the finned-tube element is not completely enclosed. A cover may have a top, front, or inclined outlet.

An *enclosure* is a shield of solid material installed so that the finned-tube element is completely enclosed at both front and rear. An enclosure may have an integral back or may be installed tightly against the wall so that the wall itself forms the back. An enclosure may have a top, front, or inclined outlet.

Finned-tube units are available in four tube sizes from 1 in. to 2 in. IPS with various fin arrangements. The resistance to the flow of water or steam is rather low so that a

separate system of distribution piping is generally not required.

The heating capacity of finned tube varies over a wide range depending on tube size, fin size and thickness, spacing of fins, and the materials used. A major portion of the heat transferred to the room to be heated is by convection.

The heating capacity of finned-tube heat-distributing units is determined by test. The Institute of Boiler and Radiator Manufacturers has developed a testing and rating code for finned-tube type of radiation.[4]

Heat loss calculations for finned-tube heating systems are the same as those used for other types of radiation. Ratings are expressed in Btuh per linear foot or square feet Equivalent Direct Radiation per linear foot for steam, and Btuh per linear foot for water.

RATINGS OF HEAT-DISTRIBUTING UNITS

Radiators

A standard method of testing radiators was adopted by the ASHAE in 1927.[4] This Code was withdrawn but the method is still used in industry. It provides for a standard test room, the temperature of which is to be maintained at 70 F, measured in the center of the room at an elevation of 5 ft above the floor. The steam temperature in the radiator is to be 215 F, which corresponds to 15.6 psia. The weight of condensate per hour, under these standard conditions, multiplied by the difference in the enthalpy of the steam entering the radiator and that of the condensate leaving the radiator, gives the radiator output in Btu per hour (Btuh).

If the rating in square feet Equivalent Direct Radiation is desired, divide this output by 240 (which by definition is the Btuh equivalent of 1 sq ft EDR). Correction of output from non-standard test conditions to output at 70 F room and 215 F steam may be made by multiplying the output under test by the ratio

$$\frac{(\text{Standard temperature difference, steam} - \text{air})^{1.3}}{(\text{Test temperature difference, steam} - \text{air})^{1.3}}$$

Convectors

The generally accepted method of testing and rating both ferrous and non-ferrous convectors, is given in *Commercial Standard* CS140-47, *Testing and Rating Convectors*,[5] which has been developed cooperatively by the Convector Manufacturers Association, the Institute of Boiler and Radiator Manufacturers, other members of the trade, and the National Bureau of Standards. This *Commercial Standard* contains details covering construction and instrumentation of the test booth or room and procedures for determining both steam and water ratings.

Steam ratings are expressed in square feet EDR and Btu per hour (Btuh). Water ratings are expressed in Btuh at specified water temperature drops and average water temperatures.

Under the provisions of *Commercial Standard* CS140-47 the rating of a top outlet convector is established at a value not in excess of the test capacity (which is the heat extracted from the steam or water in the convector under standard test conditions). For convectors with other types of enclosures or cabinets a percentage that varies up to a maximum of 15 percent depending on the height and type of enclosure or cabinet is added for heating effect.[6, 7] The additions made for heating effect must be shown in the manufacturer's literature.

Table 6 .... Factors to Convert I=B=R Finned-Tube Steam Ratings to Hot Water Ratings at Temperatures Indicated

Average Radiator Temperature	Factor	Average Radiator Temperature	Factor
150	0.45	190	0.78
155	0.49	195	0.82
160	0.53	200	0.86
165	0.57	205	0.91
170	0.61	210	0.95
175	0.65	215	1.00
180	0.69	220	1.05
185	0.73		

Baseboard

The generally accepted method of testing and rating baseboards is covered in the *I = B = R Testing and Rating Code for Baseboard Type of Radiation*.[8] This Code contains details covering construction and instrumentation of the test booth or room, procedures for determining steam ratings, and licensing provisions for the obtaining of approval of these ratings.

Baseboard ratings include an allowance for heating effect of 15 percent, added to the test capacity. The addition made for heating effect must be shown in the manufacturer's literature.

Steam ratings are expressed in Btuh per linear foot or square feet EDR per linear foot. Water ratings are determined from steam ratings and are expressed in Btuh per linear foot, at specified water flow rates and average water temperatures.

Finned-Tube

The generally accepted method of testing and rating finned-tube heat-distributing units is covered in the *I = B = R Testing and Rating Code for Finned-Tube Type of Radiation*.[9] This Code contains details covering construction and instrumentation of the test booth or room, procedures for determining steam and water ratings, and licensing provisions for the obtaining of these ratings. Steam ratings are expressed in Btuh per linear foot or square feet EDR per linear foot. Water ratings are determined from steam ratings and are expressed in Btuh per linear foot at specified water flow rates and average water temperatures.

The rating of a finned-tube unit in an enclosure having a top outlet is established at a value not in excess of the test capacity (which is the heat extracted from the steam or water in the unit under standard test conditions). For finned-tube with other types of enclosures or covers a percentage is added for heating effect which varies up to a maximum of 15 percent depending on the height and type of enclosure or cover. The additions made for heating effect must be shown in the manufacturer's literature.

Corrections for Non-Standard Conditions

The heat output of a radiator, convector, baseboard or finned-tube heat-distributing unit is an exponential function of the temperature difference between the air in the room and the heating medium in the room-heating unit, or, expressed as an equation

$$H = c(t_s - t_a)^n \qquad (1)$$

where

H = heat output, Btu per hour (Btuh).

c = a constant determined by test from Equation 1.

t_s = average temperature of heating medium, Fahrenheit. For hot water the arithmetical average of the entering and leaving water temperatures is used.

t_a = room air temperature, Fahrenheit. Air temperature 60 in. above the floor is generally used for radiators while the entering air temperature is used for all other types of heating units.

n = an exponent which equals 1.3 for cast-iron radiators, 1.4 for baseboard radiation, and 1.5 for convectors. For finned-tube units n varies with both air and heating medium temperatures. Correction factors to convert outputs at standard rating conditions to outputs at other conditions are given in Tables 6, 7, and 8.

Table 7 .... Correction Factors for Direct Cast-Iron Radiators and Convectors[a]

Steam Press. (Approx.) Gage Vacuum In. Hg	Steam Press. (Approx.) Abs Lb per Sq In.	Heating Medium Temp. F Steam or Water	Factors for Direct Cast-Iron Radiators — Room Temperature F 80	75	70	65	60	55	50
22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62
20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44
17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28
14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15
10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05
6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96
Lb per Sq In.									
1	15.6	215	1.10	1.05	1.00	0.96	0.92	0.88	0.85
6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76
15	30	250	0.81	0.78	0.76	0.73	0.70	0.68	0.66
27	42	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49

Steam Press. (Approx.) Gage Vacuum In. Hg	Steam Press. (Approx.) Abs Lb per Sq In.	Heating Medium Temp. F Steam or Water	Factors for Convectors — Inlet Air Temperature F 80	75	70	65	60	55	50
22.4	3.7	150	3.14	2.83	2.57	2.35	2.15	1.98	1.84
20.3	4.7	160	2.57	2.35	2.15	1.98	1.84	1.71	1.59
17.7	6.0	170	2.15	1.98	1.84	1.71	1.59	1.49	1.40
14.6	7.5	180	1.84	1.71	1.59	1.49	1.40	1.32	1.24
10.9	9.3	190	1.59	1.49	1.40	1.32	1.24	1.17	1.11
6.5	11.5	200	1.40	1.32	1.24	1.17	1.11	1.05	1.00
Lb per Sq In.									
1	15.6	215	1.17	1.11	1.05	1.00	0.95	0.91	0.87
6	21	230	1.00	0.95	0.91	0.87	0.83	0.79	0.76
15	30	250	0.83	0.79	0.76	0.73	0.70	0.68	0.65
27	42	270	0.70	0.68	0.65	0.63	0.60	0.58	0.56
52	67	300	0.56	0.54	0.53	0.51	0.49	0.48	0.47

[a] To determine the size of a radiator or a convector for a given space, multiply the heat loss of the space in Btu per hour by the proper factor from the above table and select radiator or convector having an equivalent Btu per hour rating.

An alternate method is to divide the heat loss in Btu per hour by 240 and multiply the result by the proper factor from the above table and select radiator or convector having an equivalent square foot rating.

To determine the heating capacity of a radiator or a convector under conditions other than the basic ones with the heating medium at a temperature of 215 F, and the room temperature at 70 F in the case of a radiator, and the inlet air temperature at 65 F in the case of a convector, divide the heating capacity at the basic rating conditions by the proper factor from the above table.

Table 8 .... Correction Factors for Finned-Tube and Baseboard Radiation[a]

Steam Pressure (Approx.) Gage Vacuum In. Hg	Steam Pressure (Approx.) Abs Lb per Sq In.	Heating Medium Temp. F Steam or Water	Factors for Finned-Tube — Inlet Air Temperature F 80	75	70	65	60	55	50
22.4	3.7	150	2.80	2.50	2.20	1.95	1.81	1.67	1.54
20.3	4.7	160	2.34	2.14	1.94	1.75	1.62	1.50	1.37
17.7	6.0	170	2.01	1.86	1.70	1.56	1.46	1.36	1.26
14.6	7.5	180	1.76	1.65	1.53	1.42	1.32	1.24	1.15
10.9	9.3	190	1.53	1.45	1.36	1.28	1.20	1.12	1.03
6.5	11.5	200	1.38	1.31	1.24	1.16	1.09	1.02	0.95
Lb per Sq In.									
1	15.6	215	1.17	1.12	1.06	1.00	0.95	0.90	0.85
6	21	230	1.00	0.97	0.93	0.90	0.86	0.82	0.78
15	30	250	0.88	0.85	0.82	0.78	0.75	0.71	0.67
27	42	270	0.75	0.73	0.70	0.68	0.65	0.62	0.59
52	67	300	0.62	0.60	0.58	0.56	0.54	0.52	0.50

Steam Pressure (Approx.) Gage Vacuum In. Hg	Steam Pressure (Approx.) Abs Lb per Sq In.	Heating Medium Temp. F Steam or Water	Factors for Baseboard — Inlet Air Temperature F 80	75	70	65	60	55	50
22.4	3.7	150	2.86	2.61	2.38	2.20	2.03	1.89	1.76
20.3	4.7	160	2.38	2.20	2.03	1.89	1.76	1.64	1.56
17.7	6.0	170	2.03	1.89	1.76	1.64	1.54	1.44	1.38
14.6	7.5	180	1.76	1.64	1.55	1.44	1.38	1.29	1.23
10.9	9.3	190	1.54	1.44	1.37	1.29	1.22	1.16	1.09
6.5	11.5	200	1.38	1.29	1.23	1.16	1.09	1.05	1.00
Lb per Sq In.									
1	15.6	215	1.16	1.10	1.05	1.00	0.95	0.92	0.88
6	21	230	1.00	0.96	0.92	0.88	0.84	0.81	0.77
15	30	250	0.86	0.82	0.79	0.76	0.73	0.70	0.68
27	42	270	0.73	0.70	0.68	0.66	0.63	0.61	0.59
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49

[a] To determine the heating capacity under conditions other than the basic ones with the heating medium at a temperature of 215 F, and the inlet air temperature at 65 F, divide the heating capacity at the basic rating conditions by the proper factor from the above table.

Enclosure, Paint, Humidity Effect

The general effect of an enclosure placed about a direct radiator is to restrict the air flow, and diminish the proportion of output due to radiation. Enclosures of proper design may, however, improve the heat distribution within the room as compared to the heat distribution obtained with an unenclosed radiator.[4, 8]

For a radiator or cast-iron baseboard the finish coat of paint affects the heat output. Oil paints of any color will give about the same results as unpainted black or rusty surfaces, but an aluminum or a bronze paint will reduce the heat emitted by radiation. The net effect may be a reduction of 10 percent or more in the total heat output of the radiator.[10, 11, 12]

Some commercial enclosures and shields for use on direct radiators are equipped with water pans for the purpose of

adding moisture to the air in the room. Tests[28] show that an average evaporative rate of about 0.235 lb per (hr) (sq ft of water surface) may be obtained from such pans, when a radiator is steam heated and the relative humidity in the room is between 25 and 40 percent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 percent on a zero day.

REFERENCES

[1] *I = B = R Baseboard Heating Systems Installation Guide* No. 5 (Institute of Boiler and Radiator Manufacturers, 1953, 2nd ed.).

[2] A. P. Kratz and W. S. Harris: *A Study of Radiant Baseboard Heating in the I = B = R Research Home* (University of Illinois, *Engineering Experiment Station Bulletin* No. 358, 1945).

[3] *I = B = R Testing and Rating Code for Finned-Tube Type Radiation* (Institute of Boiler and Radiator Manufacturers, 1951, 1st ed., with addenda, 1954).

[4] ASHVE code for testing radiators (ASHVE TRANSACTIONS, Vol. 33, 1927, p. 18). (This code was withdrawn in 1956.)

[5] Commercial Standard for Testing and Rating Convectors (U. S. Department of Commerce, CS 140-47, 1947).

[6] C. Brabbée: The heating effect of radiators (ASHVE TRANSACTIONS, Vol. 33, 1927, p. 33).

[7] A. C. Willard, A. P. Kratz, M. K. Fahnestock, and S. Konzo: Investigation of heating rooms with direct steam radiators equipped with enclosures and shields (ASHVE TRANSACTIONS, Vol. 35, 1929, p. 77 or University of Illinois, *Engineering Experiment Station Bulletin* No. 192). A. C. Willard, A. P. Kratz, M. K. Fahnestock, and S. Konzo: *Investigation of Various Factors Affecting the Heating of Rooms with Direct Steam Radiators* (University of Illinois, *Engineering Experiment Station Bulletin* No. 223).

[8] *I = B = R Testing and Rating Code for Baseboard Type of Radiation* (Institute of Boiler and Radiator Manufacturers, August 1952, 2nd ed.).

[9] E. A. Allcut: *Heat Output of Concealed Radiators* (University of Toronto, *School of Engineering Research Bulletin* No. 140, 1933).

[10] K. F. Rubert: *Heat Emission from Radiators* (Cornell University, *Engineering Experiment Station Bulletin* No. 24, 1937).

[11] W. H. Severns: Comparative tests of radiator finishes (ASHVE TRANSACTIONS, Vol. 33, 1927, p. 41).

[12] J. R. Allen: Heat loss from direct radiation (ASHVE TRANSACTIONS, Vol. 26, 1920, p. 11).

[13] A. P. Kratz: *Humidification for Residences* (University of Illinois, *Engineering Experiment Station Bulletin* No. 230, p. 20).

BIBLIOGRAPHY

A. C. Willard, A. P. Kratz, and M. K. Fahnestock: ASHVE RESEARCH REPORT No. 926—The application of the eupatheoscope for measuring the performance of direct radiators and convectors in terms of equivalent temperature (ASHVE TRANSACTIONS, Vol. 39, 1933, p. 303).

W. J. McConnell and C. P. Yaglogiou: The Kata thermometer—its value and defects (*U. S. Public Health Service Report*, Reprint No. 953, September 5, 1924, p. 2293).

C.-E. A. Winslow and Leonard Greenburg: The thermo-integrator—a new instrument for the observation of thermal interchanges (ASHVE TRANSACTIONS, Vol. 41, 1935, p. 149).

C.-E. A. Winslow, A. P. Gragge, Leonard Greenburg, I. M. Moriyama, and E. J. Rodee: The calibration of the thermo-integrator (*The American Journal of Hygiene*, July 1935, p. 137).

T. Bedford and C. G. Warner: The globe thermometer in studies of heating and ventilation (*The Journal of Hygiene*, Vol. 34, No. 4).

Residential baseboard radiation (*Reference Section, Air Conditioning, Heating and Ventilating*, November 1957, p. 81).

Commercial and industrial finned-tube radiation (*Reference Section, Air Conditioning, Heating and Ventilating*, December 1957, p. 71).

CHAPTER 15

UNIT VENTILATORS AND UNIT HEATERS

Unit Ventilators: Classification, Ratings, Capacity, Requirements, Application, Control, Location, Exhaust Vents, Window Downdraft Prevention, Mechanical Cooling; Unit Heaters: Classification, Application, Heating Medium, Type, Location, Sound Level, Ratings, Control, Piping, Maintenance.

UNIT ventilators and unit heaters are discussed in this chapter. The types, characteristics, method of rating, and procedures for selection and application of these products are described.

UNIT VENTILATORS

The term *unit ventilator* denotes an assembly, the principle functions of which are to heat, ventilate, and cool a space by the introduction of outdoor air in quantities up to 100 percent of its rated capacity. The heating medium may be steam, hot water, gas, or electricity. The essential elements of a unit ventilator are fans and motor, heating element, dampers, filters, and outlet grilles (or diffusers), all encased in a housing.

Unit ventilators are used primarily in schools, meeting rooms, offices and other areas where the density of occupancy indicates the need for controlled ventilation. The typical unit is equipped with a system of control that permits the heating, ventilating, and cooling effect to be varied while the fans are operating continuously. In normal operation, the discharge air temperature from a unit is varied in accordance with the room requirements. When heating is required, the air delivered is above room temperature. When the heat generated within the room by occupants, lights, sun, etc., is sufficient to cause overheating, the temperature of the air delivered is below that of the room. It is customary to equip unit ventilators with control devices that prevent the delivery of air at a temperature low enough to cause discomfort.

Classification

The various types of unit ventilators can be classified according to the three following methods:

1. *Heating Medium.* Four types of heating elements are used: (*a*) *steam*, (*b*) *hot water*, (*c*) *gas-fired*, and (*d*) *electric*.

2. *Type of Unit.* Under this classification there are three basic types to be considered: (*a*) *Floor-mounted* type, (*b*) *ceiling-mounted* type, and (*c*) *auditorium* type which are used in vertical, horizontal, or inverted positions. Figs. 1 and 2 show typical *floor-mounted* units. Fig. 3 shows a typical *ceiling-mounted* unit, while Fig. 4 shows a typical *auditorium-type* unit in horizontal position.

3. *Arrangement of Elements.* Unit ventilators may be of the (*a*) *draw-through*, or (*b*) *blow-through* arrangement. Fig. 1 shows a typical *draw-through* unit and Fig. 2 shows a typical *blow-through* unit.

Ratings of Unit Ventilators

Unit ventilators are customarily cataloged with two ratings: the *standard air rating* and the *anemometer air rating*. The *standard air rating* is obtained in accordance with the ASHAE *Standard Code for Testing and Rating Steam Unit Ventilators*.[1] This code requires 0 F entering air temperature, unit fan operating at rated speed with standard louver or grille on the outlet, and dry saturated steam at a temperature at the unit corresponding to a pressure of 16.7 psia (218.5 F). The *standard air rating* of the unit is the delivery in cubic feet per minute, converted to standard air at 70 F.

The *anemometer air rating* is peculiar to school ventilation. It originated as a convenient field measurement for checking air quantities, and is the basis of rating for ventilation requirements under some state and local building codes. The anemometer rating (air capacity) is obtained by averaging the air velocities obtained by an anemometer (propeller or averaging type) held over equal subdivisions of the outlet grille at a distance of 2 in. from the grille, and multiplying the average velocity by the gross area of the grille. The anemometer rating is based on the final temperature of the air leaving the grille while the unit is delivering outdoor air and recirculated air in the proper proportion and the heating element is supplying heat as specified. This

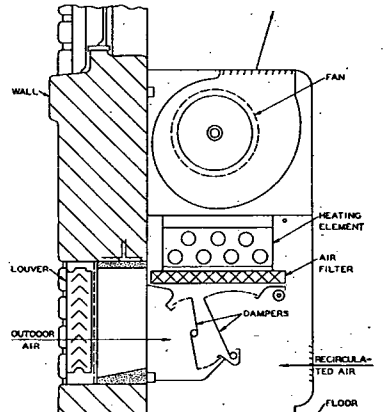


Fig. 1. . . . Typical Floor-Mounted Unit Ventilator—Draw-Through Type

method of rating is subject to error from differences in types and calibration of instruments, grille design, and the care with which the rating test is conducted. Because of this, the *standard air rating* is used by manufacturers to rate unit ventilators.

Table 1 shows the air delivery capacities by the two methods of rating and the approximate room heating equivalent in square feet of direct radiation (EDR) of an intermediate size of heating element. Heating elements are available for higher and lower capacities.

Heating Capacity Requirements for Unit Ventilators

Since a unit ventilator has the dual function of introducing outdoor air for ventilation and maintaining a specified room temperature, the heat required by the unit may be similarly divided into (1) *heat required for ventilation* H_v and (2) *surplus heat* H_s. The *surplus heat* is available for maintaining room temperature. If auxiliary radiation is installed, the *surplus heat* requirement may be reduced by a corresponding amount. The sum of H_v and H_s is the total heat H_t to be supplied by the unit ventilator.

These quantities of heat are related by the following equations:

$$H_v = 0.24\ W\ (t - t_o) \quad (1)$$

$$H_t = 0.24\ W\ (t_f - t_o) \quad (2)$$

$$H_s = H_t - H_v = 0.24\ W\ (t_f - t) \quad (3)$$

$$W = d\ 60\ Q \quad (4)$$

$$H_t = H_v + 0.24\ d\ 60\ Q\ (t - t_o) \quad (5)$$

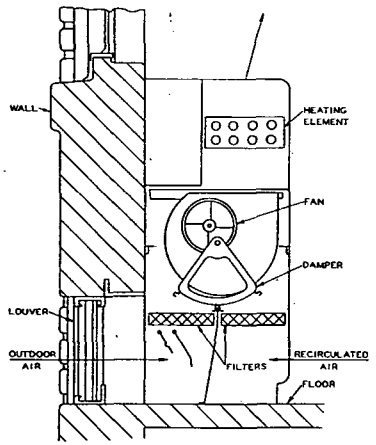


Fig. 2 .... Typical Floor-Mounted Unit Ventilator—Blow-Through Type

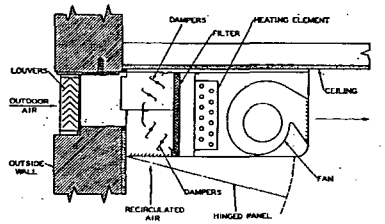


Fig. 3 .... Typical Ceiling-Mounted Unit Ventilator

where

d = density of air, pounds per cubic foot (0.075 lb per cu ft for Standard Air by definition).
H_s = surplus heat, Btu per hour.
H_v = heat required to warm air for ventilation, Btu per hour.
H_t = total heat requirements for both heating and ventilation, Btu per hour.
Q = volume of air handled by the ventilating equipment, cubic feet per minute.
t = temperature to be maintained in the room, Fahrenheit
t_o = outdoor temperature, Fahrenheit.
t_f = temperature of the air leaving the unit, Fahrenheit.
W = weight of air circulated, pounds per hour.
0.24 = specific heat of air at constant pressure (approximate value).

Example 1: The heat loss of a certain room is 24,000 Btu per hour, and the ventilating requirements are 1000 cfm. If the room temperature is to be 70 F and all air is taken from the outdoors at 0 F, what will be the total heat demand on the unit if it is required to provide for both the heating and ventilating requirements (combined system)?

Solution: Since the surplus heat is available to replace the heat loss of the room,

$$H_s = 24{,}000 \text{ Btu per hour.}$$

Substituting in Equation 5:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 1000\ (70 - 0)$$

$$= 99{,}600 \text{ Btu per hour}$$

$$t_f = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2 \text{ F}$$

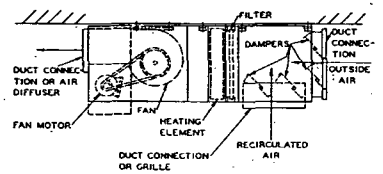


Fig. 4 .... Typical Auditorium Unit Ventilator in Horizontal Position

Table 1 .... Typical Capacities of Unit Ventilators for an Entering Air Temperature of 0 F

Cubic feet of air per minute		Total capacity in square feet, equivalent direct radiation	Capacity available for heating the room, square feet equivalent direct radiation	Final air temperature F
Anemometer rating	Standard air rating			
750	500	214	56	95
1000	750	320	84	95
1260	1000	427	112	95
1560	1250	534	141	95

If in *Example 1* a 1000 cfm (Standard Air) unit were required, but only 25 percent of the air introduced were outdoor air, the solution is:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 0.25 \times 1000\ (70 - 0)$$

$$= 42{,}900$$

$$t_f = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2$$

The only difference from *Example 1* is that the ventilation load has been reduced.

Heating Capacity Ratings

Heating capacity ratings for *steam* unit ventilators are obtained in accordance with the ASHAE *Standard Code for Testing and Rating Steam Unit Ventilators*.[1] Under this code the capacities are established for varying air entering temperatures in addition to the standard rating. Capacity ratings include the *entering air temperature*, the *total heating capacity*, the *surplus* or *heating capacity*, and the *final air temperature*.

Hot water unit ventilators have heating capacity ratings based on various entering water temperatures, flow rates or temperature drops, and entering air temperatures.

Gas-fired unit ventilators have heating capacity ratings obtained in accordance with regulations established by the American Gas Association. Heating capacities include the *total heat input* and the *total heat output*, in Btu per hour, for various entering air temperatures.

Electric unit ventilators have heating capacity ratings based on the heat input to the electric heating element, in kilowatts or Btu per hour.

The air capacities of hot water, gas-fired, and electric unit ventilators are obtained in accordance with the ASHAE *Standard Code for Testing and Rating Steam Unit Ventilators*.[1]

Applications of Unit Ventilators

Items to be considered in the application of unit ventilators are: (1) unit air capacity, (2) percent minimum outdoor air, (3) heating capacity, (4) cycle of control, and (5) location of unit.

The primary considerations in the selection of the *unit air capacity* are the mild weather cooling capacity and the number of occupants in the space. Other factors to be considered are state and local code requirements, volume of the room, density of occupancy, and the usage of the room. A safe rule for determining unit air capacity is to allow a total air quantity of 30 cfm per person, or six to nine room air changes, per hour, whichever is greater. With this quantity of air handled, it is possible to obtain satisfactory cooling in mild weather. Table 2 gives recommended room volume circulation rates for different types of spaces served. For rooms facing East, South or West, higher values should be used.

The minimum amount of outdoor air for ventilation is determined after the total air capacity has been established. It may be governed by state or local codes or may be calculated by the engineer to meet the ventilating air needs of the particular application.

The *heating capacity* of a unit to meet the heating requirement can be determined from manufacturers tables. Selection of *heating capacity* should always be made after determining the basic size of unit by selecting the unit *air capacity*.

Automatic Control of Unit Ventilators

Many different *cycles of control* are available. The principle difference in the various cycles pertains to the amount of outdoor air delivered to the room during normal periods of occupancy. Usually a room thermostat controls both a valve to regulate the heat supply and a damper to regulate the supply of outdoor air. An air-stream thermostat in the unit prevents the discharge of air below the desired minimum temperature. Unit ventilator control cycles provide the proper sequence for the following stages:

Warm-up Stage. All control cycles function to provide rapid warm-up by having the units provide full heat with the outdoor damper closed. Thus 100 percent room air is recirculated and heated until the room temperature approaches the desired temperature level.

Heating and Ventilating Stage. As the room temperature rises into the operating range of the thermostat, ventilation is accomplished by the partial or complete opening of the outdoor air damper according to the cycle used. Auxiliary heating equipment is shut off. As the room temperature continues to rise the unit ventilator heat supply is throttled.

Cooling and Ventilating Stage. When the room temperature rises above the normal level, cool air is discharged into the room. The room thermostat accomplishes this by throttling the heat supply, finally shutting it off, and opening the outdoor air damper to prevent overheating of the room. The air stream thermostat frequently takes control during this stage to prevent the discharge temperature from falling below a set level.

The three basic *cycles of control* commonly used are as follows:

Cycle X. 100 percent of outdoor air is admitted at all times except during the *warm-up stage.*

Table 2 .... Recommended Outdoor Air Capacities for Mild Weather Cooling

For North Exposure

Type of space	Recommended outdoor air delivery—room volumes per hour
Classrooms	6–9
Laboratories, Forge Shops	6–8
Metal Working Shops	6–8
Offices	6–8
Card Rooms, Lodges	6–9
Private Offices	6–9
Cafeterias & Kitchens	4½–7

Cycle Y. A minimum amount of outdoor air (normally 25 to 50 percent) is admitted during the *heating and ventilating stage*. This percentage is gradually increased to 100 percent, if needed, during the *cooling and ventilating stage*.

Cycle Z. Except during the *warm-up stage* a variable amount of outdoor air is admitted as needed to maintain a fixed temperature of the air entering the heating element. This is controlled by the air-stream thermostat which is set low enough (often 55 F) to provide cooling when needed.

Night Control

For maximum economy it is general practice to maintain the building at reduced temperature at night and over weekends and vacations. Several control arrangements to accomplish this are in common use. One uses the unit as a convector at night. Another turns the fans on and off as in the case of a unit heater whenever the unit convective capacity is insufficient. Still another arrangement uses the convective capacity of the unit supplemented by the convective capacity of auxiliary equipment installed within the room. More elaborate arrangements are available, permitting operation at daytime temperatures in some spaces and lowered temperatures in others for maximum economy during night occupancy of part of the building. For general information on controls, see Chapter 43.

Location of Unit Ventilators

The location of the unit is important. Wherever possible it should be placed against the outside wall and near the centerline of the room. It is difficult to obtain proper air distribution if the unit is installed either on an inside wall or in the corner of the room. Standard unit models are set against the wall, or may be recessed into the wall; and discharge the air stream upward. Ceiling models discharge the air horizontally.

Air Exhaust Vents

The location and size of exhaust vents are not of particular importance on systems employing unit ventilators. However, many states have regulations which require vents in public buildings. When vents are used, it is important that they be sized and located properly. Vents should be fitted with manual dampers so that the exhaust can be regulated for proper air distribution. Back-draft dampers should be installed in the vent opening to prevent cold air from entering through the vent.

It has become common practice, where regulations permit, to exhaust into the corridor through louvers in the classroom door, with powered or gravity exhaust from the corridor to the outdoors. Thus, the slight pressurizing of the building by the unit ventilators provides the force to exhaust the entire building. It is common practice also to use the exhaust from the classroom as secondary ventilation by passing the exhaust air through wardrobes and lockers. Many state and local codes make this arrangement mandatory.

Window Downdraft Prevention

Increasing use of large window areas has presented a window downdraft problem caused by chilled air flowing down the windows and into the occupied areas of the room. Three basic means of combatting these downdrafts have been developed for use with unit ventilators. These are shown in Fig. 5 and operate as follows:

Window sill heating employs finned radiation of moderate capacity installed along the wall under the window area. Heated air rises by convection upward and counteracts the downdraft by tempering it and diverting it upward to eliminate the cold draft.

Window sill recirculation is obtained by placing the return air intake along the window sill. Room or return air to the unit includes the cold downdrafts, takes them out of the occupied area of the room and thus eliminates the problem.

Window sill discharge directs a portion of the unit ventilator discharge air into a delivery duct along the sill of the window. The unit discharge air, delivered vertically at the window sill, provides room-wide air distribution and combats downdraft by upwardly-directed streams of air.

Unit Ventilators For Mechanical Cooling

With increased use of school buildings during the summer months, some school building programs incorporate mechanical cooling or provisions for including it in the future. In either instance, the unit ventilators provide the same function during the heating season as the unit ventilators previously discussed in this chapter, using hot water as the heating medium. Chilled water is used to provide summer air conditioning. Units designed only for heating and ventilating are not suitable for use with chilled water. When provision for future mechanical cooling is made, the supply and return piping, piping insulation, unit ventilators and controls are usually installed initially as required for complete air conditioning. Water chillers, cooling towers, or evaporative condensers, and associated accessories are installed later when mechanical cooling is required.

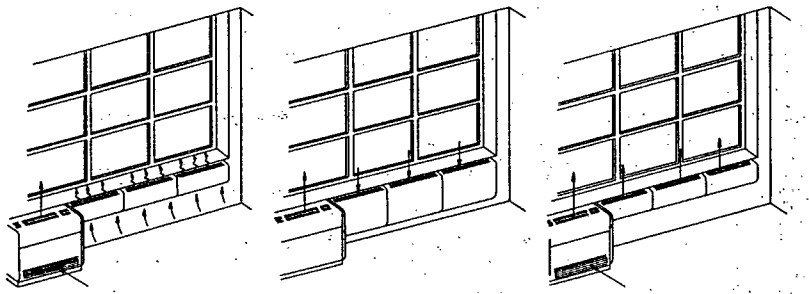


Fig. 5.... Three Methods of Preventing Window Downdraft

UNIT HEATERS

The term *unit heater* denotes an assembly of elements, the principal function of which is heating. The essential elements of a unit heater are a fan and motor, a heating element, an enclosure, and a directional outlet. Filters, dampers, duct collars, combustion chambers, and flues may also be included. Some types are shown in Figs. 6 to 13.

Classification

The various types of unit heaters in present use can usually be classified according to one or more of the following methods:

1. *By heating medium:* Under this classification there are five heating media to be considered: (*a*) steam, (*b*) hot water, (*c*) gas, (*d*) oil, (*e*) electric.

2. *By type of fan:* Under this classification there are two types of fans to be considered: (*a*) *propeller* and (*b*) *centrifugal*. Propeller fan units may be of the *horizontal-blow* or *down-blow* type. Centrifugal fan units may be of the smaller *cabinet* type, or larger *industrial* type. Either may be arranged for the delivery of air horizontally, or vertically up or down.

3. *By arrangement of elements:* Under this classification there are two types of units to be considered: (*a*) the *draw-through type*, in which the fan draws air through, and (*b*) the *blow-through type*, in which the fan blows air through the heating element. Direct-fired unit heaters are always of the blow-through type.

Application

Unit heaters have three principal characteristics, as distinguished from gravity heating units: (1) relatively large heating capacities in compact casings, (2) the ability to project heated air in a controlled manner over a considerable distance, and (3) a relatively low installed cost per Btu output. They are therefore usually employed in applications where the heating capacity requirements, or the physical volume of the heated space, or both, are larger than those that can be adequately or economically handled by gravity heating units.

Where the heating capacity requirement is large, the installation of a small number of unit heaters will usually prove more economical than will the use of a multiplicity of gravity heating units of the same total capacity. Where the volume of the space is large, and especially where ceilings are high, unit heaters can provide comfort through uniform heat distribution, and economy of operation by confining of the heat to the occupied zone and eliminating stratification in a manner which is difficult to attain with gravity radiation. Unit heaters located overhead permit 100 percent utilization of floor space. They may often be used to advantage in specialized applications requiring spot heating, or intermittent heating, such as the blanketing of outside doors in industrial plants, or in corridors and vestibules. Unit heaters are also employed where filtration of the heated air is required, and they may be modified to provide ventilation where the introduction of outdoor air is required.

Unit heaters are used principally for heating commercial and industrial structures such as garages, factories, warehouses, showrooms, stores and laboratories, and, for heating corridors, lobbies, vestibules, and similar auxiliary spaces in all types of buildings.

Unit heaters may be applied to a number of industrial processes, such as drying and curing, in which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption, such as fog removal in dye houses, or for the prevention of condensation on ceilings or other cold surfaces of buildings in which process moisture is released. When such conditions are severe, it is necessary that the units be equipped to draw air from outdoors in sufficient volume to provide a rapid air change, and that they operate in conjunction with exhaust ventilators or fans for exhausting the moisture-laden air. (See discussion of condensation in Chapter 10.)

Factors Affecting Application

There are four major factors to be considered in the application of unit heaters: (1) the heating medium to be employed, (2) the type of unit, (3) the location of the unit for proper heat distribution and (4) the permissible sound level.

Heating Medium

The selection of the proper heating medium is usually determined on an economic basis, and requires an examination of the first cost, the operating cost, and the conditions of use. Steam and hot water unit heaters are relatively inexpensive, but require a boiler plant and piping system, the per-unit cost of which generally decreases as the number of units installed increases. Gas- and oil-fired units are more costly, but the fuel lines and vent flues required are comparatively inexpensive, and the overall installation cost will usually be lower where a small number of units is used. Electric units are relatively inexpensive to purchase and install, but are usually associated with a high operating cost per Btu output unless electric rates are very favorable.

Steam or hot water units are most frequently used in new installations in which the number of units is large enough to justify the expense of a new boiler and piping system, and in existing installations or additions in which the system is of sufficient capacity to handle the additional load. *High-pressure steam* or *high-temperature hot water* units are usually employed in larger installations or where a high temperature medium is also required for process work. Low-pressure steam and conventional hot water units are usually installed in smaller installations and in those primarily concerned with comfort heating.

Gas-fired and oil-fired unit heaters are usually preferred in installations in which the number of units required does not justify the expense of a new boiler and piping system, where the space or the time required to construct a new boiler plant is not available, or where individual metering of the fuel supply is required, as in a shopping center. Gas-fired units are usually of the horizontal propeller type or the industrial centrifugal type. Oil-fired units are largely of the industrial centrifugal type. Some codes limit the use of direct-fired unit heaters in certain applications.

Electric unit heaters are used where low cost electric power is available, or for isolated locations, intermittent use, supplementary heating or temporary service requirements. Typ-

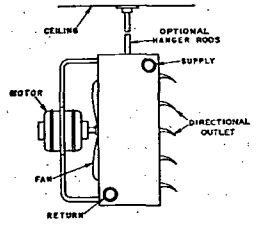


Fig. 6 .... Propeller Fan Unit Heater—Horizontal-Blow Type

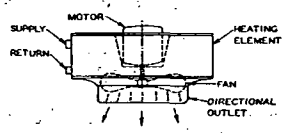


Fig. 7 .... Propeller Fan Unit Heater—Down-Blow Type

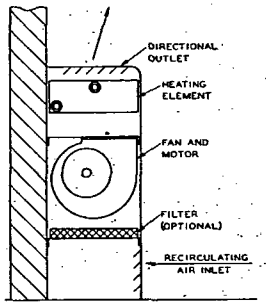


Fig. 8 .... Floor-Mounted Centrifugal Unit Heater—Cabinet Type

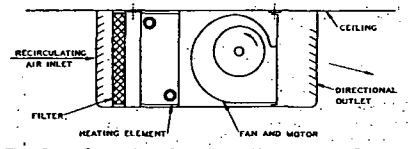


Fig. 9 .... Suspended Centrifugal Unit Heater—Cabinet Type

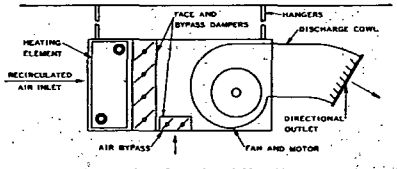


Fig. 10 .... Suspended Centrifugal Unit Heater—Industrial Type—Bypass Controlled

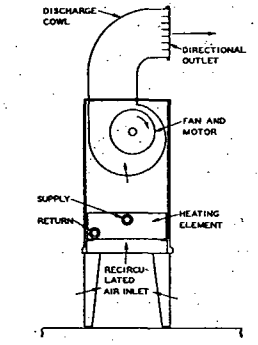


Fig. 11 .... Floor-Mounted Centrifugal Unit Heater—Industrial Type—Valve Controlled

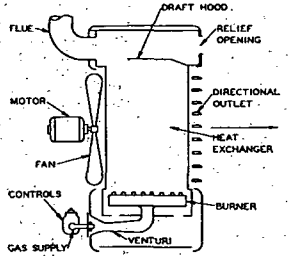


Fig. 12 .... Gas-Fired Propeller Fan Unit Heater

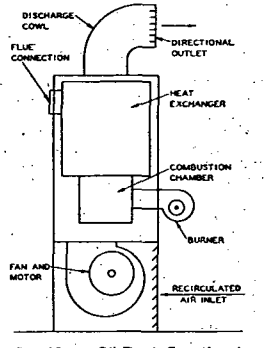


Fig. 13 .... Oil-Fired Centrifugal Fan Unit Heater—Floor Mounted

ical applications are ticket booths, watchmen's offices, factory offices, locker rooms, and other isolated rooms scattered over large areas. Electric units are particularly useful in isolated and untended pumping stations or pits where they may be thermostatically controlled to prevent freezing temperatures.

Type of Unit

Propeller fan units are usually used in free-delivery applications where the heating capacity and distribution requirements can best be met by units of moderate output used singly or in multiples, and where filtration of the heated air is not required. *Horizontal-blow* units are usually associated with low to moderate ceiling heights. *Down-blow* units are employed in high-ceiling spaces, and where floor and wall space limitations dictate an out-of-the-way location for the heating equipment.

Industrial centrifugal fan units are applied where heating capacities and space volumes are large, or where filtration of the heated air, operation against static resistance, or the introduction of outdoor air are required. Down-blow or horizontal-blow units may be used, depending on the requirement.

Cabinet unit heaters are used for applications where a more attractive appearance is desired. They are suitable for free delivery or low static pressure duct applications, may be equipped with filters, and can be arranged to discharge either horizontally, or vertically up or down.

Location for Proper Heat Distribution

Location of the units for proper heat distribution is of utmost importance in the application of unit heaters. In general, the units must be selected, located, and arranged so as to provide complete heat coverage and at the same time maintain acceptable air motion and temperature in the working or occupied zone. Proper application depends on a number of factors: the size, number, type and direction of blow of the units, the mounting height, the outlet velocity, the outlet temperature, the air volume and the type of directional outlet employed. Many of these factors are interrelated, and manufacturers therefore publish data for specific units showing heat coverage and mounting heights for various outlet velocities and outlet arrangements. Such data should normally be consulted in selecting unit heaters for a particular application.

In general, however, *mounting height* and the area of heat coverage increase with increased air volume and outlet velocity, and decrease with increased final temperature. The mounting height may be governed by space limitations or by the presence of equipment such as display cases or machinery. Both noise and air velocity in the occupied zone generally increase with increased outlet velocities and must therefore be considered.

The blow is dependent to a marked degree on the *temperature of air leaving the heater* as well as upon its velocity and volume. Increased final temperature will reduce the area of heat coverage and the mounting height. Care must be exercised in the application of down-blow unit heaters. The higher the mounting height, the lower must be the outlet temperature of the air leaving the heater in order that the heated air will be forced down into the occupied zone.

Since the outlet temperature of air from a unit heater increases with the temperature of the heating medium, unit heaters for high-pressure steam or high-temperature hot water should be designed to produce approximately the same leaving air temperature as would be obtained from a lower temperature heating medium.

In order to obtain the desired air distribution and heat diffusion, unit heaters are commonly equipped with directional outlets, adjustable louvers, or fixed types of diffusers. For a given unit with a given discharge temperature and outlet velocity, the mounting height and heat coverage can vary widely with the type of directional outlet, adjustable louver, or diffuser employed.

Other factors which may influence the heat coverage must also be considered. Obstructions, such as columns, beams, partitions, or machinery, either in the discharge air stream or in the approach area to the unit, can reduce the heat coverage substantially. The presence of strong drafts or other air currents will also reduce the coverage. Exposures such as large glass areas, or outside doors, especially on the windward side of the building, require special attention, and units should be arranged so as to blanket such exposures with a curtain of heated air which will intercept the cold drafts.

For area heating, horizontal-blow unit heaters in exterior zones should be located so as to blow along the exposure or toward it at a slight angle. Where possible, multiple units should be arranged so that the discharge air streams support each other and create a general circulatory motion in the space. Interior zones under exposed roofs or skylights should be completely blanketed. Down-blow units should be arranged so that the heated areas from adjacent units overlap *slightly so as to provide complete coverage.*

For spot heating of individual spaces in larger unheated areas, single unit heaters may be employed, but allowance must be made for the inflow of unheated air from adjacent spaces and the consequent reduction in heat coverage. Isolation of such spaces by partitioning or enclosure is always desirable from a heating standpoint.

Horizontal unit heaters should have their discharge outlets located well above the head level, and both horizontal *and vertical units should be so applied that the heated air* stream will be delivered to the occupied zone at temperatures and velocities which are not objectionable. Where possible, units should be so located that the discharge air flow is into open spaces such as aisles and not directly on the occupants.

Manufacturers' catalogs usually give suggestions for the best arrangements of various unit heaters, recommended mounting heights, heat coverages for various outlet velocities, final temperatures, and directional outlets.

Sound Level

Consideration must be given to the problem of noise generated by unit heaters if they are applied in spaces having a requirement for low sound levels. All unit heaters generate some noise, and whether or not that noise will be acceptable depends upon the type of space in which it is applied. For example, units with noise levels acceptable in *a manufacturing plant might be too noisy for use in a* show-room. A unit heater for a particular application must therefore be selected for satisfactory sound level as well as for adequate heating capacity and proper heat distribution.

Ratings of Unit Heaters

It is standard practice to rate all types of unit heaters on the basis of the amount of heat delivered by the air in Btu per hour above an entering air temperature of 60 F.

Steam. Rating of steam unit heaters has been standard-

ized by a code[3] in which the following items are the basis of rating: dry saturated steam at 2 psig pressure at the heater coil; air at 60 F (29.92 in. Hg barometric pressure) entering the heater; and heater operating free of external resistance to air flow.

The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature increases. The heating capacity for any condition of steam pressure and entering air temperature other than standard may also be determined by means of a standard procedure prescribed by this code.

Hot Water. A standard for the rating of hot water type unit heaters has also been established by code[4] in which the following items are the basis of rating: entering water at 200 F; water temperature drop 20 F entering air at 60 F (29.92 in. Hg barometric pressure); and heater operating free of external resistance to air flow. Variations in entering water temperature, entering air temperature and water

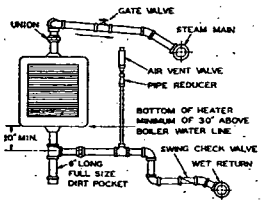


Fig. 14 .... Unit Heater Connection to Low-Pressure Steam Closed Gravity System

flow rate will affect capacity. This code prescribes a method of translating the heating capacity as obtained under test conditions to other conditions of air and water temperature.

Gas-Fired. Gas-fired unit heaters are rated in terms of both input and output in accordance with the approval requirements of the American Gas Association.

Oil-Fired. Ratings of oil-fired unit heaters are based on heat delivered at heater outlet in Btu per hour.

Electric. Electric unit heaters are rated on the energy input to the heating element, expressed in terms of kilowatts, or Btu per hour.

Effect of Resistance Upon Capacity. Unit heaters are customarily rated at free delivery type units. If outside air intakes, air filters, or ducts on the inlet or discharge are used, a reduction in air and heating capacity will result because of this added resistance to air flow, unless compensating changes are made in fan speed. This reduction in capacity will depend upon the characteristics of the heater, and on the type, design, and speed of the fans, so that no specific percentage reduction can be assigned for all heaters at a given added resistance. The heat output to be expected under other than free delivery conditions should be secured from the manufacturer.

Effect of Inlet Temperature. Changes in entering air temperature will influence the total heating capacity in most unit heaters and the final temperature in all units. Since many unit heaters are located some distance from the occupied zone, consideration should be given to the possible differences in temperature of the air actually entering the unit from that being maintained in the heated area. This is especially true with down-blow unit heaters. In general, the temperature gradient (temperature difference per foot of elevation) when using unit heaters, is less than when using gravity heating units.[5]

Higher velocity units and units with lower discharge air temperature, will maintain lower temperature gradients than units with higher discharge temperatures. Valve-controlled or bypass-controlled units employing continuous fan operation will maintain lower temperature gradients than will units employing intermittent fan operation. Directional

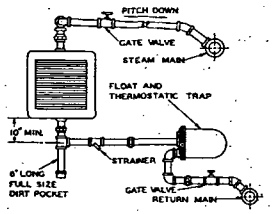


Fig. 15 .... Unit Heater Connection for Low-Pressure Steam Open Gravity or Vacuum Return System

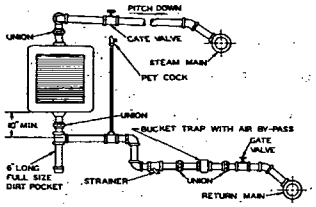


Fig. 16 .... Unit Heater Connection to High-Pressure Steam System

control of the discharged air from a unit heater can also be an important factor in effecting satisfactory distribution of heat and reducing floor-to-ceiling temperature gradients.

When some outdoor air is introduced, the temperature of the mixture of outdoor and recirculated air must be calculated and used as the entering air temperature at the heater. For recirculating heaters located *at the floor* or with *intakes at the floor*, the temperature of air entering the heater should be assumed to be the same as that to be maintained in the room itself.

Automatic Control of Unit Heaters

The controls for a steam or hot water unit heater can either provide (1) *on-off operation* of the unit fan or (2) *continuous fan operation* with modulation of the heat output.

For *on-off* operation a room thermostat is used to start and stop the fan motor. A limit thermostat, often strapped

to the supply or return pipe prevents fan operation in the event heat is not being supplied to the unit.

Continuous fan operation eliminates the intermittent blasts of hot air resulting from on-off operation, and the stratification of temperature from floor to ceiling which often occurs during the *off* periods. In this arrangement a proportional room thermostat controls a valve modulating the heat supply to the coil or a bypass around the heating element. A limit thermostat or auxiliary switch stops the fan when heat is no longer available.

Control of direct-fired and electric units is usually accomplished by intermittent operation of the heat source under

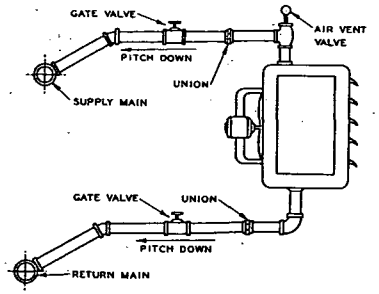


Fig. 17 .... Connection of Horizontal Unit Heater to Hot Water System

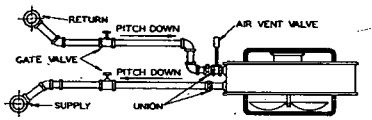


Fig. 18 .... Connection of Vertical Unit Heater to Hot Water System

control of the room thermostat, with a separate fan switch to run the fan when heat is being supplied. For a more general discussion of automatic control, refer to Chapter 43.

Unit heaters may be used in summer as a means of providing air motion. In such cases the source of heat is shut off and the thermostat is provided with a bypass switch that will permit fan operation independently of the controls.

Piping Connections For Unit Heaters

Piping connections for steam unit heaters are similar to those for other types of fan blast heaters. The piping of unit heaters must conform strictly to the system requirements while at the same time permitting the heaters to function as intended. The basic piping principles for steam systems are discussed in Chapter 26.

Rapid condensation of steam, especially during heating-up periods, is characteristic of steam unit heaters. The return piping must be planned to keep the heating coil free of condensate during periods of maximum heat output, and the steam piping must be ample to carry a full supply of steam to the unit to take the place of that condensed. Adequate sizes of piping are especially important where a unit heater fan is operated under on-off control because the condensate rate fluctuates rapidly.

Recommended piping connections for unit heaters[6] are shown in Figs. 14 to 18.

In steam systems the branch from the supply main to the heater should pitch toward the main, and be connected to its top to prevent condensation in the main from draining through the heater where it might reduce capacity and cause noise.

The return piping from steam unit heaters should provide a minimum drop of 10 in. below the heater so that the head of water required to overcome resistances of check valves, traps, and strainers will not cause condensate to remain in the heater.

Dirt pockets shown in Figs. 14 to 16 are essential, and strainers are recommended as an additional means of retaining dirt and scale which might affect operation of check valves and traps. Return piping between the heater and strainer should be of the same size as the heater outlet. Strainers should always be installed in the steam supply line if the heater is equipped with the steam-distributing type of coils or is valve-controlled.

An adequate air vent is required for low-pressure closed-gravity systems. The vertical pipe connection to the air vent should be at least ¾ in. IPS to permit separation of the water from the air passing to the vent.

In vacuum systems, if thermostatic instead of float-and-thermostatic traps are used, a cooling leg must be provided ahead of the trap.

In high-pressure systems it is customary to vent the air through a pet cock which is left slightly open at all times. Where possible it is advisable to install pressure reducing valves to permit operation of the heaters at low pressure. Traps used must be suitable for the operating pressure encountered.

In connecting piping to hot water unit heaters, the piping should be pitched to permit escape of air to the high point in the piping where it can be vented to the atmosphere. An air vent at the heater is used to facilitate air removal or to vent the top of the heater. Provision should be made for complete drainage of the system, including nipple and cap drains or drain cocks where units are located below mains.

Maintenance

Regular inspection based upon a schedule determined by the amount of dirt in the atmosphere will insure maximum economy in operation and maintenance of heating capacity.

Heating elements should be cleaned when necessary by brushing or blowing with high-pressure air, or by using a steam spray. Removal of the heating element and washing with a mild alkali solution, followed by thorough rinsing with water, may be necessary in certain installations.

Dirty fan blades may cause unbalance and vibration with consequent noise and damage to bearings. Hence, fan blades should be cleaned when necessary. Vibration or noise may also be caused by improper fan position or loose set screws.

There is considerable difference in the attention required by the various types of motors used with unit heaters. This is particularly true of lubrication, the instructions for which must be carefully followed for trouble-free operation. Excessive lubrication may cause lubricant to reach and damage the motor. An improper lubricant may cause failure of bearings. Instructions for care of the motor on any unit

heater should therefore be obtained from the manufacturer and should be kept at the unit.

Fan bearings and drives must be given the lubrication and other attention specified by the manufacturer. If the unit is direct-connected, the couplings should be inspected periodically for wear and alignment. V-belt drives should have all belts replaced with a matched set if one belt shows wear.

Periodical inspections of traps, check valves, and air valves, and the replacement of worn parts are important maintenance items. Strainers should be cleaned regularly.

When filters are incorporated in the unit, they must be cleaned or replaced when dirty.

For prevention of corrosion in unit heaters, see Chapter 55, Corrosion and Water Formed Deposits, Causes and Prevention, in this edition, and also Bulletin 12, *The Care and Maintenance of Steam and Hot Water Unit Heaters*, published by the Air Moving and Conditioning Association, Inc.

REFERENCES

[1] *ASHVE Standard Code for Testing and Rating Steam Unit Ventilators* (ASHVE TRANSACTIONS, Vol. 38, 1932, p. 25).

[2] G. L. Larson, D. W. Nelson, and R. W. Kubasta: ASHVE RESEARCH REPORT No. 936—Investigation of air outlets in class room ventilation (ASHVE TRANSACTIONS, Vol. 38, 1932, p. 463). F. C. Houghten, Carl Gutberlet, and M. F. Lichtenfels: ASHVE RESEARCH REPORT No. 1017—Air supply to classrooms in relation to vent flue openings (ASHVE TRANSACTIONS, Vol. 41, 1935, p. 279).

[3] *Standard Code for Testing and Rating Steam Unit Heaters* (Adopted jointly by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and Industrial Unit Heater Association, *IUHA Bulletin* No. 10, January 1950, 2nd ed.).

[4] *Standard Code for Testing and Rating Hot Water Unit Heaters* (Adopted jointly by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and Industrial Unit Heater Association, *IUHA Bulletin* No. 11, April 1953, 2nd ed.).

[5] G. L. Larson, D. W. Nelson, and O. C. Cromer: ASHVE RESEARCH REPORT No. 958—Temperature gradient observations in a large heated space (ASHVE TRANSACTIONS, Vol. 39, 1933, p. 243). G. L. Larson, D. W. Nelson, and John James: ASHVE RESEARCH REPORT No. 1011—Tests of three heating systems in an industrial type of building (ASHVE TRANSACTIONS, Vol. 41, 1935, p. 185).

[6] *Piping, Trapping and Venting Steam and Hot Water Unit Heaters (Air Moving and Conditioning Association, Inc., Bulletin* No. 15).

CHAPTER 16

UNITARY AIR-CONDITIONING EQUIPMENT

Definitions, Classification, and Components of Unit-type Equipment, Remote and Self-contained Units, Room Air Conditioners, Ratings of Unit Air Conditioners, Application of Unitary Equipment, Unit Air Coolers

THIS chapter presents the physical characteristics of air-conditioning and air-cooling units; a suggested procedure for selection of units; and some of the factors involved in the application of unitary equipment. In general, factory-produced unit equipment can be obtained to accomplish all of the functions possible from field assemblies, but the advantages of unit equipment are most apparent in small and moderate capacities. Above 12,000 cfm capacity, or approximately 40 tons of refrigeration capacity, handling and assembly costs generally favor the use of field-assembled units. Multiple application of unitary equipment is frequently justified for large gross tonnage installations where zoning or a minimum amount of ducts is desirable.

DEFINITIONS

The term *air-conditioning* unit has been loosely used as a name for all types of factory produced air-handling, cooling, or heating units. A joint committee[1, 2, 3] on Rating Refrigerating Equipment has defined the various types of unitary equipment as follows:

1. A *Cooling Unit* is a specific air-treating combination consisting of means for air circulation and cooling within prescribed temperature limits.[1]

2. An *Air-Conditioning Unit* is a specific air-treating combination consisting of means for ventilation, air circulation, air cleaning, and heat transfer, with control means for maintaining temperature and humidity within prescribed limits.[2]

3. A *Cooling Air-Conditioning Unit* is a specific air-treating combination consisting of means for ventilation, air circulation, air cleaning, and heat transfer, with control means for cooling and maintaining temperature and humidity within prescribed limits.[1]

4. A *Self-Contained Air-Conditioning or Cooling Unit* is one in which a condensing unit is combined in the same cabinet with the other functional elements. Self-contained air-conditioning units are classified[3] according to the method of rejecting condenser heat (water-cooled, air-cooled, and evaporatively cooled), method of introducing ventilation air (no ventilation, ventilation by drawing air from outdoors, ventilation by exhausting room air to the outdoors, or ventilation by a combination of the last two methods), and method of discharging air to the room (free delivery or pressure-type).

5. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.[1]

6. A *Pressure-Type Unit* is for use with one or more external elements which impose air resistance.[1]

7. A *Forced-Circulation Air Cooler* is a factory encased assembly of elements by which heat is transferred from air to refrigerants.[3]

CLASSIFICATION OF UNIT-TYPE EQUIPMENT

Field assembled apparatus as described in Chapter 19 can be designed in shape, size, and capacity for any application, with the refrigeration and heating system exactly balanced to a specific load condition.

To obtain the economies of factory production, component assemblies are available in the size ranges of greatest usage. In addition to assemblies of high-side equipment in condensing units, the fan-coil units are factory made in various types as indicated by the preceding definitions.

If the condensing unit and the cooling and heating coil surfaces are carefully selected, and if proper consideration is given to reduction of piping losses, the performance of the combined system will compare favorably with field assembled apparatus. *A system in which the air handling unit is separated from the condensing unit, is called a remote system.* The conditioning unit is designated as a remote unit. Capacities of remote units usually range from 2 to 100 tons.

The self-contained air-conditioning or cooling units are widely used in capacities through 15 tons, and are available in larger sizes. Each model is designed for a specific cooling capacity and range of air delivery. Experience has demonstrated that well designed and properly applied equipment of this type produces excellent results in commercial comfort installations. In most designs of the larger self-contained units, the equipment can be adjusted to approximate the load calculations through a selection of the air quantity. An increase of air quantity in this type of equipment increases the sensible heat proportion, and a decrease of air quantity tends to provide a lower leaving dew point and a high proportion of moisture removal. However, the usual field practice is to adjust the fan speed for quietness and air distribution rather than for ratio of latent to sensible loads as indicated by a load estimate.

To meet the requirements of individual comfort in small rooms and offices, self-contained units, called room coolers, find extensive and economical application. These units are usually restricted to summer and intermediate season operation, and range from ⅓ to 1½ tons of refrigeration capacity.

A special application of remote units is found in the unit air cooler which is used extensively in refrigeration work. Its primary function is to reduce temperatures in insulated and sealed storage spaces, and humidity control is a secondary consideration. Because of the small temperature differences between the coil and room temperatures, unit coolers handle three to five times as much air per ton as remote units used in air conditioning.

The attic fan or exhaust fan is sometimes referred to as a cooling unit, but since it contains no element of heat transfer, it is treated in Chapter 22, Fans.

COMPONENTS OF UNIT-TYPE EQUIPMENT

The unitary equipment is designed for the air-conditioning loads of most common occurrence. Highly special projects and projects of infrequent occurrence may require field-assembled equipment for best results. Field modifications

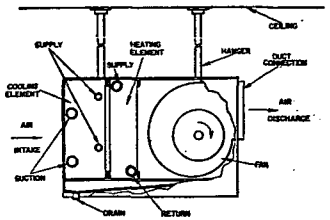


Fig. 1 . . . . Horizontal Remote Type Unit Air Conditioner

generally prove inadvisable except by the most experienced engineering personnel. The basic design considerations of unitary equipment are discussed in the following paragraphs.

Remote Units

Remote units can be obtained in two general classes, horizontal as shown in Fig. 1 and vertical as in Fig. 2. The construction of both is essentially the same, except for the drain pan and filter locations.

Casings. Casings are generally constructed of sheet metal with angle iron frames and with removable panels for access to coil connections, blower bearings, filters, and drain. Casings should be airtight. Panels should be tight-fitting with cam or similar fastenings for easy opening. Panel openings at coils for heavy units should be large enough to receive coils after the casing is suspended. Frames should be fitted with lugs strong enough to suspend horizontal units. Nonmetallic casings are of advantage in small remote units for reducing sound, particularly when propeller-type fans are used.

Insulation. Remote units are available with waterproof and verminproof, sound and heat barrier insulation on the inside of the casing. They are also available with flanges and flanged access doors to permit insulation after installation.

Drain Pans. Because of the corrosive effect of mild picric, carbonic, and sulfurous acids absorbed by condensate, drain pans are usually made of 14 gage or heavier metal. They should be hot dipped galvanized after fabrication, or otherwise treated to resist corrosion. Some manufacturers extend the drain pan under the entire unit, but in any case it should extend far enough to catch any condensate carried over from the coils. The drain connection should be readily accessible for cleaning and, in air-conditioning work, should be generously sized and trapped. Some municipal codes require a minimum size of 1¼ IPS.

Blowers. The usual practice among manufacturers is to use blowers having external resistance capacity in keeping with the size and type of application for which the unit is designed. *For example, the larger self-contained air conditioners* have blower and motor arrangements for delivering the design air quantities against the larger external static pressures that are encountered when applying these units for use with supply and return-air duct systems. In a well designed unit, the relation of the blower and the coil surface is arranged, through testing, to assure adequate uniformity of air distribution over the heat-transfer surface.

Some of the smaller suspended-type units, Fig. 3, use propeller fans with a trailing edge blade in order to obtain required pressure characteristics with quiet operation. Since most of the motors driving these fans are direct-connected and use brushes for starting, adequate access should be provided for maintenance and inspection.

Cooling Coils. The cooling and dehumidifying coils used in unit air conditioners are essentially the same as those used in central station units. A proper balance of coil capacity with the other components of the packaged air conditioner is an essential of good design and performance. Ample coil capacity is a requirement for good operation at light loads and low air quantities.

Where multiple circuits are used in the larger coils, equal distribution of the cooling medium to the various circuits is vital in order to develop full capacity of the coil. The cooling coils also perform the function of dehumidification. To prevent carryover of condensate, some coil designs require eliminator sections even at the normal face velocities of 400 to 600 fpm.

In addition to condensing the moisture out of the air, the design and arrangement of the cooling coil should provide for quick and effective draining of the condensate into the pan. Proper manufacturing practices to provide for uniform fin spacing and laboratory development to determine coil-surface arrangements are among the requirements for good drainage from the cooling and dehumidifying coils of self-contained air conditioners.

Heating Coils. Heating coils of unit air conditioners are usually conventional blast coils, and can be obtained with or without non-freeze steam distribution features. They usually match the cooling coils in face area, and are one or two rows deep depending on the heating requirements. Where coils are selected for hot water and have more than two rows of tubes, the air resistance and the space requirements of the cooling and heating coils should be carefully checked.

Humidification. Spray-type humidifiers are usually used in remote systems, but in some cases pan-type humidifiers or steam humidifiers are also used. Some condensation on the inside of the unit casing may occur with possible water damage if the unit is located in a cold space without adequate insulation. Spray-type humidifiers should be located so that no carryover of moisture occurs.

Filters. It is almost axiomatic that all units should have filters. Some small suspended units of 1 ton capacity or less, with low coil-face velocities and propeller fans, are equipped

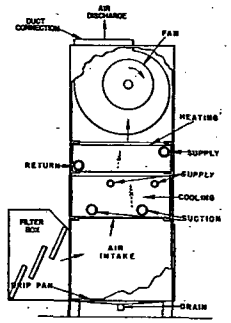


Fig. 2 . . . . Vertical Remote Type Unit Air Conditioner

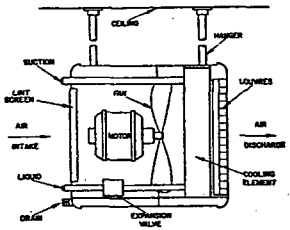


Fig. 3 . . . . Suspended Propeller Fan Type Unit Air Conditioner

only with lint screens or operate without filters, but in them the coils must be periodically cleaned and there is constant danger of clogging of the drain, with the possibility of water damage. Filters used are usually of the throw-away type, although cleanable filters are available for most of the larger units. Care should be taken to insure adequate filter surface, since the cross-sectional area of the unit is seldom adequate for filter area. V-shaped or staggered filter arrangements are quite commonly used to increase filter area.

Motors. In some units where the motor is mounted inside, adequate access for maintenance and clearance for tightening belts are imperative. When motors are mounted in this manner, the heat equivalent of the motor input must be added to the heat load to be absorbed by the system, and this requires a lower exit air temperature at the coils. Usually, however, the motor is located outside of the casing where it is readily accessible for service. In this case, only the brake horsepower required by the fan is transformed into heat to be included in the load calculations.

In either application, motors should be selected with adequate horsepower to handle the design volume of air against the resistance of the system when the coils are wet, and then checked against the possible horsepower requirements for the increased volume of air obtained when the coils are dry.

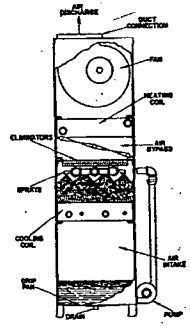


Fig. 4 . . . . Spray-Type Remote Unit Air Conditioner

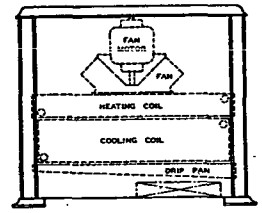


Fig. 5 . . . . Remote Floor Type Room Unit Air Conditioner

Other Types of Remote Units

Features of various other types of remote air conditioners are given in the following paragraphs.

Spray-Type Unit. Fig. 4 shows a spray-type unit used by designers who prefer air-washing and coil-wetting features. These units are equipped with a pump that sprays water or brine over the coils. Due to the direct mixing of the condensate and the spray, provision must be made for overflow in summer and replacement of water evaporated in winter.

Dehumidifying Units. In a further modification of spray-type units, absorbent brine solutions such as lithium chloride are used to remove moisture from the air. As explained in Chapter 42, the latent heat of the moisture removed is changed to sensible heat, so that coils must be used as after-coolers to obtain the right dry-bulb temperatures. Factory-produced units are also available for use with solid adsorbents such as silica gel.

Another recent development as a manufactured item is the small refrigeration-type dehumidifying unit for control of moisture in residential basements.

Remote Room Units. For individual rooms, with cooling load requirements of ½ to 1½ tons, remote units are available in attractive casings for installation within the room. A suspended type is shown in Fig. 3 and a floor type, such as is

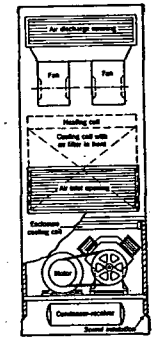


Fig. 6 . . . . Self-Contained Water-Cooled Air Conditioner

usually installed in place of an existing radiator, is shown in Fig. 5. Furnished with chilled water from a central plant, these units offer a satisfactory method of conditioning existing office, hotel, and apartment rooms. These units may be obtained with filters and outdoor air connections, but for most satisfactory applications are used as supplements to the central systems that supply properly conditioned and filtered air to the areas served.

Induction units, using primary conditioned air under pressure to induce local circulation, are described in Chapter 19.

SELF-CONTAINED UNITS

A typical large self-contained air-conditioning unit is shown in Fig. 6. It is essentially a remote vertical-type conditioner mounted on top of a sound-insulated enclosure containing the condensing unit. Air distribution is obtained by means of grilles mounted in the discharge plenum, when the unit is located in the conditioned area. Duct distribution of conditioned air can be obtained by removing the plenum, connecting directly to the blower discharge, and safing the top of the unit.

The heat generated by the compression of refrigerant gases and that given off by the electric motor are removed from the compressor compartment in four ways: (1) by the use of a water coil in the compressor compartment, (2) by utilizing the cold suction gases, (3) by drawing part of the return air through the compressor compartment; and (4) by circulating room air through the compressor compartment by means of a fan attached to the motor shaft.

Most self-contained conditioners are constructed with enough sound-isolation material for installation directly in the conditioned spaces of the higher-sound-level commercial-type of applications. Special sound treatment means are to be supplied as part of the installation when applying these conditioners to spaces having low sound-level requirements. Conditioners may be installed outside the conditioned area and the connecting supply and return-air ducts can be acoustically treated as required for the specific installation.

ROOM AIR CONDITIONERS

A room air conditioner is a factory-made encased assembly designed primarily as a unit for mounting in a window or through a wall, or as a console, for free delivery of conditioned air to an enclosure and without ducts for conditioned air supply or return. It includes a prime source of refrigeration and dehumidification and means for circulating and cleaning air, and may also include means for ventilating, heating, or performing other functions.[4] Most room air conditioners are used in residences and apartments, although many are installed in small commercial establishments. A number of multistory buildings have been completely conditioned with room units.

Although most room air conditioners are designed for conventional window mounting, many are made for mounting in casement windows, walls, or as console types in front of windows or walls. In such units, the arrangement of the basic components is often quite different from that employed in window units because emphasis may be placed on minimum depth or height rather than on width. Since great strides have been made in recent years in the reduction of overall unit dimensions, the current trend is toward units that can be mounted essentially flush with the inner wall face and protrude very little beyond the outer face. A recent innovation is the small, light-weight *portable* room air conditioner.

Most room air conditioners have air-cooled condensers although water-cooled versions have appeared from time to time. In air-cooled units, condensate disposal is typically provided by equipping the condenser fan with a slinger ring which picks up condensate and throws it on the condenser coil where it is evaporated into the condenser air stream and coincidentally provides some evaporative condensing effect.

Sizes

Room air conditioners are available in capacities ranging from about 3000 Btuh up to 18,000 Btuh or more. Sizes have been popularly designated in terms of the nominal horsepower of the compressor motor, although this is not necessarily an accurate indication of unit capacity. Units are available in sizes up to 2 hp, with the ¾ hp and, more recently, the 1 hp units being most popular.

The National Electric Code[5] limits the amperage rating of motor-operated equipment installed on multi-outlet branch circuits to 50 percent of the circuit rating, and on single-outlet branch circuits to 80 percent of the rating. For the typical 14-gage (15 ampere) house circuit, this effectively limits room air-conditioner amperage ratings for multi-outlet and single branch circuit installations to 7.5 and 12 amperes respectively. Many room air conditioners are designed for maximum performance within these amperage limits, regardless of nominal compressor horsepower.

Component Descriptions

Although many different types and variations of components have been used in room air conditioners, it is possible to give a fairly typical picture of current production. Hermetically-sealed refrigerant circuits are now universally employed, and the compressors are usually of the welded-shell reciprocating type operated by capacitor-start, capacitor-run, or permanent-split capacitor motors. Both evaporator and condenser coils are usually of aluminum plate fin and copper coil construction. Capillary tubes are normally employed as the expansion device, although valves are sometimes used for special purposes. Both centrifugal and axial-flow fans are used, sometimes one of each, with the latter equipped with a slinger ring for condensate disposal. One shaded-pole or permanent-split capacitor fan motor, mounted on the condenser side or in the partition, is frequently employed to operate both fans although in some designs two fan motors are used. A throw-away or cleanable type of filter is standard equipment. Grilles and an increasing number of other parts are being constructed of plastic. A slide-out chassis, containing the complete refrigerant and air circuits, is often used. Nonflammable and nontoxic refrigerants (ASA B9.1, Group I Classification[6]) are used exclusively.

Features

A thermostat and adjustable louvers for directing the conditioned air have become standard features on most units. A ventilation or outdoor-air damper is almost always included and sometimes an exhaust air damper. Two-speed fan motors are frequently used to give quieter operation during off-peak load conditions. Special filters and other air purification devices are featured on some models.

Some room air conditioners are designed as heat-pump units to provide heating during the intermediate seasons, or for year-round conditioning in some climates. Resistance heating has also been incorporated into some units for the same purpose.

Electrical Characteristics

Room air conditioners are normally designed to operate on single-phase, 60-cycle power at 115, 208, or 230 volts. Units for the export market are made for 50-cycle operation. Attachment plug requirements, which are related to the *National Electrical Code* ampere limitations, are specified by Underwriters' Laboratories, Inc.[7]

The rapid public acceptance of room air conditioners has presented special problems to the utility companies who must supply power to meet the electrical demands of these units during summer heat waves. As a result, recommended maximum starting currents and minimum power factors have been established for room air conditioners and are listed in *ARI Standard* 110-56.

Rating

Room air conditioners are tested for rating under the provisions of the *ASRE Standard* 16-56[8] at standard rating conditions of 80 F db and 67 F wb room air temperature and 95 F db and 75 F wb outdoor air temperature for air-cooled units. Rating requirements are specified in the *ARI Room Air-Conditioner Standard* 110-56. That organization is presently conducting a room air-conditioner rating compliance program.

Other Design Considerations

Air-flow rates vary considerably, with evaporator air quantities of 350–400 cfm per ton and condenser air quantities of 650–700 cfm per ton being approximately representative of current practices. Sensible heat factors usually average about 0.70, and performance factors (Btu per watt hour) about 6.5.

Room air conditioners should be designed to operate satisfactorily at the maximum conditions they might normally encounter. They should also be expected to perform without dripping or blowing of water from the unit. Standard test requirements for such considerations have been established by ARI in *ARI Standard* 110-56.

Safety requirements for room air conditioners are prescribed in the *American Standard Safety Code for Mechanical Refrigeration* B9.1 and the Underwriters' Laboratories *Standard for Room Air Conditioners*.

Selection

Many room air conditioners are sold over the counter without careful attention being given to matching the capacity of the unit with the requirements of the load. The ARI has available a fairly simple Cooling Load Estimate Form for Room Air Conditioners[9] which can be used to make a reasonably proper selection. Most manufacturers also have cooling load calculation forms available and many have even simpler quick-selection charts or devices.

Installation

An installation accessory package is usually furnished with most units. These are designed so that an installation can be completed in an hour or so by one or two men using a few common tools. Accessory packages and wall sleeves are sometimes available for casement windows and through-wall mountings.

The only connection that is necessary for air-cooled units is to insert an electrical plug into a convenience outlet of the proper rating.

Controls for Room Cooling Units

Control devices provided for self-contained cooling units generally will include all necessary means for automatic operation. Provision is also made for adding auxiliary external controls, when desired. Remote units are not generally equipped with controls. Control of remote units and the addition of auxiliary controls for self-contained units can be made by application of the control principles outlined in Chapter 19 Central Systems for Air Conditioning and Chapter 43 Automatic Control.

RATINGS OF UNIT AIR CONDITIONERS

Two standards have been used for rating and testing of unit air conditioners: (1) *Standard Method of Rating and Testing Air Conditioning Equipment*,[1] covering all types of air-conditioning units except the self-contained type and (2) *Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling*[2] covering the self-contained type. (*ASRE Standard Methods of Rating and Testing Air Conditioners, ASRE Standard* 16-56, covers rating and testing of all types which operate nonfrosting when cooling and dehumidifying at standard rating conditions.)

The standard rating of a self-contained unit, for the conditions specified in Table 1, includes all items which apply to the function of a unit such as: (1) name of unit, (2) functions which unit performs, (3) data on cooling, (4) data on heating, (5) data on air flow, and (6) data on humidification.

The standard rating conditions for unit air conditioners, other than the self-contained type, are identical with those in Table 1, except that entering wet-bulb temperature for cooling is expressed as 50 percent relative humidity (66.7 F wet-bulb) instead of 67 F wet-bulb temperature. In addition, the saturated suction refrigerant temperature for comfort cooling is specified at 40 F. This condition is omitted from Table 1 for self-contained units because it is not separately controllable in the testing of a unit that includes the evaporator and condensing unit.

APPLICATION OF UNITARY EQUIPMENT

One of the chief advantages resulting from use of factory produced units is the saving in installation and field assembly labor, a factor that should always be kept in mind when selecting a location for these units. Because of their compactness, the tendency exists to put them in closets, storage rooms, and other inaccessible places, where installation is so difficult that much of this cost advantage is lost.

Access panels are provided on units for the proper servicing and maintenance of the equipment. Adequate outside clearance at these panels is essential. Wherever there is danger of freezing of coils or clogging with dirt, sufficient clearance should be available for replacing them without removing parts of the building.

The outstanding source of difficulties in unitary systems is usually dirty filters. The characteristics of the light-weight fans used are such that air volume drops off rapidly with increase in static resistance. Since changing of filters is an unpleasant duty likely to be neglected unless it can be done easily, the operator should be at the same level as the filters, rather than under them, when they are being removed.

Variable-pitch motor pulleys are usually provided for adjustment of the air quantity. The fans and motors of most

Table 1.... Standard Rating Basis for Self-Contained Air-Conditioning Units

Functions	Types of Units	Rating Condition		
		Item	Description	Value
All	All	a	Barometric pressure	29.92 In. Hg
	Water-cooled, air-cooled, and evaporatively cooled condensers	b	Unit ambient and air entering room—air inlet	
			(1) Dry-bulb	80 F
			(2) Wet-bulb	67 F
		c	Ventilation air	See Note
Cooling	Water-cooled condensers	d	Water temperature entering unit	75 F
		e	Water temperature leaving unit	95 F
	Air-cooled and evaporatively cooled condensers	f	Air entering outdoor air inlet	
			(1) Dry-bulb	95 F
			(2) Wet-bulb	75 F
Heating	All types provided with heating function	g	Unit ambient and total air entering unit	70 F
		h	Heating medium, pressure or temperature	
			(1) Dry saturated steam	16.7 psia
			(2) Water in	180 F
			(3) Water out	160 F
Humidifying	All types provided with humidifying function	i	Unit ambient	70 F
		j	Total air entering unit	
			(1) Dry-bulb	70 F
			(2) Wet-bulb	53 F
Air circulation	All	k	Filters	New and clean

Note: Rating shall be based on both ventilation and recirculated room air entering at 80 F dry-bulb and 67 F wet-bulb temperature. (The *Note* as given in the code has been condensed in order to remove material not pertinent to this chapter.)

conditioners are selected for the usual range of external resistances commonly used with the specific size of conditioner. The larger conditioners are capable of serving larger areas and therefore require fan and motor capacity for the greater external duct resistances that may be imposed on such conditioners.

When locating unitary equipment, floor and beam loadings should be carefully checked. Suspended horizontal units can add 50 to 100 lb per square foot to the loading on the floor above. Should this floor be already heavily loaded, or be a roof structure designed for a 40 lb per square foot snow load, excess beam deflection may occur and cause cracking of plaster or concrete fireproofing. A small fire, normally of little consequence, may cause a rupture of a heavily loaded structure and permit the equipment to drop with extensive property damage. Self-contained units should be carefully installed since their weights run as high as 200 lb per square foot. When they are installed in street floor shops, the extra precaution of placing a column beneath them in the basement is an inexpensive method of reducing vibration, as well as providing insurance against overloaded floor beams.

The services required for operation of unitary equipment should conform to the many restrictive, but necessary, local municipal codes. Existing buildings seldom are wired adequately for the electrical load imposed by the starting of an air-conditioning compressor on any branch circuit. Even the smallest room cooler can draw enough current to reduce the voltage of a lighting circuit to the point where it is visibly apparent. This voltage drop may even affect the life of the unit due to the relatively slow starting. The cost of a separate electrical circuit of adequate capacity from the main panel is more than justified; it is a necessary expense in the majority of installations.

A water supply of adequate capacity and pressure is necessary to prevent overloading of electrical equipment by high head pressures. The average city water supply pressure is adequate for installations up to the third floor. Since most water-cooled units require about 20 psi pressure, including control valve losses, it is important that any units served by gravity from roof tanks be checked carefully if located less than 40 ft below the tank.

Drain connections from condensers should flow to an open and properly trapped sink as required by most city codes. This prevents back pressures on the city water system in the event of condenser failure. A check valve should also be installed in the water supply as a further precaution against contamination.

When installing small remote or self-contained units with outdoor air connections in buildings more than 6 stories high, the effect of wintertime stack action in elevator and stairwells requires special attention. This stack action is the cause of negative pressures on the lower floors, tending to draw cold air through the units, and positive pressures on the upper floors preventing adequate ventilation and disrupting air distribution. It can also cause annoying whistling at door openings and lead to serious complaints in hotels and offices. Wherever the removal of such units is impracticable, it is important that carefully fitted, felt-edged dampers be installed in the outdoor air intakes and that they have adequate locking devices.

One further consideration when installing self-contained units in conditioned areas is to ensure that space is adequate for any maintenance or repairs to be required in future years.

Sound Isolation

Both suspended and vertical floor-mounted units can transmit vibration through the supports. Wherever such transmission of sound might be objectionable, the supports should be isolated through rubber-in-shear or other sound deadeners. (For design of suitable sound deadeners see section Controlling Vibration from Machine Mountings in Chapter 25.)

UNIT AIR COOLERS

Unit air coolers are intended principally for product cooling, but are often used for cooling spaces to low temperatures. They differ from normal air-conditioning units only in features required to produce lower temperatures. In using such units, dry-bulb temperatures should be considered first, after which room moisture conditions and air distribution should

be considered, depending on the product being cooled or stored.

Products to be cooled or stored may be divided into four general classifications based on their sensitiveness to moisture loss.[3] Selection of the proper temperature differential between storage-room air temperature and evaporating refrigerant temperature will produce adequate moisture control for all practical purposes. The differentials required for gravity circulation and for forced air circulation are different, the latter requiring closer differentials. The four classes and differentials which have been successfully used are as follows:

Class 1. Products requiring high room-moisture content, which are highly susceptible to moisture loss and are being stored for extended periods, for example, eggs.

Gravity Circulation	18 F or less.
Forced Air Circulation	6 to 9 F.

Class 2. Products similar to Class 1 but which are being stored only for a limited time as in retail stores where the turn-over is usually 3 days or less, for example, cut meats.

Gravity Circulation	18 to 22 F.
Forced Air Circulation	9 to 12 F.

Class 3. Products requiring moderate room moisture content and which are moderately susceptible to moisture loss, for example, carcass meats and some fruits.

Gravity Circulation	21 to 27 F.
Forced Air Circulation	12 to 20 F.

Class 4. Products from which there is a very low moisture loss or none at all and where room moisture content is not important, for example: canned goods, furs, woolens, bottled goods, dried fruits, foods packaged in good moisture vapor barrier packages.

Gravity Circulation	25 F and up.
Forced Air Circulation	20 F and up.

When using Class 4 application, with finned evaporators and automatic defrosting at each cycle of the condensing unit, it is important to avoid using too wide differentials with very low refrigerant temperatures. In many applications, prime-surfaced evaporators prove unsuitable for this type of defrosting and will require some positive method of defrosting.

Unit coolers can perform satisfactorily in installations requiring accurate control of relative humidity, air motion, and dry-bulb temperature, and thereby prevent excessive weight loss, mold and slime growth, and moisture absorption by hygroscopic materials such as dried fruits.

Unit coolers, especially in the smaller sizes, are very similar in appearance to unit heaters. Copper, aluminum, or steel prime or finned-surface tubes are used. Propeller or centrifugal fans either blow or draw room air over the tubes. The fan and coil are generally enclosed in a casing provided with a drip pan. The motor horsepower requirements are a function of resistance due to coil construction and arrangement, and air volume required. Fin spacing is based principally on operating temperatures. For operation below 35 F, fin spacing is preferably not more than four to the inch. Above 35 F it may run as high as 8 fins per in., depending on room temperature and refrigerant temperature used. Both direct expansion refrigerants and brine are used successfully as cooling mediums.

Unit coolers may be arranged for either free or duct delivery. Face velocities vary, depending principally upon the intended application of the unit. In the larger sizes particularly, speed adjustment of the fan is generally provided to permit variation of the air delivery. While unit coolers are usually installed in the storage space, remote installation combined with appropriate ductwork may be required by space or other considerations. Units are available for floor, wall, or ceiling mounting, thus providing an upward, downward, or horizontal discharge. Power, refrigerant, and drip pan connections are required, plus additional connections for defrosting, if necessary.

For storage temperatures below 35 F, some positive means of defrosting is mandatory. At 35 F or higher storage temperatures, cycling of the condensing unit with low-pressure control at proper settings will provide automatic defrosting at each cycle. At above 35 F design refrigerant temperatures, frost and ice formation will not normally occur.

Various methods of defrosting are used. In one method, hot gas is supplied to the interior of the tubes uniformly throughout the coil. The entire refrigerant circuit is thus contacted to obtain complete defrosting of all frosted surfaces. Electric defrosting generally involves the incorporation of heating elements within the construction of the coil, or the use of strip heaters in a dampered closed-air circuit. Warm water may be sprayed over the coil surface for defrosting. With storage temperatures above 32 F, defrosting may be accomplished by shutting down the refrigeration system and circulating the room air over the coil. In every case, defrosting requires a cessation of refrigeration on the unit being defrosted. Where continuous operation of the system is desired, a brine spray over the coil may be used unless it might damage the product in storage, or cause too much corrosion of room fittings, ducts, etc.

Ratings

Since various means of expressing unit cooler capacity are utilized in the industry, different manufacturers suggest different methods of selection. The engineer should be aware of the conditions and factors which affect rating, selection, and performance of a cooler. These items are discussed in following paragraphs.

The refrigerating capacity of the unit may be either gross or net, the latter being less than the gross by an amount equal to the heat equivalent of the input to the unit cooler motor. In either case, the capacity should be given for a particular air volume. Air throw data are also valuable.

Dry or flooded rating conditions should be stated, as well as temperature level. The temperature level determines whether the coil surface is wetted or frosted, and it will also establish the refrigerant-side pressure drop for any given load applied to a specific unit cooler. The refrigerant-side pressure drop increases as the evaporating temperature decreases, and thus temperature level exercises a significant effect on the average coil-surface temperature and the consequent condensing-unit selection.

Coil capacity rating is usually expressed as total heat absorbed with no distinction between sensible and latent heat. The *rating* expresses the capacity in terms of Btu per (hour) (Fahrenheit degree temperature differential *between the refrigerant and the air*). The term *basic rating* may be used. This is the Btu per hour absorbed with one Fahrenheit degree differential between room air and evaporator refrigerant temperature. When the total load has been obtained from the load calculations, a temperature differential between the air and the refrigerant is selected. It is based on product classification previously discussed. The extent of dehumidification will be a function of this temperature differential. Moisture conditions in the storage space are dependent upon the correct selection of this temperature differential. It offers a quick and sufficiently accurate practical basis for coil selection.

Where close control of relative humidity is desired, heating coils or electric heaters for reheat may have to be added. The use of air-conditioning psychrometric techniques is only required for accurate humidity control.

Cooling equipment is usually rated on the basis of the overall room-to-refrigerant temperature differential instead of using the intermediate average coil-surface temperature. If the tentatively selected unit cooler does not possess the proper capacity, it may be possible to adjust the fan speed to a new air volume. At the new air volume and the new overall differential, the cooler may be able to deliver the necessary cooling capacity. If not, the procedure must be repeated with another size of unit cooler, the final selection being based on the proper balance between unit cooler and the condensing unit to maintain the Btu per hour heat removal and temperature difference at design room temperature.

Procedures for rating and testing room coolers are given in an ASRE Standard[8] which establishes four groups of conditions (numbered I to IV) under which units may be rated. Many manufacturers establish and publish their ratings in accordance with this standard in which forced circulation air coolers are classified according to air side surface conditions as (1) dry coil, (2) sprayed coil, and (3) spray (no coil); or according to type of air delivery to room as (1) free delivery fan, or (2) pressure fan. Natural convection air coolers are classified according to type as (1) external baffle, (2) built-in baffle, and (3) without baffle.

Arrangement and Operation

The refrigerant is usually supplied to the evaporator through a thermostatic expansion valve, thus obtaining dry expansion in the evaporator. In other designs, float-valve feeding is used as a means of metering the refrigerant into the evaporator. The refrigerant circuit in the coil may be of the continuous-flow type from inlet to outlet, or it may have means for internal recirculation of the refrigerant. The coils having the *dry-expansion* or *continuous flow from inlet-to-outlet* circuits may be of the upward feed or the downward feed arrangements. Upward flow is generally regarded as producing more refrigerant holding capacity and more effective use of the surface. However, with some refrigerants and in some systems, the downward-feed system has been found to produce better oil-return characteristics. Many direct-expansion installations use liquid-vapor heat exchangers to increase the coil utilization without undue risk of returning liquid refrigerant to the compressor.

Where two or more evaporator coils are to be attached to a single condensing unit, and different evaporator temperatures are desired, a back pressure valve may be installed to limit the minimum evaporating temperature of the warmer coils. This valve also finds application where fluctuation in the evaporator temperature prevents accurate control of air temperature and humidity.

Except in product precooling, the objective of air distribution is to absorb the heat load as it comes into the room while maintaining a desirable degree of ventilation-air movement through hygroscopic cargoes that are not encased by vapor-barriers. In cold storage practice, this generally requires the securing of as uniform an air distribution as possible to every part of the product zone, with employment of diffusion of air over the product and the prevention of direct blasts against the product. These requirements must be recognized when selecting the unit coolers and their outlets and when selecting the locations for the unit coolers. It is important to note that the quantity of air in motion in the refrigerated space is not only that passing through the coils. A quantity of air many times in excess of the air handled by the units is always set in motion by the induction effect of the moving cooled air. Therefore, in evaluating the velocity for any given area, the total air set in motion must be considered. It is a function of the type of outlet, the discharge air velocity, and the unit location relative to restrictive walls and product.

Unit location is also important from the standpoint of occupant comfort, low-velocity outlets being preferred for floor type units. High-velocity outlets are acceptable in fur storage vaults, ice cream hardening rooms, and other spaces where air motion is not an important factor.

In general, unit air coolers should not be suspended in front of door openings, or close to them where moist warm air will be drawn directly into the unit each time the door is opened, thereby causing excessive frosting and loss of capacity. Better performance will be obtained by placing the unit air cooler where the air will be discharged toward the door. If the shape of the space is such that this location would result in excessive air velocity over the product, then the location of the unit air cooler should be changed so that the air is discharged parallel to the wall in which the door is located.

Control of Unit Air Coolers

Although most unit air coolers can be adapted to any control cycle, continuous fan operation is recommended to avoid stratification and wide fluctuations in space temperature. Should the unit be completely self-contained, control of the direct-expansion refrigeration unit may be obtained from the temperature of the recirculated air and from suction pressure. In the case of multiple unit systems supplied with refrigerant or chilled water from a central source, a valve in the supply to each cooling coil may be controlled thermostatically from space temperature.

REFERENCES

[1] Joint Committee of the American Society of Refrigerating Engineers, American Society of Heating and Ventilating Engineers, Refrigerating Machinery Association, National Electrical Manufacturers' Association, and Air Conditioning Manufacturers' Association: *ASRE Circular* No. 13-42.*

[2] Joint Committee of the American Society of Refrigerating Engineers, American Society of Heating and Ventilating Engineers, Refrigerating Machinery Association, National Electrical Manufacturers' Association, and Air Conditioning Manufacturers' Association: *ASRE Circular* No. 16.*

[3] *ASRE Standard Methods of Rating and Testing Forced-Circulation and Natural Convection Air Coolers for Refrigeration* (*ASRE Circular* No. 25-44).

[4] *Room Air Conditioners* (*Air-Conditioning and Refrigeration Institute Standard* 110-56).

[5] *National Electrical Code* (*National Board of Fire Underwriters Pamphlet* No. 70, Paragraph 195).

[6] *American Standard Safety Code for Mechanical Refrigeration* (American Standards Association, B9.1-1953).

[7] *Standard for Room Air Conditioners* (Underwriters' Laboratories, Inc., UL484, September 1957, 2nd ed.).

[8] *Methods of Rating and Testing Air Conditioners* (*American Society of Refrigerating Engineers Standard* 16-56).

[9] *Cooling Load Estimate Form for Room Air Conditioners* (*Air-Conditioning and Refrigeration Institute Standard* 120-56).

[10] *Air Conditioning Refrigerating Data Book—Applications* (American Society of Refrigerating Engineers, 1956-57, Chapter 34).

* ASRE has combined *Circulars* No. 13-42 and No. 16 in *ASRE Standard* No. 16-56, *Methods of Rating and Testing Air Conditioners.*

CHAPTER 17

ELECTRIC HEATING

Resistors, Heating Elements, Electric Heating Units, Electric Space Heating Applications, Types of Electric Heating Systems, Equipment and Installation Methods, Calculating Capacities, Induction and Dielectric Heating, Power Considerations

ELECTRICITY as a source of heat represents thermal energy in a refined form, easily applied to space heating, and readily distributed and controlled. For many applications, including use to supplement heating systems of other types, the compactness, simplicity, responsiveness, accuracy of control, safety, and cleanliness of electric heating may carry greater weight in choice of method than operating cost and initial investment. Electric heat often is more expensive on a direct-heat-equivalent basis than heat from conventional fuels. Economical service requires careful application in the design of system, adaptation of building structure, choice of control devices, and in method of operation by the user.

Using the power-to-heat conversion constant, 1 kilowatt equals 3,413 Btuh, resistance heating calculations are expressed directly in terms of kilowatt power requirements.

Definitions of *Resistor, Heating Element*, and other terms applying to resistance electric heating practices will be found in Chapter 1.

All equipment, materials, and construction used in electric heating systems should comply with the requirements of the *National Electrical Code*, with local codes, and be listed as approved by Underwriter's Laboratories Inc., or other recognized certifying agency.

ELECTRIC HEATING UNITS

An electric heating unit is a frame, casing, or other supporting means containing one or more heating elements, electric terminal connections or leads, and electrical wiring and insulation, all assembled into a unit. There are many varieties of housing, material, and arrangement of heating elements, operating temperatures, ratio of radiant to convective heat delivery, directional control for radiation, natural or forced convection, automatic or manual control, and electric load limiting features. Many specific types of units are discussed in the section Equipment and Installation Methods.

RESISTORS AND HEATING ELEMENTS

Electric resistors usually are composed of metal-alloy wire or ribbons, non-metallic carbon compounds in rod or other shapes, or printed circuits. Heating elements may have exposed resistor coils mounted on insulators, metallic resistors embedded within refractory insulation encased in a protective metal sheath, or a printed circuit encased in glass sheets. Fins or extended surfaces may be used to add heat-dissipating area. Elements are made in many forms. Strip elements are used for clamping to surfaces for heat transfer by conduction in some convection air heaters and some low temperature radiant heaters. Ring and plate elements are common in electric ranges and many small air heaters. Metal or oxide conductive films on glass and ceramics have been used usually in panel form. Tubular elements may be immersed in liquids, may be used bare, may be formed into coils, or may be cast into metal or plastic, and used in water heaters, ranges, and air heaters.

Cloth fabrics, incorporating flexible resistor wires, are employed for many low temperature purposes such as heating pads, sheets, blankets, aviators' clothing, and some radiant panel heating installations.

Special incandescent lamps with tungsten or carbon filaments and glass or quartz envelopes are designed to produce maximum energy in the infra-red portion of the spectrum and applied as radiant heaters.

ELECTRIC SPACE HEATING APPLICATIONS

Complete electric heating systems are now finding wide application not only in residences but also in many commercial and industrial establishments. Auxiliary electric space heating is often used for convenience, for instance: (*a*) for local use when the main heating system is shut down, (*b*) for local use where the main heating system is inadequate, (*c*) for temporary heating, (*d*) for special conditions involving separate or remote control, (*e*) for isolated locations, (*f*) for cases where minimum initial cost is the dominant factor. Portable units readily satisfy some of these requirements.

Compact electric heating units are sometimes installed in main supply or branch ducts of central-fan steam and water systems to provide the final temperatures and relative humidities required for comfort or process air conditioning. Electric heaters installed for use primarily in the heating cycle can also be utilized for reheat in the cooling cycle.

In control of systems, the flow of air over the electric heaters should be constant while the input to the heaters should be varied. This is necessary because of the basic difference between electric heating and other methods such as steam heating. Steam is approximately a constant temperature source of heat at any given pressure, and, consequently, a change in air flow over a steam coil does not greatly alter the coil surface temperature. Hence, the heat output of a steam coil varies in proportion to the air volume, but the surface temperature of the coil remains about the same. With electric heat the energy input and output are constant, and if the electrical energy input remains constant, the temperature of the heating elements will vary inversely with the air flow. Therefore, face and bypass dampers should not be used in electric heating, because the reduction in air flow and the corresponding rise in heater temperature would endanger the operating life of an electric heating element.

Automatic regulation of the electrical input is usually accomplished through a proportional-type step controller that energizes or de-energizes heaters in increments small enough to prevent excessive cycling. The electrical input per heater, or step, is normally determined by the permissible change in supply air temperature per step. High-limit thermostats are sometimes used to de-energize the heaters if the heater tem-

Table 1 . . . . Principal Types of Electric Space Heating Systems

A. Radiators, Convectors, Unit Heaters—Built-In and Portable
   1. Metallic resistor, high or low temperature
   2. Unit ventilator
   3. Baseboard-type resistor
   4. Resistor with fan, including unit heater
   5. Resistor unit placed in room-type air conditioner
   6. Resistor with focusing radiation reflector
   7. Steam radiator with immersion electric element
   8. Floor furnaces
B. Panel-Type Installations
   1. Ceiling panel
      a. Rigid composite panels containing electrically conductive rubber
      b. Flexible composite sheets containing embedded metallic filaments or conductive rubber
      c. Electric conductors embedded in plaster on lath
      d. Electric conductors embedded in plaster or similar material between two layers of plaster board
   2. Wall panel
      a. Rigid panels, as for ceilings
      b. Flexible sheets, as for ceilings
   3. Floor panel
      a. Electric conductors embedded in concrete slab
C. Central Hot Water Systems
   1. Water from electric heater pumped to radiators or convectors in rooms
   2. Same, with thermal storage for off-peak operation
   3. Fuel-fired boiler converted to electricity
   4. Heat pump systems—See Chapter 39
D. Central Warm Air Systems
   1. Resistance-element bank in housing or air duct
   2. Fuel-fired furnace converted to electricity
   3. Heat pump systems—See Chapter 39

peratures become excessive for any reason such as a reduction in air flow. Heaters must be de-energized on fan shutdown. (See Chapter 43 for general information on controls.)

TYPES OF ELECTRIC SPACE-HEATING SYSTEMS

The inherent adaptability of electric energy for transmission, conversion, and regulation, and the relatively high unit cost in terms of thermal equivalent, as compared with fuels traditionally used for space heating, necessitates and justifies greater attention to efficient utilization.

Types of electric heating equipment and complete heating systems in current use are listed in Table 1. The sequence shown is for convenient reference only, and does not indicate the relative extent of use, quality of performance, or installation cost and operating expense. Installations of all types listed are in successful operation, but performance and cost depend in large measure on proper application to the local conditions.

EQUIPMENT AND INSTALLATION METHODS

Convector with Metallic Resistors

Heating units for wall mounting, recessed or surface type, are made with resistors of incandescent bare wire or lower temperature bare wire or sheathed elements. An inner liner or reflector is usually placed between resistors and casing to provide a secondary air passage promoting circulation and minimizing rear casing temperature. Depending on details of unit construction, the convection heat delivery constitutes between 40 and 70 percent of total output, with the remainder dissipated into the room as radiant heat.

Ratings usually are from 1000 to 8000 watts (3400 to 27,300 Btu per hr) with some models down to 500 and up to 15,000 watts. Voltages are the standardized values of 120, 208, and 240. Similar models equipped with air circulation fans are available, giving convection heat delivery up to 90 percent. Other types, using similar components but without a fan, are arranged as *floor furnaces* to be installed between floor joists. A manual switch and thermostat integral with the unit are usually provided.

The location of electric convectors should be governed by the same principles that determine arrangement of steam or hot water convectors with respect to air movement.

Small portable units are available with ratings up to 1650 watts for operation at 120 volts. Higher wattage units are made for 208 or 240 volt application, and require suitable heavy duty receptacles.

Unit Ventilators

Unit ventilators with electric heating elements of the metal sheath type with extended surface are used to provide heating and ventilating of school classrooms. Overheat switches are usually provided for each element in the heating section to de-energize the elements in case of interrupted fan operation. Ratings are available up to 30 kw in the larger sizes of units. Auditorium unit ventilators of the same type are rated up to 300 kw. The total capacity of the unit ventilator is made up by numerous smaller heating elements so temperature control can be accomplished by energizing or de-energizing the elements one at a time. The application of unit ventilators is described in Chapter 15, Unit Ventilators and Unit Heaters.

Unit Heaters

Electric unit heaters include a suitable built-in fan which circulates room air over heating elements; they are employed for the same uses as other forms of unit heaters (steam, hot water, gas, oil), if conditions are favorable to electric heating. They are especially adaptable for supplemental heating, for small occupied rooms in otherwise unheated buildings and for unattended equipment enclosures in which temperature must be maintained above freezing. Location and arrangement of such unit heaters are discussed in Chapter 15.

Baseboard-Type Unit

Metal casing proportioned to resemble and replace conventional baseboard along walls contains one or more resistors placed horizontally. These should be rated from 80 to 250 watts (270 to 850 Btuh) per linear foot of baseboard unit. Resistors may be bare wire, sheathed element, cast grid, finned, or ceramic extended surface. The vertical dimension is usually 4 to 9 in. and projection from face of wall from 1½ to 3½ in. Location follows the same principles applying to hot water type baseboard installation described in Chapter 14. The maximum surface temperature of the housing should be limited to 190 F.

Baseboard-type units using glass panels 6 to 8 inches high and 30 to 42 inches long are available for operation at maximum surface temperature limited to 250 F.

Large high-wattage unit-type heaters are available for ceiling or high wall mounting. These are similar to steam and hot water unit heaters in appearance and function. Resistors may be placed in room air conditioners to permit operation as forced air heaters when desired.

Resistor Portable Heater with Focusing Reflector

An example of the focusing-reflector-type heater is the parabolic or circular bowl having, in its center, an incandescent wire coil resistor on a ceramic core, often screwed into a porcelain or other approved type receptacle. Usual ratings are 500 to 1650 watts for 120 volt use. Heat emitted is 75 to 85 percent radiant.

Sectional focusing-trough-type reflectors are made with low temperature resistors or with *quartz lamps* to be suspended from the ceiling or bracketed from the wall. Rated capacities are 200 to 1500 watts per linear foot of unit. Installations of this kind are usually to handle the perimeter heat loss. In a modified form, similar units are employed for infra-red heating and drying in industrial processes.

Steam Radiator with Immersion Element

Radiators equipped with an immersion element screwed into a bottom opening, are usually sealed after charging with water and antifreeze compound. A safety valve or thermal relief plug is provided to protect against excessive pressure and temperature in case heat dissipation is unduly curtailed by obstructions to air convection. The steam-and-water intermediate heat transfer does not affect overall thermal efficiency, but gives the unit operating characteristics adapted for certain applications. Units may be permanently connected by conduit, or used as portable heaters with cords.

Ceiling-Panel Installations with Conductive Rubber

Panels of laminated construction containing electrically conductive rubber, with rating of 22 watts (75 Btu) per sq ft, for 115 and 230 volt service, are built in standard sizes 3 x 4 ft, 4 x 4 ft, and 4 x 6 ft. They consist of a layer of special rubber about 1/64 in. thick sealed between layers of phenolic resin-impregnated electrical insulating paper. The assembly is covered on each side with a sheet of aluminum foil. Overall thickness is 1/16 in.; weight is 0.4 lb per sq ft. Metal strips embedded along edges of the rubber sheet serve as potential leads. Electric terminal blocks are provided on each panel and standard raceway moldings are furnished for wiring connections. An earlier type of construction using asbestos-board backing to form rigid panels ¼ in. thick has been discontinued.

Under normal operating conditions, the surface temperature of an automatically controlled ceiling panel is about 100 F. When panels are turned on in a cold room or with continuous operation, the maximum temperature reached is 120 F.

Since it is not permissible to cut heating panels, the entire pattern for ceiling must be planned in advance, with particular attention to coverage near exposed walls. Attachment to smooth surfaces may be made by a special adhesive applied on the ceiling along a 1½ in. fastening margin provided around the four sides of each panel. A highly sensitive room thermostat with narrow operating differential should be used in each room.

Ceiling-Panel Installations with Embedded Conductors

Resistors or cables for embedding in ceilings to form panel-heating installations are electrically insulated with coverings resistant to high temperature, water absorption, aging effects, and chemical action with plaster, cement, and soil. Cable units identified by a color code are furnished in nominal standard lengths from about 75 to about 1800 ft, and are rated from 200 to 5000 watts for standardized voltages of 120 or 240. The outside diameter of the cables is usually about 1/8 in. Non-heating leads 7 ft long are attached to each unit to connect at thermostat or connection box without running cable beyond the ceiling. The length of cable or leads must not be altered. The identification label showing rating may not be removed. Manufacturer's instructions should be followed regarding installation procedures.

The lath used is of non-metallic fire-resistant type. The cable is secured to the lath with staples spaced not over 16 in. apart. Each cable is tested after fastening on lath, and after first plaster coat has been applied, for continuity of circuit and for insulation resistance of at least 100,000 ohms measured to ground. The minimum spacing between cable passes is 1½ in. and is generally used near cold walls, with up to double that spacing near interior areas of the room. Non-heating leads outside the ceiling should be installed by approved wiring methods. Cables must be kept away from metallic materials and from areas of ceiling additionally heated by recessed or surface mounted lighting fixtures. All general power and light wires should be run above thermal insulation or at least 2 in. above the heated ceiling surface.

For plastered ceilings, non-insulating plaster is applied in two or three coats with the first coat troweled in the same direction in which the cable runs. Ceiling cables, properly engineered, installed, and operated will not cause ceiling cracks. While new plaster is drying out, the system should not be energized and the range and rate of temperature change should be kept low by other heat sources or by ventilation until plaster is thoroughly cured. The use of vermiculite or other insulating plaster is contrary to code provisions since it causes cables to overheat.

For laminated ceiling construction, cables are stapled to non-metallic lath as for plaster, and then a second layer of non-insulating ceiling board is applied with a layer at least 1/8 in. of plaster or similar compound to sandwich the cable firmly in place and improve heat transfer.

A highly sensitive room thermostat with narrow operating differential is recommended for each room. Chapter 30 includes data on panel heating with electric panels.

Electric Wall Panels

Cable embedded in walls similar to ceiling construction is occasionally found in Europe. Because of possibility of damage due to nails driven for hanging pictures or from building alteration, most codes prohibit such panels in the United States. Interference with radiation caused by furniture placement and interior decorations reduce the desirability of wall panels.

Radiant Convector Wall Panel

Glass electric heating units, sometimes designated as radiant panels, depend on the heating effect produced by passage of current through a thin coating of conductive material fused to one face of a panel of ¼-in. thick special glass. The conductive layer may be sprayed-on aluminum or *printed* metallic oxide patterned to form a grid several thousandths of an inch thick, or fused-on material to give a uniform coating less than 0.0001 in. thick over the entire active face of panel. Normal glass operating temperatures are 300 to 400 F, with maximum permissible temperature around 650 F.

Electric wiring connections are made by a variety of means

to the panel, with leads provided for easy field connection. The panel is supported on insulators within a metal frame, with a reflector behind the glass arranged to provide space for air circulation. Protective guards are usually provided to reduce hazards to a minimum.

Glass units are usually rated between 600 and 3000 watts, for standardized voltages of 120, 208, or 240 volts. Starting current for some types exceeds normalized current by 10 to 50 percent. However, there are types with substantially no current surge. Frame sizes vary from about 30 x 24 in. for 1000 watt size to 42 x 6 in. for baseboard models, and are arranged for recessed or for surface mounting. Thermostats integral with the unit are optional. Portable units with cord are available.

Radiant panels using tubular elements built or cast into extended aluminum panels have emissivity characteristics similar to glass panels.

Electric Floor Panels

Resistor cables of the same type used for ceilings are used for floor panel systems in concrete slabs. Precautions to be observed in construction of slabs for panel heating are covered in Chapter 12 and 30. Non-insulating cement finish 1⅝ in. thick is poured and troweled above cables fixed to a minimum 3 in. slab. In some areas, cables may be stapled to wood nailing strips fixed in the surface of the rough slab. When the slab is made of light-weight insulating concrete, cables may be stapled directly to the slab. Alternative special anchors are available to hold the cable at proper spacing during pouring, and are held by nails driven by hand or powder impact drivers. Otherwise, periodic temporary fastening with daubs of cement, plaster of Paris, strips of masking tape, or other nonconductive material must be used.

With monolithic finish, nailing strips are omitted and the cable is strung on frames with nail spacers; frames are removed when enough concrete has been poured to hold the cable in place. If desired, magnesite flooring, ceramic tile flooring, asphalt tile, or wall to wall carpeting may be used on the floor. Insulating-type concrete may be used below cable, but must not be used around or over cable.

Central Hot Water Systems

Heating systems of hot water type using radiators or convectors, discussed in Chapter 28, may be operated using an electric hot water boiler containing immersion elements (resistors). Resistors may be interlocked to prevent energizing when the circulating pump is not operating. The boiler water holding capacity should be sufficient to minimize cycling.

For off-peak operation where electric rate considerations justify, a water heating tank of large storage capacity may be employed. The system may be designed for a water temperature of 250 to 275 F at pressures up to 75 psig, with suitable piping systems for this pressure. An automatic valve may provide 140 to 160 F water at the pump by mixing hot water from the tank with cooler water from the return main. Another method employs the *flash* principle, withdrawing water at high storage temperature into a low-pressure separating chamber where steam is obtained as a result of the pressure reduction; however, power for pumping is substantially greater with this steam-accumulator method. Thermal insulation on boiler tank and piping must be adequate to minimize losses. High investment, space required for equipment, and loss of much of the convenience and economy of operation available with other forms of electric heat, tend to limit the application of this type of system.

Central Warm Air Systems

Systems employing resistors mounted in a frame or housing may be fitted into ducts with a fan forcing the air to the various spaces to be heated, or may replace the heat exchanger of fuel-fired furnaces to convert them to electrical operation. Much of the convenience and economy of operation available with electric heat is lost by such central system applications.

Heat Pump Systems

Heat pump systems are discussed in Chapter 39. Heat pump systems using air as a heat source lose capacity rapidly as the outdoor temperatures decrease, while water and earth heat source pumps sized for cooling also frequently reach the *balance point* at outdoor temperatures above outdoor design. Deficiency at temperatures below the *balance point* can be supplied by supplementary electric resistance elements. When these supplemental heaters are placed in individual spaces rather than centrally, the electrical demand is minimized, the individual spaces may be maintained at slightly different temperatures, and local reheat during the cooling cycle is readily available. An alternative provision for cold weather peaks is thermal storage in chemical salts and by other devices. Such processes are still in the development stage.

The installed cost of residential heat-pump systems is higher than that of resistance types. Resistance heating systems, except the minority using air circulation through ducts, do not lend themselves to consolidation with summer air conditioning. Use with an independent cooling installation may be preferable and attained with lower investment cost.

CALCULATING CAPACITIES

The procedure outlined in Chapter 12 for calculating the heating load may be used for electric systems. Load expressed in Btu per hour is converted to kilowatts by the divisor 3413 Btu per kilowatt. The National Electrical Manufacturers Association has published the *NEMA Manual for Electric House Heating* describing methods and giving heat loss factors for calculating load directly in kilowatts. All the energy applied to a resistor transforms itself into heat, unaffected by temperatures of the surrounding air and of surfaces receiving radiated heat. However, both electric power input and heat output are directly affected by voltage at the resistor terminals, being proportional to square of the voltage. Thus, a resistor rated 1000 watts at 240 volts, if used on a circuit at 220 volts, delivers 840 watts or 16 percent under the rated value, while at 208 volts the shortage is 25 percent.

The arbitrary addition of a percentage *safety margin* on the calculated heating load is not recommended. Such practice would increase the peak electric demand and often the cost of electric service.

The most economical electric heating systems from an operating standpoint are of decentralized type, with a thermostat provided on each unit or for each room. This permits each room to compensate for heat contributed by auxiliary sources such as sunshine, lighting, and appliances. This arrangement also gives a better diversity of the power demand due to non-coincidence of electric load from all units of an installation. Manual switches are often provided to permit cutting off heat or reducing temperature in rooms when not in use. When such operation is practiced, consideration should be given to provide adequate capacity for warm-up as compared to a system maintaining a constant temperature.

For sake of economy buildings intended to be heated electrically should be well constructed, have adequate thermal insulation and storm windows or double-glazed windows in more severe climates to minimize heat loss and weatherstripping to minimize infiltration. The relatively high electric cost for thermal energy, about $5.40 per million Btu at $0.018 per kwhr, for example, as compared with about $1.50 for oil at $0.16 per gal when utilized with 70 percent efficiency, gives economic justification for substantial expenditure to reduce heat consumption. The large majority of dwellings heated economically by electricity have heat factors (quotient of seasonal energy consumption in kilowatt hours, divided by degree days and by volume of gross heated space expressed in thousands of cubic feet) between 0.15 and 0.3. The features necessary to hold consumption within these limits result in construction that for 5000 degree-day climate with 0 F outdoor design temperature, gives a calculated heating load of 3400 to 5100 Btu per hr per 1000 cu ft gross volume. Heat factors up to 0.4 are not uncommon, but experience shows that such high heat requirement may result in excessive operating cost. General practice is to provide adequate thermal insulation with electric heating installations.

POWER CONSIDERATIONS

Rates for Electric Service

The cost of electricity varies due to several factors. The main elements are the annual charges, including taxes, on capital invested and those expenses necessary to keep the electric utility system at all times in a state of *readiness to serve* the load. The cost of generating the energy is secondary. Electricity is not stored, but must be produced instantaneously when and as required by the user. Consequently, the time of power use, both daily and seasonally, and relation between rate-of-use or demand and the energy consumed within a period (load factor) have large effects on costs and the rates charged. Special low rates sometimes are available during certain prescribed *off-peak* hours of use, when the system load is low.

Space heating is a load whose magnitude is determined basically by weather, and is affected to a much lesser extent by the electric customer's use of his premises. Most of the heating loads in a region occur simultaneously, thereby tending to create peaks for which the capacity of the electric system must be adequate. Accordingly, rates contain, in one way or another, charges both for demand and energy. Demand may be indicated directly by a demand meter or be derived from manufacturers' rating data shown on nameplates of the heating equipment. In block-type rates, where demand is not specifically mentioned, it is in part reflected by higher charges per kilowatt-hour in the earlier blocks.

Control of Electric Demand

To obtain electricity for space heating at minimum cost, it is necessary to keep the user's kilowatt demand as low as possible, by provisions in original design of the installation and by judicious methods of operation. Excess heating capacity should be avoided, but without jeopardizing satisfactory performance in cold weather. A variety of control methods can be applied to minimize both demand and consumption. Decentralized control with a thermostat in each room or on each heating unit, together with manual switch to cut off any unoccupied room, is one method. Sequence switching whereby electric service to individual rooms or circuits is shifted in rotation by an automatic timing device and sequence is modified by outdoor thermostat or by total electric load, is another.

Load-limiting controls of several types have come into use. These are generally arranged to measure the customer's total power demand, which for a residence may be grouped as lighting and miscellaneous appliances, refrigerator and water heater, cooking range, and the space heating system. When the demand exceeds a preset value, one or more heating circuits are cut off progressively in rooms least affected by the interruption. Experience shows that with electric floor or ceiling panel heating, temperature drop occurs slowly, at the rate of about 1 deg per hour for concrete slabs and 2 deg for plaster construction or rigid panels properly backed with thermal insulation. Moreover, electricity consumed by lighting, appliances, and cooking ranges is contributing some useful heat at such times.

Another form of limiting control for 120/240-volt 3-wire circuits provides complete or sequence transfer of space heating units from 240 volts to 120 volts, thereby reducing the electric input to one quarter whenever the total load of general-service component exceeds a preset limit. This method lends itself to use of an outdoor thermostat actuating a relay, whereby in mild weather the entire heating system operates at 120 volts and with lengthened cycles obtains more uniform room temperature. For reducing temperature at night, if desired, a clock-operated master thermostat can be provided to lower the heating-system voltage. On some electric utility systems, centralized control by means of a pilot-wire or carrier-current actuating a relay is applied to house-heating installations.

With any type of control, time-delay relays should be used in order that upon restoration after an emergency service interruption or whenever a master thermostat calls for heat, the individual circuit or units will come on non-simultaneously over a period of a few minutes. This is advisable especially for types of resistor units that have a power input, when cold, as much as 50 percent above the rated value at normal operating temperature.

Operating Costs

With increased numbers of electric heating installations, operating cost estimates and experience records are accumulating. Experience records are indicating that actual consumption may be less than conventional calculations and comparisons would indicate. This is particularly true with thoroughly insulated structures or for applications having short hours of maintained temperatures and long reduced temperature periods.

The *NEMA Manual for Electric House Heating* applies the following formula for estimating the cost of electric heating:

$$\text{Annual kilowatt-hour consumption} = \frac{HL \times DD \times C}{TD}$$

where

C = constant (see text below).
HL = heat loss of building, kilowatts.
DD = annual degree days for area.
TD = difference between indoor and outdoor design temperature, Fahrenheit.

The constant C depends on a number of variables such as weather conditions in the locality, orientation, design and construction of the building, living habits of the occupants,

Table 2 . . . . Relation of Resistor Voltage to Heat Delivery

Condition	Volts	Heat Delivered %
Electric system, nominal value	120/240	103.3
For design of equipment, at terminals	118/236	100
Range, as basis for design		
Minimum	110/220	86.8
Maximum	124/248	110.5
On secondary distribution system		
Favorable zone, as to voltage conditions		
Minimum	110/220	86.8
Maximum	125/250	112.0
Tolerable zone		
Minimum	107/214	82.2
Maximum	127/254	116.8
Emergency conditions	90/180	58.3

design of heating system, and internal sources of heat. Experience with many thousands of electric heating installations has shown that a conservative value for C is 18.5. Values of C as low as 12 have been reported on individual installations. It is recommended that a value of C of 18.5 be used unless local experience of a statistical nature has established a more reliable value for the particular area concerned in the estimate.

For block-type electric rates, multiplying the annual kilowatt-hour energy consumption by the cost per kilowatt-hour indicates the annual cost. For demand rates, the assistance of the local utility company should be secured to estimate demand and energy uses by the month on the particular rate involved. It may be pointed out that residences with thermostatic control *in each room* generally establish a maximum of 75 percent of the total calculated heat load as the highest demand requirement for heating.

Voltage Requirements

The preferred nominal system voltage at point of electric utilization by equipment, for single-phase 3-wire systems necessary with space heating, is 120/240 volts, as stipulated by standards of the electrical industry (*EEI Publication* R-6 and *NEMA Publication* No. 117 issued May 1949). For household heating appliances mentioned in the standards, such as air heaters, water heaters, and cooking ranges, the *equipment voltage rating for design* is specified as 118/236 volts, and the range of *voltages to be used as a basis of design* extends from 110/220 minimum to 124/248 maximum. However, natural variations exist from time to time in the conditions at different points in any secondary distribution system and will affect both voltage level and range of fluctuations. Under emergency conditions on electricity supply systems, voltages of the order of 90/180 may be encountered. Heat delivery by resistors with these voltage values, expressed in percent of rated delivery with the normal 118/236 volts (at terminals) is given in Table 2.

Voltages stipulated for secondary distribution systems are at point of service entrance to the building; the drop of voltage in the house supply wiring to terminals of the heating equipment may be 2 to 3 percent, thus reducing heat delivery by some 5 to 8 percentage points below the *favorable zone* and *tolerable zone* values included in the last column of Table 2. Accordingly, it is necessary the designers of electric heating installations obtain specific information from the local electric utility company on both existing and anticipated future conditions at the location, before specifying the capacity and voltage for equipment, wiring, and controls.

BIBLIOGRAPHY

W. F. Friend: Electric house heating—load characteristics and economics (*Midwest Power Conference Proceedings*, April 1951).

E. E. Parks: Electric house heating (*Electrical Engineering*, August 1951).

Electric House Heating (*Rural Electrification Administration Bulletin* 142-1, December 1957).

F. A. Compton: *Complete Electric House Heating* (*Edison Electric Institute Bulletin*, May 1950).

C. E. Simpson: House heating experience (*Electrical World*, October 9, 1948).

B. H. Martin and T. W. Newberry: Heating by electricity in Tennessee valley area (*Heating and Ventilating*, May 1948).

H. G. Kelsey: Longview house heating data (*Electrical West*, September 1945).

W. B. Morrison: Electric storage heating serves new Oregon school (*Heating, Piping and Air Conditioning*, June 1949).

R. E. Sinclair: Electric house heating load characteristics (*Electrical West*, September 1950).

J. B. Cochran: House heating load characteristics as they affect wiring costs (*Electrical World*, April 12, 1947).

H. C. Bender: Heat factor formula to calculate electric house heating (*Electrical World*, May 12, 1945).

W. F. Friend: Modulating and load-limiting controls for electric house heating (*American Power Conference Proceedings*, March 1953, and *Heating and Ventilating*, August 1953, p. 82).

Methods developed for built-in radiant heat (*Electrical West*, December 1948).

R. J. Lorenzi and J. F. Schreiber: Performance of electrical system of panel heating with four stages of insulation (*Heating, Piping and Air Conditioning*, January 1949).

L. N. Roberson: Radiant heating by electricity (*Heating and Ventilating*, September 1946, p. 89).

R. S. Tice: Low-voltage high-current radiant heat (*Electrical West*, December 1947).

Applications of radiant energy (Illuminating Engineering Society, *Lighting Handbook*, 1952, Section 18).

Radiant Glass Heating Panels (National Bureau of Standards, *Technical News Bulletin*, May 1953).

P. R. Achenbach: Radiant glass heating panels (*Heating and Ventilating*, January 1953, p. 83).

EEI-NEMA Preferred Voltage Ratings for A-C Systems and Equipment (*Edison Electric Institute Publication* No. R-6, May 1949, and *National Electrical Manufacturers Association Publication* No. 11, May 1949).

Standard Handbook for Electrical Engineers (McGraw-Hill Book Co., New York, 1952).

R. E. Sinclair: Short method for estimating electric house heating load (*Air Conditioning, Heating and Ventilating*, January 1955).

R. L. Boyd: Heat schools electrically? (*Heating, Piping and Air Conditioning*, December 1956).

NEMA Manual for Electric House Heating (National Electrical Manufacturers Association, June 1957).

J. C. Beckett: Cost comparison: resistance space heating vs fuel-fired systems (*Electrical Construction and Maintenance*, October 1956).

Space heating: what happens in the 6,000 degree-day zone (*Electrical World*, March 19, 1956).

W. R. New: Serving the all-electric home (*Electrical World*, March 19, 1956).

R. L. Boyd: It's here . . . the all electric school (*Electrical Construction and Maintenance*, February 1957).

Handbook of Electrical Applications (Edison Electric Institute).

W. J. Novak: Electricity and combustible fuels (*Electrical Construction and Maintenance*, April 1957).

CHAPTER 18

WARM AIR HEATING SYSTEMS

Forced Warm Air Systems; Air Distribution; Supply Outlets and Return Grilles; Duct Construction; Simplified Methods of Design for Perimeter Systems of Loop, Radial, and Extended-Plenum Types; Return Duct System for Perimeter Installations; Inside Wall Delivery Systems; Design of Large Systems; Automatic Controls; Adjustment of System; Warm Air Ceiling Panel Systems; Summer Operation; Gravity Warm Air Systems

WARM air heating systems may be conveniently divided into two classifications depending upon the manner in which the motive power for circulating the warm air is supplied. In *gravity* systems, the motive head is due to the difference in weight between the heated air leaving the top of the furnace casing and the cooler return air entering the bottom of the casing. In a *forced* warm air heating system, all or part of the motive head is supplied by a fan. Although a great many gravity warm air heating systems are in use, only a limited number of new gravity installations are now being made. Emphasis will therefore be placed upon the design and installation of forced warm air heating systems.

FORCED WARM AIR SYSTEMS

In forced warm air heating systems, the air circulation is effected by motor-driven centrifugal fans, commonly referred to as blowers. The advantages of forced air systems are:

1. The furnace may be placed in any part of the structure.
2. Distribution ducts can be made small enough to be inconspicuous and out of the way, or be completely concealed from view where desired.
3. Circulation of air is positive, and in a properly designed system, can be controlled in such a way as to give a comfortably uniform temperature distribution.
4. Humidity control is readily attained.
5. The air may be cleaned by filters or other means.
6. If properly designed or suitably adapted, the same air distribution system can be used for summer cooling as for winter heating.
7. The use of the fan permits flexibility in the location of supply and return grilles as required to obtain proper distribution of air for comfort.
8. Ventilation air may be positively introduced and conditioned.

The construction features of forced warm air furnace units and the function and selection of the various parts of a system are discussed in Chapter 35 and other publications.[1]

AIR DISTRIBUTION

The conditions of comfort obtained in a room are influenced greatly by the type of supply outlet used, and the locations of the supply outlets and return grilles. In general it has been found that changes in the type, air velocity, and location of the supply outlet affect the room conditions much more than the changes in the location of the return grilles. Three methods of locating outlets and grilles are in common use. One method is to locate the supply register near the floor so that the warm air from the register *blankets* a cold wall, and mixes with the cold air descending from the exposed walls and glass. A second method is to locate the supply openings near the floor, or high in the side wall, on the inside wall, and the return openings near the greatest outside exposure. A third method is to locate all supply openings around the outside wall, near the source of the greatest heat loss, usually beneath the windows, and to use grilles designed to blanket the cold area. This causes mixing of the warm air delivered with the cool air from the heat loss area and the cold air from infiltration, thus effectively preventing drafts. It has been called *perimeter* heating.

In any case, the warm air registers should be so located and so designed that the air stream never discharges directly against people at rest. Tests[2] in Warm Air Research Residences No. 1 and No. 2 at the University of Illinois, have indicated that nearly continuous blower operation gave better results than intermittent operation. This type of operation is more commonly achieved by suitable adjustments of air quantity and fan switch settings than by continuous operation of the blower.[3]

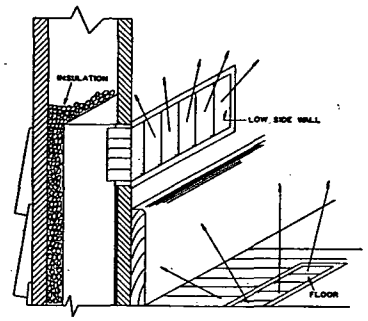
Supply Outlets and Return Grilles

The type of supply outlet that should be used depends upon the type of distribution system to which it is applied. With perimeter duct systems, the supply outlets are usually located in the floor, in the baseboard, or low in the side-wall and underneath a window. In order to be most effective, perimeter diffusers must deliver the air upward and in a fan-shaped pattern so as to blanket the window or cold wall with warm air, thus mixing with and tempering the cold air which usually descends along a cold surface. Furthermore, since delivery of the air outward into the room, rather than upward, is likely to impinge upon the occupants it should be avoided. With certain types of perimeter diffusers, the air velocity at the diffuser face is limited only by the pressure available at the diffuser and the noise characteristics of the outlet.

Tests with *inside wall* supply systems conducted in Warm Air Heating Research Residence No. 1 have indicated that comparable results are obtainable with either high side-wall or baseboard registers, if proper registers and air velocities are selected.

Baseboard registers should be of a deflecting-diffuser type that will direct the air downward toward the floor and diffuse it at the same time. For baseboard registers, air velocities over 500 fpm should be avoided as they may cause discomfort.

High side-wall registers should be of a type that will deliver the air horizontally or in a slightly downward direction, and should be so located as to avoid impingement of air on ceiling or wall. Directional flow diffusing type regis-



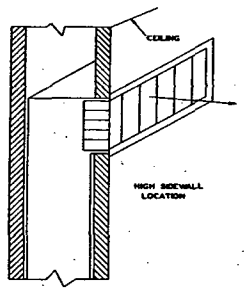
[a] *Registers set to direct air upward along the wall at as wide an angle as possible.*

Fig. 1.... Recommended Type of Floor or Low Side-wall Installation on Outside Wall[a]

ters should be used to insure best results. Register air velocities should be such that the air velocity will be about 50 fpm three-quarters of the distance from the register to the opposite wall.

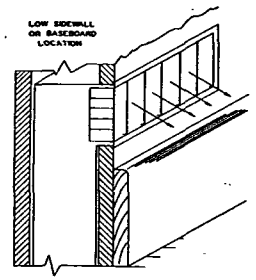
Velocities through registers may be reduced by the use of registers larger than the connecting ducts. Diffusers should be used to spread the air uniformly over the register face. Basic rules for the location and selection of registers, together with explanations of factors affecting operation, are given in Section 6 of Manual 4, Sections 3 and 4 of Manual 10, or Section B of Manual 7, published by the National Warm Air Heating and Air Conditioning Association.[4, 5, 6]

Registers should be well proportioned and decorated to harmonize with the trim. Air supply registers should be



[a] *Horizontal vanes, in back or front, to give downward deflections not to exceed from 15 to 22 deg.*

Fig. 2.... Recommended Type of High Side-wall Installation on Warm Wall[a]



[a] *Vertical bars with adjustable deflection, or fixed vertical bars with deflections to right and left not exceeding about 22 deg. For low side-wall location, the deflection for horizontal, multiple vane registers should not exceed 22 deg. For baseboard locations, the deflection for horizontal, multiple vane registers should not exceed about 10 deg.*

Fig. 3.... Recommended Type of Baseboard and Low Side-wall Installation on Warm Wall[a]

equipped with dampers, and all registers should be sealed against leakage around edges. The register types shown in Figs. 1, 2, and 3 have been recommended as standard by the National Warm Air Heating and Air Conditioning Association.

Return-air grilles may be located in hallways, near entrance doors, under windows, in exposed corners, or inside walls, depending on location of supply outlets. Baseboard returns are preferable to floor grilles. It is usually considered desirable to have them located in or near outside walls when supply outlets are located on inside walls. With this type of system, it is desirable to locate return grilles in as many rooms as possible. When used in perimeter systems, the return grilles may be centrally located. In basementless struc-

Table 1.... Recommended Thickness for Duct Materials

Round Ducts Diameter, In.	*Minimum Thickness*		*Minimum Weight of Tin-Plate*
	Galv. Iron, U.S. Gage	*Aluminum B&S Gage*	
Less than 12	30	26	IC (107 lb)
12 or more	28	26	IX (135 lb)
Rectangular Ducts Width, In.	*Minimum Thickness*		*Minimum Weight of Tin-Plate*
	Galv. Iron, U.S. Gage	*Aluminum B&S Gage*	
Ducts Enclosed in Partitions			
14 or less	30	26	IC (107 lb)
Over 14	28	26	IX (135 lb)
Ducts Not Enclosed in Partitions			
Less than 14	28	26	—
14 or more	26	24	—

Note: The table above is in accordance with the requirements of the National Board of Fire Underwriters to provide adequate fire protection. Industry practice is to use heavier gage metals where maximum duct widths exceed 24 in.

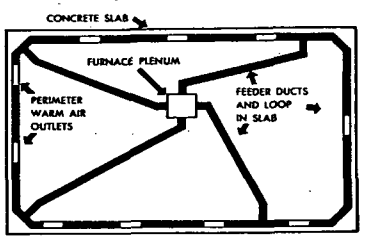


Fig. 4.... Perimeter-Loop System with Feeder and Loop Ducts in Concrete Slab

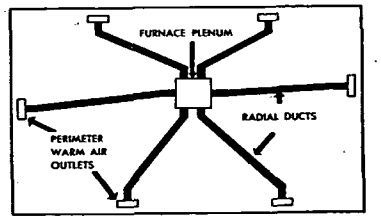


Fig. 6.... Perimeter-Radial System with Feeder Ducts in Concrete Slab or Crawl Space

tures, it is usually convenient to locate the return openings either high in the wall or in the ceiling. So far as the heating is concerned, however, this is merely a matter of convenience because low returns have been found to work equally well. It is usually not recommended in residential systems that any one return serve an area having a heat loss of more than 60,000 Btuh.

Dampers

Suitable dampers for air direction or volume control are essential to any duct system. Special care must be used in the design of any system, to minimize resistance. Sharp elbows, angles, and offsets should be avoided. Three types of dampers are commonly used. *Volume dampers* are used to completely cut off or reduce the flow through ducts. *Splitter dampers* are used where a branch is taken off a main trunk. *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. A damper with positive locking device should be provided for each main or duct branch. Labels placed on ducts should indicate the room being served. Damper positions should be marked to indicate proper summer and winter settings, and to reveal tampering.

DUCT CONSTRUCTION

The ducts may be either round or rectangular in cross section. The centerline radii of elbows should preferably be not less than one and one-half times the pipe diameter for round pipes. The throat radius of a turn in a rectangular duct should be at least one-half the duct dimension in the plane of the turn. Warm air ducts passing through cold spaces or located in exposed walls, should have 1 to 2 in. of insulation.

Ductwork must be permanent, rigid, non-buckling, and non-rattling. Joints in ductwork should be airtight. Galvanized-iron or aluminum sheet are usually used in the construction of ducts, but other materials may be used under the proper circumstances. All return ductwork within six feet of the winter air conditioner shall be of noncombustible material.

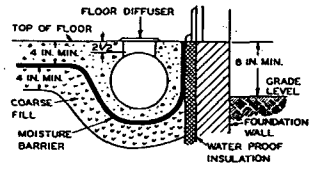
Requirements of the National Board of Fire Underwriters state that heating supply ducts shall be constructed entirely of noncombustible material equivalent in structural strength and durability to the specifications in Table 1.

Supply ducts must be securely supported by metal hangers, straps, lugs, or brackets. No nails should be driven through duct walls and no unnecessary holes should be cut in them.

Supply air stacks should not be installed in outside walls unless it is impracticable to do otherwise. If it is necessary to do this, the stack must be effectively insulated against exposure. Failure to do this will materially reduce the heating or cooling capacity of the branch duct and will probably result in unsatisfactory comfort conditions in the room.

No supply air duct should come in contact with masonry walls. All supply ducts which pass through a masonry wall should be covered with at least ½ in. of insulation where they pass through the wall.

All supply ducts or outlets should be equipped with an adjustable, locking type damper for air volume control. The damper should be installed in the branch duct whenever it



The minimum thickness of concrete over the loop duct should be 2½ in. and continuous mesh reinforcing not less than 6 x 6–10 gage and 18 in. wide should be placed in the concrete and centered over the duct.

Fig. 5.... Cross-Section of Slab Construction Containing Perimeter Duct[a]

Table 2.... Diameter of Feeders in Perimeter-Loop System

Btuh per Feeder	*Length of Feeder in Feet*	
	0–15 Feet	*16–30 Feet*
	Inches	*Inches*
Up to 7,999	6	6
8,000 to 8,999	6	7
9,000 to 10,999	7	7
11,000 to 11,999	7	8
12,000 to 12,999	7	8
13,000 to 17,000	8	8

is accessible. Splitter dampers for volume control are not recommended.

Return systems having more than one return intake may be equipped with balancing dampers. This is particularly desirable with systems in which the supply outlets are located on the inside wall or in the ceiling of the conditioned rooms.

Special attention should be given to the problem of noise elimination. The metal duct connection to and from the furnace casing and fan housing should be broken by strips of fire resistant fabric. Motors and mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with the fan housing. Installation of a fan directly under a return air grille is usually avoided.

SIMPLIFIED METHODS OF DESIGN

Simplified methods of design have been developed for several types of forced warm air heating systems. In general, the specific procedure chosen depends upon the type of system and the type of structure in which it is to be used. In most cases, more than one type of system may be used in a specific structure. It is therefore necessary to first choose the system desired and then determine the applicable design procedure.

Perimeter-Loop Systems in Concrete Slab Construction

The perimeter-loop system,[4] shown in Fig. 4, conveys the warm air through embedded ducts to perimeter diffusers. The air is delivered from the furnace to *feeder ducts* which extend radially to the *loop duct*. The air flows through the loop duct to the diffusers. The ducts serve the dual purpose of conveying the warm air and of warming the floor. Although the design of the concrete slab is not specifically the concern of the mechanical engineer, it is absolutely necessary that he understand the considerations involved lest improper construction be used with consequent adverse effect on the performance of the heating system. For information on slab construction, as well as duct placement, duct insulation, etc., see Manual 4 of the National Warm Air Heating and Air Conditioning Association.[4] A section of typically installed loop duct is shown in Fig. 5.

Simplified design procedures are contained in Manual 4. One of these procedures subject to the following limitations: total heat loss 100,000 Btuh, maximum perimeter 210 ft, and maximum length of uninsulated feeder duct 30 ft for area served by one furnace; is the following:

1. Calculate the heat loss for each room using the methods explained in Chapter 12. For subfloor and edge losses,[3] see Manual 3 of the National Warm Air Heating and Air Conditioning Association.
2. Determine the required furnace-bonnet capacity from the heat loss of the building.
3. Locate the diffusers on the plan. For best heating results, if a room has a heat loss of more than 8,000 Btuh or if it has two or more exposed walls, use two or more diffusers. Proportion the Btuh heat loss of each room among all diffusers in that room.
4. Locate the feeder ducts. First, divide the heat loss of the structure by 15,000 to determine the probable number of feeders. Second, locate the ducts so that they connect with the perimeter-loop at areas of greatest heat loss or in rooms where warm floors are particularly desirable. Third, locate the feeder ducts so that no more than 3 diffusers are in the section of loop duct between any two feeders. (The final number of feeder ducts may be larger than that originally estimated.)
5. Determine the size of feeder ducts using Table 2. Assume that the Btuh required by each diffuser will be supplied by the feeder nearest it. Determine the feeder duct diameter from its length and the total diffuser Btuh delivery carried by it.
6. Determine the total diffuser free area required in each room using Table 3. Measure the length of feeder duct under each room. Add to this *one-half* of the length of the loop duct, and determine the total diffuser free area required. Apportion this among all diffusers in the room.
7. The diameter of the loop duct should be uniform and equal to the diameter of the largest feeder duct.

Perimeter-Radial Systems in Concrete Slab Construction

The perimeter-radial system, Fig. 6, is similar to the perimeter-loop system in that the supply ducts are embedded in the concrete slab. The feeder ducts of the radial system extend directly from the subfloor plenum to the perimeter diffusers, and there is no loop duct around the perimeter of the floor. The feeders do provide floor-warming, but not to the same degree provided by the feeder and loop ducts of a perimeter-loop system. The same considerations of slab construction and edge insulation apply to both systems.

A simplified design procedure[4] using 6-in. diameter duct and subject to limitations of: 1,000 to 1,200 sq ft maximum area, not more than approximately 15 ft along perimeter between diffusers, and no radial duct substantially longer than 20 ft, is the following:

1. Calculate the heat loss of each room or area using the method explained in Chapter 12. For subfloor and edge losses, see[3] Manual 3 of the National Warm Air Heating and Air Conditioning Association.
2. Determine the required furnace-bonnet capacity from the heat loss of the building.
3. Locate the diffusers on the plan. Allow at least one diffuser for each exposed wall in each room or area of the building and preferably beneath window areas. For residential installations, limit each diffuser to a maximum of 7,000 Btuh. This is a preliminary estimate of the number of diffusers needed.
4. Measure the length and count the number of elbows in each feeder duct. Determine Btuh capacity of each feeder duct from Table 4. Use Table 4 (Section A) for side-wall and floor diffusers and Table 4 (Section B) for baseboard diffusers.
5. If the total Btuh capacity of all feeders to each room or area equals or exceeds the heat loss of that room or area, the number of feeders initially chosen is satisfactory. If the heat loss exceeds the Btuh capacity of the feeders, use and locate additional diffusers and feeder ducts.
6. The cfm values in Table 4 are used for designing the return-duct system.

Perimeter-Radial System in Crawl Space or Basement Construction

The perimeter concept of warm air heating has been applied to crawl-space and basement homes by locating the diffusers in the perimeter location and by warming the area underneath the floor of such structures. Although the construction of the crawl space and the foundation wall is the responsibility of the architect and the general contractor, it is important that the engineer check the construction to make certain that it will not adversely affect the performance of the heating system. Details on crawl-space construction are shown in Manual 4 of the National Warm Air Heating and Air Conditioning Association.[4] The crawl space or basement should be heated to provide warm floors.

A simplified design procedure[4] for an individual duct system is summarized as follows:

1. Calculate the heat loss of each room or area using the

Table 3 . . . . Minimum Diffuser Free Area Required for Perimeter-Loop Systems

Section A. Use This Table If Length of Feeder which Delivers Major Portion of Warm Air to Room is from 2 to 10 Ft.

Heat Loss of Room Btuh	Length of Feeder Duct under Room Plus ½ Length of Perimeter Duct Under Room, Ft.						
	0–9 Ft.	10–19 Ft.	20–29 Ft.	30–39 Ft.	40–49 Ft.	50–59 Ft.	60–69 Ft.
	Total Diffuser Free Area Required for Room, Square Inches						
0 to 3,999	20						
4,000 to 5,999	32	27	23	18			
6,000 to 7,999	43	39	34	29	25	20	
8,000 to 9,999	55	50	45	41	36	32	27
10,000 to 11,999	66	61	57	52	48	43	38
12,000 to 13,999	77	72	68	63	59	54	50
14,000 to 15,999	88	84	79	75	70	66	61
16,000 to 17,999	100	95	91	86	82	77	72
18,000 to 19,999	111	106	102	97	93	88	84
20,000 to 21,999	122	118	113	109	104	100	95
22,000 to 23,999	134	129	125	120	116	111	106
24,000 to 25,999	145	140	136	131	127	122	118
26,000 to 27,999	156	151	147	142	138	133	129
28,000 to 29,999	168	163	159	154	150	145	141
30,000 to 31,999	179	174	170	165	161	156	152
32,000 to 34,000	190	185	182	176	173	167	164

Section B. Use this Table If Length of Feeder which Delivers Major Portion of Warm Air to Room is from 10 to 20 Ft.

0 to 3,999	23	18	18				
4,000 to 5,999	36	31	26	21	16	16	
6,000 to 7,999	49	44	39	34	29	24	19
8,000 to 9,999	61	56	51	46	41	36	31
10,000 to 11,999	73	68	63	58	53	48	43
12,000 to 13,999	86	81	76	71	66	61	56
14,000 to 15,999	99	94	89	84	79	74	69
16,000 to 17,999	111	106	101	96	91	86	81
18,000 to 19,999	123	118	113	108	103	98	93
20,000 to 21,999	135	130	125	120	115	110	105
22,000 to 23,999	148	143	138	133	128	123	118
24,000 to 25,999	161	156	151	146	141	136	131
26,000 to 27,999	174	168	164	158	154	148	144
28,000 to 29,999	186	181	176	171	166	161	156
30,000 to 31,999	198	194	188	184	178	174	168
32,000 to 34,000	211	206	201	196	191	186	181

Section C. Use This Table if Length of Feeder which Delivers Major Portion of Warm Air to Room is from 20 to 30 Ft.

0 to 3,999	27	21	21				
4,000 to 5,999	42	36	30	24	18		
6,000 to 7,999	57	51	45	39	33	27	21
8,000 to 9,999	72	66	60	54	48	42	36
10,000 to 11,999	86	80	74	68	62	56	50
12,000 to 13,999	101	95	89	83	77	71	65
14,000 to 15,999	116	110	104	98	92	86	80
16,000 to 17,999	131	125	119	113	107	101	95
18,000 to 19,999	145	139	133	127	121	115	109
20,000 to 21,999	160	154	148	142	136	130	124
22,000 to 23,999	175	169	163	157	151	145	139
24,000 to 25,999	190	184	178	172	166	160	154
26,000 to 27,999	205	199	193	187	181	175	169
28,000 to 29,999	220	214	208	202	196	190	184
30,000 to 31,999	234	228	222	216	210	204	198
32,000 to 34,000	249	243	237	231	225	219	213

Table 4 .... Btuh Capacities and CFM of 6-Inch Feeder Ducts in Perimeter-Radial Systems in Concrete Slab Floors

Section A. For Side-Wall and Floor Diffusers

Bonnet Pressure 0.15 in. Static — *Bonnet Temperature 170 F*

No. of Elbows	Capacity	Actual Length of Pipe from Bonnet to Diffuser—Feet															
		5	10	15	20	25	30	35	40	45	50	55	60	65	70	75	80
0	Btuh										8700	8060	7460	6920	6430	5960	5550
	Cfm										127.7	124.8	122.3	119.7	117.4	115.0	113.0
1	Btuh									8620	8000	7430	6900	6400	5930	5500	5100
	Cfm									122.8	120.0	117.6	115.3	113.0	111.0	109.0	107.3
2	Btuh								8600	7960	7390	6860	6380	5940	5530	5150	4800
	Cfm								118.5	116.2	114.0	112.0	110.0	107.8	106.0	104.5	103.0
3	Btuh							8640	8020	7450	6950	6460	6020	5600	5200	4850	4500
	Cfm							114.0	112.0	110.0	108.2	106.5	104.7	103.0	101.4	99.6	98.0
4	Btuh						8760	8160	7580	7050	6540	6080	5640	5250	4880	4550	4250
	Cfm						110.2	108.4	106.6	105.0	103.0	101.5	100.2	99.0	97.5	96.0	94.5
5	Btuh					8850	8250	7670	7140	6640	6170	5740	5340	4960	4620	4290	3990
	Cfm					106.8	105.0	103.5	102.0	100.5	99.0	97.5	96.0	94.6	93.4	92.0	91.0
6	Btuh				9050	8420	7840	7280	6750	6300	5850	5440	5060	4700	4370	4060	3760
	Cfm				104.0	102.5	100.7	99.2	97.8	96.5	95.2	94.0	92.6	91.4	90.0	88.7	87.5

For Equivalent Length Values used and included in calculating this table are: Diffuser and Boot = 40 ft; Plenum Take-off = 20 ft; Each Elbow = 14 ft. For each additional 14 ft equivalent length of boot and diffuser, count one additional elbow.

All cfm values are based on standard air.

Section B. For Baseboard Diffusers Only

Bonnet Pressure 0.15 in. Static — *Bonnet Temperature 170 F*

No. of Elbows	Capacity	Actual Length of Pipe from Bonnet to Diffuser—Feet															
		5	10	15	20	25	30	35	40	45	50	55	60	65	70	75	80
0	Btuh						8570	8000	7450	6910	6420	5960	5560	5150	4800	4450	4160
	Cfm						109.0	107.2	105.5	103.7	102.0	100.4	98.8	97.5	96.1	94.8	93.5
1	Btuh					8700	8100	7550	7000	6520	6060	5630	5250	4860	4530	4220	3930
	Cfm					105.5	104.0	102.3	100.7	99.2	97.6	96.2	95.0	93.6	92.3	91.1	90.0
2	Btuh				8890	8270	7700	7160	6660	6200	5750	5350	4980	4620	4300	4000	3700
	Cfm				102.8	101.3	99.8	98.2	96.7	95.3	94.0	92.7	91.5	90.2	89.0	87.8	86.5
3	Btuh				8440	7870	7320	6820	6350	5880	5460	5070	4700	4350	4020	3710	3440
	Cfm				98.4	96.8	95.3	93.9	92.5	91.3	90.0	89.0	88.0	86.9	85.8	84.7	83.5
4	Btuh			8690	8080	7500	6980	6500	6030	5580	5180	4800	4450	4120	3820	3520	3250
	Cfm			95.8	94.4	93.0	91.6	90.3	89.0	87.8	86.7	85.7	84.6	83.7	82.6	81.6	80.5
5	Btuh		8940	8320	7740	7200	6680	6190	5720	5320	4920	4550	4220	3890	3600	3350	3100
	Cfm		93.4	92.2	91.0	89.8	88.5	87.3	86.2	84.8	83.8	82.8	81.7	80.8	79.9	79.0	78.0
6	Btuh		8580	7960	7380	6850	6350	5880	5440	5040	4660	4320	4000	3670	3400	3150	2930
	Cfm		90.0	89.8	87.6	86.5	85.5	84.2	83.1	82.0	80.9	80.0	79.0	78.0	77.3	76.5	76.0

The Equivalent Length Values used and included in calculating this table are: Diffuser and Boot = 100 ft; Plenum Take-off = 20 ft; Each Elbow = 14 ft. For each additional 14 ft equivalent length of boot and diffuser count one additional elbow.

All cfm values are based on standard air.

Table 5 .... Btuh Capacities and Cfm of Individual Ducts in Perimeter Systems Used in Crawl-Space or Basement Construction

Section A. For 5-Inch Individual Round Pipe and for Side-wall, Floor, and Baseboard Diffusers

Bonnet Pressure 0.15 in. Static — *Bonnet Temperature 170 F*

No. of Elbows		Actual Length of Pipe From Bonnet to Diffuser—Feet															
		5	10	15	20	25	30	35	40	45	50	55	60	65	70	75	80
0	Btuh				8700	7900	7150	6450	5800	5230	4700	4230	3830	3470	3150	2860	2600
	Cfm				100.5	96.2	92.7	89.7	87.0	84.6	82.5	80.4	78.4	76.5	74.8	73.2	71.7
1	Btuh			8620	7800	7100	6470	5830	5350	4850	4370	3960	3600	3260	2950	2650	2360
	Cfm			95.8	91.7	88.7	86.0	83.8	81.6	79.7	77.7	76.0	74.2	72.5	71.2	69.8	68.5
2	Btuh		8700	7900	7180	6530	5950	5400	4920	4430	4050	3680	3350	3040	2750	2480	2220
	Cfm		90.8	87.9	85.2	82.7	80.7	78.9	77.1	75.4	73.6	72.0	70.5	69.2	67.9	66.7	65.6
3	Btuh	8800	8000	7300	6650	6070	5530	5050	4600	4180	3800	3440	3120	2830	2550	2300	2060
	Cfm	86.6	84.0	81.5	79.3	77.5	75.8	74.3	72.8	71.3	70.0	68.7	67.5	66.3	65.2	64.1	63.0
4	Btuh	8250	7520	6850	6230	5680	5180	4720	4300	3900	3550	3220	2900	2630	2360	2120	1880
	Cfm	81.0	78.8	77.0	75.3	73.7	72.2	70.8	69.5	68.2	67.2	66.0	65.0	63.6	62.5	61.5	60.5
5	Btuh	7720	7050	6420	5840	5330	4850	4430	4030	3650	3320	3020	2730	2460	2200	1950	1730
	Cfm	76.4	74.5	73.0	71.6	70.3	69.0	67.8	66.6	65.5	64.4	63.2	62.0	61.2	60.2	59.2	58.1
6	Btuh	7280	6630	6070	5530	5050	4600	4180	3800	3470	3150	2850	2550	2280	2050	1820	1600
	Cfm	72.5	71.2	69.8	68.5	67.3	66.0	64.7	63.5	62.4	61.4	60.5	59.5	58.1	57.8	57.0	56.3

The Equivalent Length Values used and included in calculating this table are: Diffuser and Boot = 85 ft; Plenum Take-off = 15 ft; Each Elbow = 12 ft. For each additional 12 ft equivalent length of boot and diffuser count one additional elbow.

All cfm values are based on standard air.

Section B. For 6-Inch Individual Round Pipe and for Side-wall and Floor Diffusers

Bonnet Pressure 0.15 in. Static — *Bonnet Temperature 170 F*

No. of Elbows		Actual Length of Pipe from Bonnet to Diffuser—Feet															
		5	10	15	20	25	30	35	40	45	50	55	60	65	70	75	80
0	Btuh										8700	8060	7460	6920	6430	5960	5550
	Cfm										127.7	124.8	122.3	119.7	117.4	115.0	113.0
1	Btuh									8620	8000	7430	7900	6400	5930	5500	5100
	Cfm									122.8	120.0	117.6	115.3	113.0	111.0	109.0	107.3
2	Btuh								8600	7960	7390	6860	6380	5940	5530	5150	4800
	Cfm								118.5	116.2	114.0	112.0	110.0	107.8	106.0	104.5	103.0
3	Btuh							8640	8020	7450	6950	6460	6020	5600	5200	4850	4500
	Cfm							114.0	112.0	110.0	108.2	106.5	104.7	103.0	101.4	99.6	98.0
4	Btuh						8760	8160	7580	7050	6540	6080	5640	5250	4880	4550	4250
	Cfm						110.2	108.4	106.6	105.0	103.0	101.5	100.2	99.0	97.5	96.0	94.5
5	Btuh					8850	8250	7670	7140	6640	6170	5740	5340	4960	4620	4290	3990
	Cfm					106.8	105.0	103.5	102.0	100.5	99.0	97.5	96.0	94.6	93.4	92.0	91.0
6	Btuh				9050	8420	7840	7280	6750	6300	5850	5440	5060	4700	4370	4060	3760
	Cfm				104.0	102.5	100.7	99.2	97.8	96.5	95.2	94.0	92.6	91.4	90.0	88.7	87.5

The Equivalent Length Values used and included in calculating this table are: Diffuser and Boot = 40 ft; Plenum Take-off = 20 ft; Each Elbow = 14 ft. For each additional 14 ft equivalent length of boot and diffuser count one additional elbow.

All cfm values are based on standard air.

method explained in Chapter 12. Consider the basement or crawl space as a part of the structure and calculate its heat loss. Information in Manual 3' supplements that contained in Chapter 12, particularly with respect to crawl-space heat losses.

2. Determine the required furnace-bonnet capacity from the heat loss of the building.

3. Locate the diffusers on the plan. Allow at least one diffuser for each exposed wall in each room or area of building, and locate them preferably beneath window areas. For residential installations limit each diffuser to a maximum of 7,000 Btuh.

4. Determine the required Btuh delivery of each diffuser. Where more than one diffuser is used per room, proportion the heat loss equally among all diffusers.

5. Measure length and count number of elbows in each duct. From Table 5, determine the duct size (5 or 6-in. diameter)

Table 5.... Btuh Capacities and Cfm of Individual Ducts in Perimeter Systems Used in Crawl-Space or Basement Construction (Concluded)

Section C. For 6-Inch Individual Round Pipe and for Baseboard Diffusers Only

Bonnet Pressure 0.15 in. Static *Bonnet Temperature 170 F*

No. of Elbows		Actual Length of Pipe from Bonnet to Diffuser—Feet															
		5	10	15	20	25	30	35	40	45	50	55	60	65	70	75	80
0	Btuh						8570	8000	7450	6910	6420	5960	5560	5150	4800	4450	4160
	Cfm						109.0	107.2	105.5	103.7	102.0	100.4	98.8	97.5	96.1	94.8	93.5
1	Btuh					8700	8100	7550	7000	6520	6060	5630	5250	4860	4530	4220	3930
	Cfm					105.5	104.0	102.3	100.7	99.2	97.6	96.2	95.0	93.6	92.3	91.1	90.0
2	Btuh				8890	8270	7700	7160	6660	6200	5750	5350	4980	4620	4300	4000	3700
	Cfm				102.8	101.3	99.8	98.2	96.7	95.3	94.0	92.7	91.5	90.2	89.0	87.8	86.5
3	Btuh					8440	7870	7320	6820	6350	5880	5460	5070	4700	4350	4020	3710	3440
	Cfm					98.4	96.8	95.3	93.9	92.5	91.3	90.0	89.0	88.0	86.9	85.8	84.7	83.5
4	Btuh			8690	8080	7500	6980	6500	6030	5580	5180	4800	4450	4120	3820	3520	3250
	Cfm			95.8	94.4	93.0	91.6	90.3	89.0	87.8	86.7	85.7	84.6	83.7	82.6	81.6	80.5
5	Btuh		8940	8320	7740	7200	6680	6190	5720	5320	4920	4550	4220	3890	3600	3350	3100
	Cfm		93.4	92.2	91.0	89.8	88.5	87.3	86.2	84.8	83.8	82.8	81.7	80.8	79.9	79.0	78.0
6	Btuh		8580	7960	7380	6850	6350	5880	5440	5040	4660	4320	4000	3670	3400	3150	2930
	Cfm		90.0	89.8	87.6	86.5	85.5	84.2	83.1	82.0	80.9	80.0	79.0	78.0	77.3	76.5	76.0

The Equivalent Length Values used and included in calculating this table are: Diffuser and Boot = 100 ft; Plenum Take-off = 20 ft; Each Elbow = 14 ft. For each additional 14 ft equivalent length of boot and diffuser count one additional elbow.
All cfm values are based on standard air.

needed. Section A is for 5-in. duct, Section B is for 6-in. duct using side-wall or floor diffusers, and Section C is for 6-in. duct using baseboard diffusers.

Perimeter Extended-Plenum Systems in Crawl-Space or Basement Construction

The considerations involved in the design of extended-plenum systems are essentially the same as those of the individual pipe system. A simplified design procedure is summarized as follows:

1. Calculate the heat loss of each room or area using the method explained in Chapter 12. Consider the basement or crawl space as a part of the structure and calculate its heat loss. Information in Manual 3[3] supplements that contained in Chapter 12, particularly with respect to crawl-space heat losses.
2. Determine the required furnace-bonnet capacity from the heat loss of the building.
3. Locate the diffusers on the plan. Allow at least one diffuser for each exposed wall in each room or area of building, and preferably beneath window areas. For residential installations limit each diffuser to a maximum of 7,000 Btuh.
4. Determine the required Btuh delivery of each diffuser. Where more than one diffuser is used per room, proportion the heat loss equally among all diffusers.
5. Measure the length of each duct and count number of elbows from the extended plenum to the diffuser. Note whether top or side plenum take-offs are to be used. From Table 6, determine the duct size needed. (Note: Use that section of Table 6, Sections A to F, which applies *both* to the type of plenum take-off and diffuser to be used.)
6. The cfm values in Table 6 are used with Table 7 for sizing the extended-plenum system. No one extended plenum should exceed 35 ft in length.

Design of Return Duct System for Perimeter Installations

With perimeter systems designed in accordance with the foregoing simplified procedures, there is a circulating head of only 0.05 in. water available on the return side of the system. It is therefore necessary that the return ducts be adequately sized and be as short and straight as practicable. The return grilles may be centrally located, and the ducts may be located in the attic or other unheated spaces provided they are adequately insulated. Return air may be taken from basements and crawl spaces, but should not be taken from confined spaces in which the furnace is located.

The following simplified procedure[4] may be used:

1. Lay out the return-air duct system. Measure the length and count the number of 90 deg turns in each return duct.
2. Total the heat loss of the area served by each return duct.
3. Determine the required duct free area from the length of duct, the number of 90 deg turns, and the heat loss using Table 8. Also determine the free area of the return grille from Table 8.

Other Perimeter Systems

Each of the foregoing procedures can be used with approved furnaces that are capable of delivering their rated cfm air volume against a total pressure of 0.20 in. water external to the furnace. This available pressure is usually distributed on the basis of three-fourths (0.15 in. water) on the supply side and one-fourth (0.05 in. water) on the return side. Other systems, such as perimeter crawl-space plenum systems, may be designed by procedures outlined in Manual 4. In addition, procedures for designing 4-in. diameter duct systems are contained[5] in Manual 10 of the National Warm Air Heating and Air Conditioning Association.

Other systems of a proprietary nature but utilizing the perimeter concept of diffuser placement are in common use. These systems should be designed and installed in strict accordance with the manufacturer's recommendations. Such systems usually utilize smaller ducts (4 in. or less) and higher pressures (above 0.20 in. water). They also may require special streamlined, low resistance fittings and special diffusers. Some systems also employ higher bonnet temperatures. When either higher bonnet pressures or temperatures, or both, are to be used, care must be taken to ascertain that the furnace employed is capable of meeting the required conditions.

Table 6.... Heating Capacities and Cfm of Branch Ducts in Extended-Plenum Perimeter Systems

Section A. Side Take-off to 5-Inch Round Pipe for Side-wall, Floor, and Baseboard Diffusers

Extended-Plenum Pressure 0.15 in. Static at 160 F *Furnace-Plenum Pressure 0.15 in. Static at 170 F*

No. of Elbows		Actual Length from Extended-Plenum to Diffuser—Feet											
		5	10	15	20	25	30	35	40	45	50	55	60
0	Btuh	8960	8100	7310	6550	5900	5310	4800	4330	3900	3510	3170	2850
	Cfm	97.2	93.1	89.4	86.0	83.0	81.2	77.7	75.5	73.1	71.1	69.3	67.8
1	Btuh	8070	7290	6580	5950	5400	4880	4400	3980	3550	3200	2900	2620
	Cfm	88.4	85.2	82.2	79.3	76.9	74.6	72.5	70.7	68.9	67.1	65.7	64.2
2	Btuh	7340	6650	6020	5460	4950	4500	4070	3680	3300	2980	2680	2410
	Cfm	81.1	78.6	76.0	73.9	71.9	70.0	68.2	66.6	64.1	63.7	62.3	61.0
3	Btuh	6810	6180	5600	5070	4610	4200	3800	3420	3080	2770	2480	2230
	Cfm	75.0	73.0	71.0	69.2	67.3	65.8	64.4	63.1	61.7	60.2	59.0	58.2
4	Btuh	6350	5750	5210	4740	4300	3900	3530	3180	2870	2570	2300	2070
	Cfm	70.4	68.8	67.0	65.4	64.0	62.5	61.1	60.0	58.8	57.7	56.7	55.9
5	Btuh	5970	5400	4900	4440	4040	3670	3330	2990	2770	2410	2160	1930
	Cfm	65.2	64.8	63.2	62.0	60.7	59.4	58.3	57.4	56.2	55.3	54.4	53.7
6	Btuh	5660	5120	4630	4190	3800	3430	3100	2800	2520	2270	2040	1800
	Cfm	63.0	61.5	60.2	59.1	58.0	56.9	55.9	55.1	54.0	53.0	52.2	51.8

Note: Where any of the first three runs from the extended-plenum exceeds 20 ft and 3 elbows, count 3 extra elbows for that run.
The Equivalent Length Values used and included in calculating this table are: Diffuser and Boot = 35 ft; Each Elbow = 12 ft; Side Take-off = 20 ft.
Tapered Extended-Plenum Starting Collar.

Section B. Top Take-off to 5-Inch Round Pipe for Side-wall, Floor, and Baseboard Diffusers

Extended-Plenum Pressure 0.15 in. Static at 160 F *Furnace-Plenum Pressure 0.15 in. Static at 170F*

No. of Elbows		Actual Length from Extended-Plenum to Diffuser—Feet											
		5	10	15	20	25	30	35	40	45	50	55	60
0	Btuh	7070	6400	5800	5260	4800	4350	3940	3550	3190	2870	2580	2310
	Cfm	77.8	75.6	73.4	71.4	69.5	68.0	66.4	64.8	63.4	62.0	60.6	59.5
1	Btuh	6570	5970	5400	4890	4430	4010	3640	3290	2950	2650	2380	2140
	Cfm	72.6	70.8	68.9	67.2	65.6	64.2	62.8	61.5	60.3	59.0	58.0	57.0
2	Btuh	6180	5610	5080	4590	4160	3770	3410	3080	2770	2480	2210	2000
	Cfm	68.4	66.8	65.2	63.7	62.3	61.1	60.0	58.7	57.6	56.4	55.3	54.4
3	Btuh	5800	5250	4750	4310	3900	3540	3200	2890	2600	2340	2100	1870
	Cfm	64.4	63.1	61.8	60.5	59.3	58.3	57.1	56.3	55.5	54.7	53.9	53.2
4	Btuh	5490	4960	4500	4080	3700	3340	3020	2720	2440	2200	1970	1750
	Cfm	61.2	60.0	58.8	57.8	56.7	55.8	54.9	54.1	53.3	52.6	51.9	51.1
5	Btuh	5200	4710	4280	3870	3510	3180	2880	2580	2310	2060	1830	1640
	Cfm	58.5	57.5	56.5	55.5	54.5	53.7	52.9	52.1	51.4	50.7	50.0	49.5
6	Btuh	5000	4520	5100	3700	3350	3010	2700	2440	2160	1910	1700	1551
	Cfm	56.0	55.0	54.0	53.4	52.4	51.7	51.0	50.5	59.7	49.1	48.6	48.1

Note: Where any of the first three runs from the extended-plenum exceeds 20 ft and 3 elbows, count 3 extra elbows for that run.
The Equivalent Length Values used and included in calculating this table are: Diffuser and Boot = 35 ft; Each Elbow = 12 ft; Top Take-off = 50 ft.
Tapered Extended-Plenum Starting Collar.

INSIDE WALL DELIVERY SYSTEMS

The concept of warm-air perimeter heating is relatively new. The older methods of introducing warm air into spaces use other outlet locations. One of the most common of these has been the use of register locations on a warm or inside

Table 6.... Heating Capacities and Cfm of Branch Ducts in Extended-Plenum Perimeter Systems (*Continued*)

Section C. Side Take-Off to 6-Inch Round Pipe for Baseboard Diffusers Only

Extended-Plenum Pressure 0.12 in. Static at 160 F — *Furnace-Plenum Pressure 0.15 in. Static at 170 F*

No. of Elbows		Actual Length from Extended-Plenum to Diffuser—Feet											
		5	10	15	20	25	30	35	40	45	50	55	60
0	Btuh	9800	9100	8430	7820	7230	6700	6200	5730	5320	4920	4550	4210
	Cfm	106.6	104.4	102.3	100.3	98.5	96.8	95.0	93.1	91.7	90.0	88.5	87.2
1	Btuh	9230	8550	7930	7360	6800	6300	5830	5400	5000	4620	4280	3960
	Cfm	100.4	98.7	97.0	95.3	93.7	92.0	90.4	88.9	87.6	86.2	84.9	83.8
2	Btuh	8740	8100	7500	6960	6420	5960	5510	5110	4730	4390	4050	3750
	Cfm	95.5	93.8	92.2	90.6	89.4	88.0	86.5	85.2	84.0	82.8	81.6	80.6
3	Btuh	8300	7690	7130	6600	6110	5680	5240	4850	4500	4170	3840	3560
	Cfm	91.0	89.5	88.0	86.7	85.5	84.2	83.0	82.0	80.9	79.8	78.8	77.9
4	Btuh	7930	7350	6800	6290	5830	5400	5000	4620	4270	3940	3630	3370
	Cfm	86.8	85.5	84.2	83.3	82.0	81.0	80.0	79.1	78.0	77.0	76.1	75.3
5	Btuh	7600	7020	6500	6020	5580	5160	4770	4410	4080	3770	3470	3190
	Cfm	83.6	82.5	81.3	80.2	79.1	78.1	77.1	76.4	75.3	74.4	73.5	72.8
6	Btuh	7300	6760	6260	5780	5350	4950	4560	4200	3900	3580	3300	3040
	Cfm	80.4	79.5	78.4	77.5	76.3	75.3	74.4	73.5	72.8	72.0	71.2	70.6

Note: Where any of the first three runs from the extended-plenum exceeds 20 ft and 3 elbows, count 3 extra elbows for that run.
The Equivalent Length Values used and included in calculating this table are: Diffuser and Boot = 100 ft; Each Elbow = 14 ft; Side Take-off = 20 ft.
Tapered Extended-Plenum Starting Collar.

Section D. Top Take-Off to 6-Inch Round Pipe for Baseboard Diffusers Only

Extended-Plenum Pressure 0.12 in. Static at 160 F — *Furnace-Plenum Pressure 0.15 in. Static at 170 F*

No. of Elbows		Actual Length from Extended-Plenum to Diffuser—Feet											
		5	10	15	20	25	30	35	40	45	50	55	60
0	Btuh	8700	8050	7450	6900	6400	5910	5470	5160	4670	4300	4000	3720
	Cfm	95.0	93.2	91.7	90.0	88.6	87.3	86.0	84.7	83.5	82.4	81.3	80.2
1	Btuh	8260	7640	7090	6560	6080	5630	5200	4820	4470	4140	3830	3530
	Cfm	90.2	88.8	87.5	86.2	85.0	83.7	82.6	81.7	80.4	79.4	78.4	77.5
2	Btuh	7880	7290	6750	6250	5800	5380	4980	4590	4260	3940	3620	3340
	Cfm	86.3	85.0	84.0	82.9	81.7	80.5	79.6	78.7	77.6	76.7	75.8	74.9
3	Btuh	7560	7000	6470	5980	5540	5130	4740	4380	4060	3730	3430	3170
	Cfm	83.0	82.0	81.0	79.8	79.0	78.0	76.9	76.0	75.1	74.2	73.3	72.5
4	Btuh	7260	6700	6200	5740	5290	4900	4510	4160	3850	3560	3290	3020
	Cfm	80.0	79.1	78.1	77.2	76.2	75.2	74.3	73.5	72.7	71.9	71.0	70.2
5	Btuh	7000	6470	5970	5490	5070	4700	4320	3990	3600	3400	3120	2870
	Cfm	77.3	76.3	75.4	74.5	73.6	72.8	71.9	71.2	70.2	69.3	68.7	68.1
6	Btuh	6770	6230	5740	5290	4890	4500	4140	3820	3520	3240	2980	2760
	Cfm	74.6	73.8	73.0	72.2	71.3	70.5	69.8	69.0	68.3	67.5	66.9	66.1

Note: Where any of the first three runs from the extended-plenum exceeds 20 ft and 3 elbows, count 3 extra elbows for that run.
The Equivalent Length Values used and included in calculating this table are: Diffuser and Boot = 100 ft; Each Elbow = 14 ft; Top Take-off = 50 ft.
Tapered Extended-Plenum Starting Collar.

wall. For such systems, the combinations of parts selected as standard by the National Warm Air Heating and Air Conditioning Association are shown in Tables 9 and 10.

A simplified method for selecting the combinations of branches, boots, stacks, and registers, is given in Manual 7 of the National Warm Air Heating and Air Conditioning Association.* In this method, the sizes of the branch ducts are obtained from two tables giving their Btu capacities. The proper combination of parts for each branch can be determined if the following information is available.

1. Location of room, that is, whether on first or second story.
2. Actual horizontal length of duct from bonnet to boot, in feet.
3. Btu loss from room to be heated.

Table 6.... Heating Capacities and Cfm of Branch Ducts in Extended-Plenum Perimeter Systems (*Concluded*)

Section E. Side Take-Off to 6-Inch Round Pipe for Side-wall and Floor Diffusers

Extended-Plenum Pressure 0.12 in. Static at 160 F — *Furnace-Plenum Pressure 0.15 in. Static at 170 F*

No. of Elbows		Actual Length from Extended-Plenum to Diffuser—Feet											
		5	10	15	20	25	30	35	40	45	50	55	60
0	Btuh							8600	7930	7310	6740	6230	5800
	Cfm							123.2	120.0	116.8	114.0	111.4	109.1
1	Btuh						8560	7900	7290	6720	6200	5740	5340
	Cfm						117.4	114.5	112.1	109.2	107.0	104.8	102.8
2	Btuh					8510	7890	7300	6740	6230	5750	5310	4960
	Cfm					112.7	110.0	107.6	105.3	103.1	101.0	99.0	97.4
3	Btuh				8540	7900	7300	6780	6270	5800	5360	4960	4610
	Cfm				108.2	105.8	103.6	101.5	99.6	97.8	95.9	94.0	92.5
4	Btuh			8580	7970	7370	6820	6330	5850	5410	5000	4620	4300
	Cfm			104.0	101.9	99.8	97.8	96.0	94.5	92.8	91.1	89.7	88.3
5	Btuh		8720	8080	7490	6940	6440	5960	5490	5080	4690	4320	4020
	Cfm		100.2	98.3	96.7	94.8	93.0	91.4	90.0	88.7	87.2	85.0	84.7
6	Btuh	8900	8240	7630	7060	6550	6070	5600	5190	4800	4430	4100	3810
	Cfm	97.0	95.1	93.4	91.9	90.2	88.8	87.3	86.2	85.0	83.8	82.6	81.5

Note: Where any of the first three runs from the extended-plenum exceeds 20 ft and 3 elbows, count 3 extra elbows for that run.
The Equivalent Length Values used and included in calculating this table are: Diffuser and Boot = 40 ft; Each Elbow = 14 ft; Side Take-off = 20 ft.
Tapered Extended-Plenum Starting Collar.

Section F. Top Take-Off to 6-Inch Round Pipe for Side-wall and Floor Diffusers

Extended-Plenum Pressure 0.12 in. Static at 160 F — *Furnace-Plenum Pressure 0.15 in. Static at 170 F*

No. of Elbows		Actual Length from Extended-Plenum to Diffuser—Feet											
		5	10	15	20	25	30	35	40	45	50	55	60
0	Btuh					8430	7800	7210	6660	6170	5700	5270	4910
	Cfm					111.2	108.8	106.3	104.4	102.2	100.2	98.4	96.7
1	Btuh				8450	7840	7250	6700	6210	5730	5300	4900	4560
	Cfm				107.2	105.0	102.8	100.7	98.9	97.0	95.1	93.4	91.9
2	Btuh			8530	7900	7320	6800	6280	5790	5350	4960	4590	4250
	Cfm			103.2	101.1	99.0	97.2	95.4	93.8	92.1	90.6	89.0	87.8
3	Btuh		8660	8020	7430	6900	6370	5900	5450	5030	4650	4300	3990
	Cfm		99.7	97.8	96.0	94.2	92.4	90.8	89.4	88.0	86.7	85.3	84.2
4	Btuh	8810	8180	7580	7010	6500	6000	5550	5150	4750	4400	4070	3780
	Cfm	96.0	94.4	92.8	91.2	89.7	88.0	86.8	85.7	84.3	83.1	82.0	81.1
5	Btuh	8390	7770	7200	6650	6160	5700	5280	4890	4500	4170	3840	3580
	Cfm	91.4	90.0	88.5	87.2	85.8	84.5	83.2	82.4	81.0	80.0	79.0	78.3
6	Btuh	7980	7380	6840	6340	5880	5430	5020	4650	4290	3970	3660	3400
	Cfm	87.5	86.2	85.0	83.8	82.5	81.4	80.2	79.4	78.3	77.3	76.3	75.6

Note: Where any of the first three runs from the extended-plenum exceeds 20 ft and 3 elbows count 3 extra elbows for that run.
The Equivalent Length Values used and included in calculating this table are: Diffuser and Boot = 40 ft; Each Elbow = 14 ft; Top Take-off = 50 ft.
Tapered Extended-Plenum Starting Collar.

4. Equivalent lengths in feet of all fittings and of the register. Fig. 7 shows the values of equivalent lengths of fittings commonly used for domestic systems.

This simplified method is applicable to structures having heat losses not in excess of approximately 120,000 Btu per hour. The capacities shown in Tables 9 and 10 are based upon the most reliable data pertaining to friction losses and temperature drops in ducts. They are also based upon a 100 deg temperature rise of the air, and a static pressure available for overcoming friction losses in the external duct system alone of 0.20 in. of water. The use of this method requires that the fan in the fan-furnace assembly will be capable not only of overcoming the resistance of the external duct system alone, but also the resistances imposed by the blower inlet, the filter, and the furnace casing.

Tables 9 and 10 are applicable for the selection of both warm air and return air branches. A depth of 8 in. has been adopted as the standard for trunk ducts, and where a branch

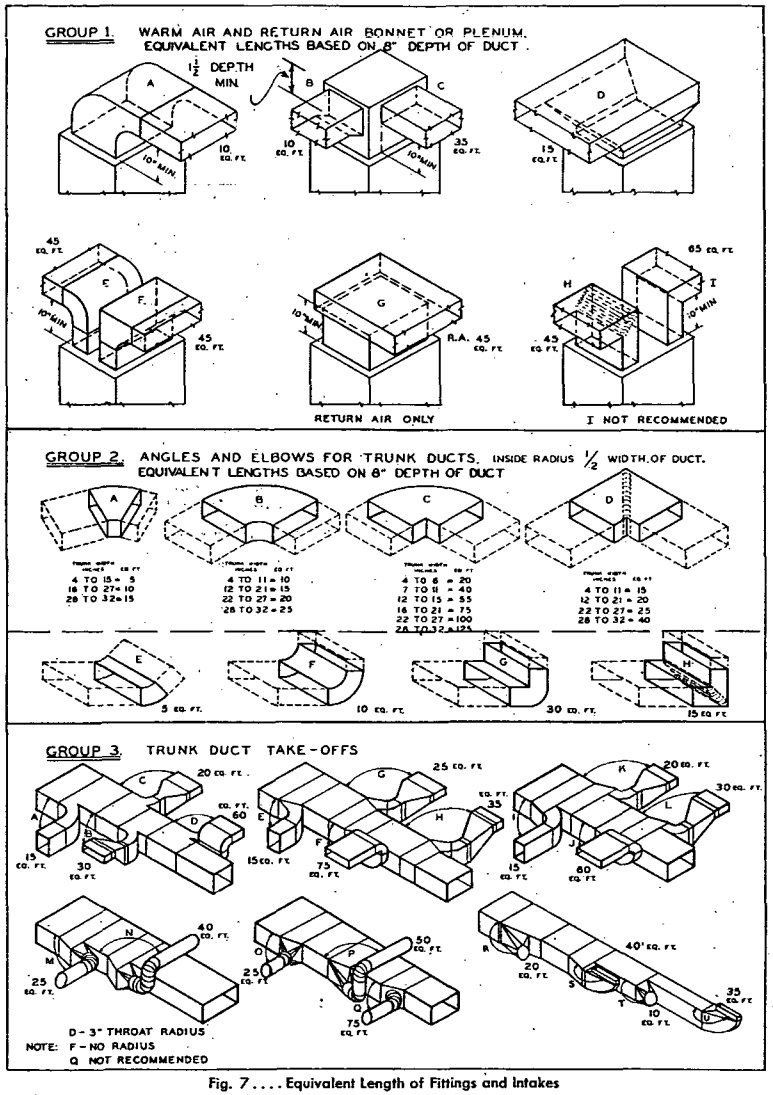
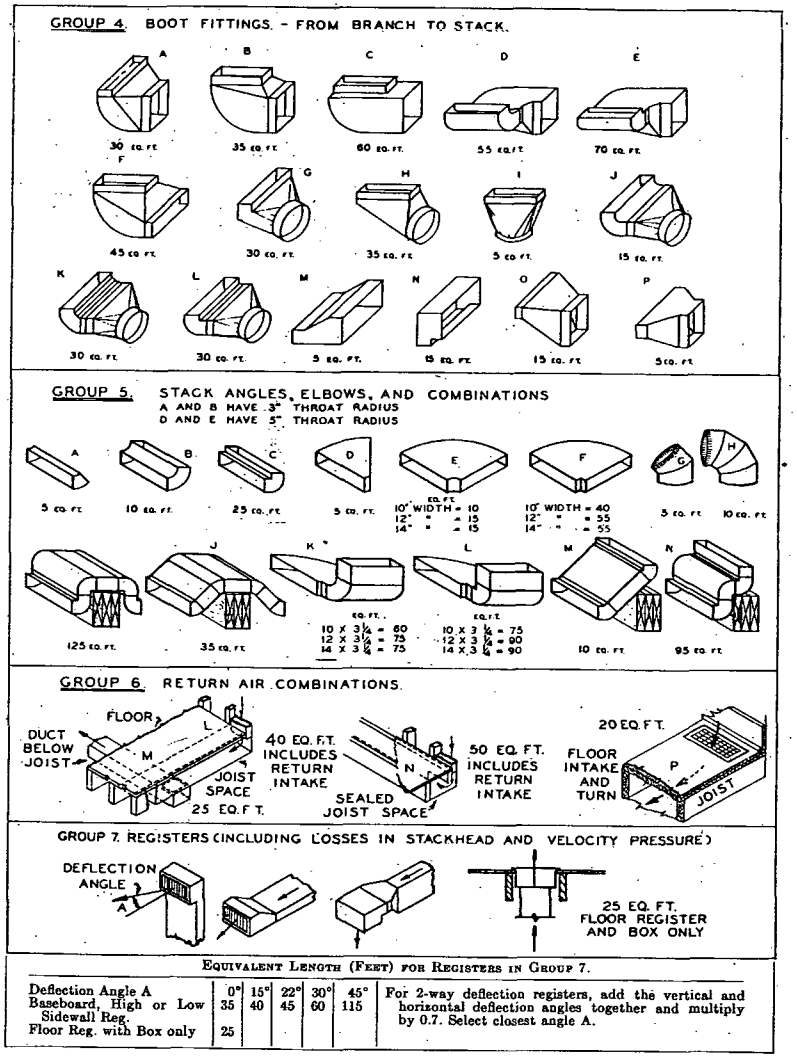


Fig. 7.... Equivalent Length of Fittings and Intakes



EQUIVALENT LENGTH (FEET) FOR REGISTERS IN GROUP 7.

Deflection Angle A	0°	15°	22°	30°	45°
Baseboard, High or Low Sidewall Reg.	35	40	45	60	115
Floor Reg. with Box only	25				

For 2-way deflection registers, add the vertical and horizontal deflection angles together and multiply by 0.7. Select closest angle A.

Fig. 7.... Equivalent Length of Fittings and Intakes (*Concluded*)

Table 7 .... Required Size of Extended-Plenum

Cfm Handled by Plenum	Plenum Dimensions Inches	Cfm Handled by Plenum	Plenum Dimensions Inches
Up to 300	8 x 8	701 to 800	20 x 8
301 to 400	10 x 8	801 to 900	22 x 8
401 to 500	12 x 8	901 to 1000	24 x 8
501 to 600	14 x 8	1001 to 1100	26 x 8
601 to 700	16 x 8	1101 to 1200	28 x 8

joins the trunk the required increase shown in the last line of Table 9 or Table 10 refers to an 8-in. trunk duct.

Where two branches form a trunk the longer branch is considered to be the trunk and its equivalent width (for 8-in. depth) is increased as required.

DESIGN PROCEDURE FOR LARGE SYSTEMS[a]

For buildings having a heat loss in excess of 120,000 Btu per hour, the design procedure given in Manual 9 of the National Warm Air Heating and Air Conditioning Association, may be used. Work sheets 9a, 9b, and 9c are available to simplify calculations.[a] The procedure consists of:

1. Calculation of design heat losses from individual spaces in the structure. The calculation of these losses is explained in Chapter 12.
2. Location of registers and return intakes on floor plan, showing types of registers, with distance from register to opposite wall and deflection of registers desired.
3. Laying out a proposed duct system for both warm air and return air sides of the system, and including details of types of fittings and the actual and equivalent lengths of each branch line from bonnet to register, without sizes. (See Fig. 7, Groups 1 through 6, for equivalent length of fittings.)
4. Determination of bonnet temperature.

If the rating sheet for a furnace-blower unit specifies a *fixed value of bonnet temperature*, find this temperature in the left-hand column of Table 11. If not specified, use the following procedure: Use Table 11 *for buildings* having a heat loss between 120,000 and 350,000 Btu per hr, or Table 12 for buildings having a heat loss greater than 350,000 Btu per hr. Select *shortest actual length*, including vertical risers and read downward in nearest column in Tables 11 or 12 until lower heavy diagonal line is reached, but do not cross line. Run horizontally to first column of table and note *bonnet temperature*. Also select *longest actual length* including vertical risers, and read downward in nearest column in Tables 11 or 12 until upper heavy diagonal line is just crossed. Run horizontally to left to obtain value for *bonnet temperature* in first column. Select as the design bonnet temperature any value between these two limits.

5. Determination of air volume to be delivered through each register and the respective register air temperatures.

Using Tables 11 or 12 and the design bonnet temperature selected, find the values of cfm per 1000 Btu for each duct length, and the corresponding register temperature.

6. Selection of register sizes and pressure losses to produce necessary throw, for the air volumes handled.

Use Tables 13 or 14 to obtain required free area and pressure loss of register. Tables 15 and 16 are used in sizing the duct system.

7. Design of duct system.
   - A. *Warm air branches.*
     - a. Use Table 16 to select maximum bonnet pressure *usually* required for the trunk carrying the maximum volume of air (cfm). If the maximum bonnet pressure is not high enough to accommodate the pressure loss through the registers, use a higher bonnet pressure. If the register pressure is critically large, it may be necessary to reduce it by either using two registers in place of one, or using smaller deflection angles.
     - b. Obtain actual duct loss by subtracting total register pressure loss, as determined from Tables 13 or 14, from maximum bonnet pressure.
     - c. Obtain the pressure drop in each duct per 100 ft by use of Table 15.
     - d. Determine duct size by means of air friction chart (such as Fig. 2, or Fig. 3, Chapter 21), volume (cfm), and pressure drop per 100 ft of duct.
   - B. *Return air branches.* (*Continued on p. 253.*)

Table 8 .... Required Size of Return Air Grille and Duct

For Pressure Drop Allowance of 0.05 In. of Water

Actual Ft		Up to 10 Ft						11 to 20 Ft						21 to 30 Ft						31 to 40 Ft					
Grille Free Area	No. of 90 Deg Turns	1	2	3	4	5	6	1	2	3	4	5	6	1	2	3	4	5	6	1	2	3	4	5	6
	Btuh	Duct Free Area																							
16	5,000	14	17	20	23	26	28	15	19	22	25	27	28	17	20	23	25	27	29	18	31	24	27	28	30
32	10,000	27	32	37	42	46	49	29	34	39	43	47	49	31	36	41	45	48	51	33	38	43	47	49	51
47	15,000	37	44	50	55	60	64	40	46	52	57	61	65	43	49	54	58	63	67	45	51	56	61	64	67
63	20,000	46	54	61	66	73	78	49	57	63	69	75	79	53	60	65	71	77	82	55	62	68	74	78	83
79	25,000	54	64	72	79	86	92	58	68	75	82	88	94	62	71	78	84	91	96	66	74	80	87	92	98
95	30,000	62	73	82	89	97	104	67	76	85	93	100	106	71	80	88	95	102	108	75	84	91	99	104	110
111	35,000	69	82	92	101	111	119	74	85	95	106	114	121	79	91	99	108	117	124	84	94	103	113	119	126
126	40,000	77	90	100	110	121	130	82	94	105	116	124	132	87	99	109	118	128	137	92	103	113	123	130	140
158	50,000	89	105	119	132	146	158	95	111	124	138	150	161	102	117	129	142	155	166	108	123	135	148	158	170
190	60,000	100	122	139	154	170	181	111	129	145	161	174	184	119	137	151	165	178	189	126	143	158	172	181	193
221	70,000	117	139	158	174	191	203	127	147	164	182	195	206	135	155	171	186	200	213	144	162	178	193	203	216
252	80,000	128	155	175	191	208	222	140	164	182	198	214	225	151	172	188	203	219	231	160	179	195	211	222	235
284	90,000	141	168	188	205	225	244	153	177	195	213	232	248	164	186	201	218	240	256	173	193	209	229	244	262
316	100,000	154	180	200	221	247	265	166	189	209	233	252	270	176	198	217	239	258	277	185	206	227	250	265	285

Note: Where a multiple number of return air ducts join together, add the individual duct free areas together to determine the free area of duct to be used from that point to the return air plenum of furnace.

Table 9. . . . Capacity and Sizing Table—First Story

For Warm Air and Return Air Stack, Branch, and Registers

Equiv. Length of Fittings and Registers	Horizontal Length Furnace to Register: Ducts[a]	Horizontal Length Furnace to Register: Insulated Attic Ducts	First Story Room Heat Loss—Btuh						
Col. a	Col. b	Col. c	Col. d	Col. e	Col. f	Col. g	Col. h	Col. i	Col. j
Up to 70 Eq Ft	Up to 7	Up to 5	7,200	12,500	16,000	19,100	25,000	32,000	80,000
	8–12	6–9	6,700	11,700	15,000	18,000	23,400	30,000	75,000
	13–17	10–13	6,100	10,800	14,000	17,000	21,600	28,000	70,000
	18–25	14–18	5,600	9,900	13,000	16,000	19,800	26,000	65,000
	26–35	19–25	4,800	8,500	11,000	14,200	17,000	22,600	56,500
	36–45	26–35	4,100	7,400	9,900	12,500	14,800	19,800	49,500
	46–55	36–45	3,500	6,400	8,700	11,000	12,800	17,400	43,500
	.....	46–55	3,000	5,500	7,700	9,600	11,000	15,400	38,500
71 to 100 Eq Ft	Up to 7	Up to 5	5,500	9,900	13,100	16,300	19,800	26,200	65,500
	8–12	6–9	5,100	9,200	12,300	15,400	18,400	24,600	61,500
	13–17	10–13	4,800	8,600	11,600	14,500	17,200	23,200	58,000
	18–25	14–18	4,500	8,100	10,900	13,700	16,200	21,800	54,500
	26–35	19–25	3,900	7,100	9,700	12,200	14,200	19,400	48,500
	36–45	26–35	3,400	6,200	8,500	10,800	12,400	17,000	42,500
	46–55	36–45	3,000	5,400	7,500	9,500	10,800	15,000	37,500
	.....	46–55	2,600	4,600	6,500	8,300	9,200	13,000	33,500
101 to 130 Eq Ft	Up to 7	Up to 5	4,600	8,500	11,300	14,300	17,000	22,600	56,200
	8–12	6–9	4,300	7,900	10,600	13,500	15,800	21,200	52,600
	13–17	10–13	4,100	7,400	10,000	12,700	14,800	20,000	50,200
	18–25	14–18	3,800	6,900	9,400	11,900	13,800	18,800	47,300
	26–35	19–25	3,300	6,100	8,400	10,500	12,200	16,800	42,000
	36–45	26–35	3,000	5,300	7,400	9,300	10,600	14,800	37,000
	46–55	36–45	2,700	4,700	6,500	8,300	9,400	13,000	32,400
	.....	46–55	2,300	4,000	5,400	7,000	8,000	10,800	27,000
131 to 165 Eq Ft	Up to 7	Up to 5	4,100	7,300	9,800	12,300	14,600	19,600	49,200
	8–12	6–9	3,800	6,900	9,100	11,700	13,800	18,200	45,800
	13–17	10–13	3,600	6,400	8,600	11,000	12,800	17,200	41,900
	18–25	14–18	3,400	6,000	8,100	10,300	12,000	16,200	40,300
	26–35	19–25	2,900	5,300	7,200	9,100	10,600	14,400	36,000
	36–45	26–35	2,600	4,700	6,400	8,100	9,600	12,800	31,800
	46–55	36–45	2,300	4,200	5,600	7,100	8,400	11,200	28,100
	.....	46–55	2,000	3,600	4,700	6,000	7,200	9,400	23,500
166 to 200 Eq Ft	Up to 7	Up to 5	3,800	6,500	8,800	11,000	13,000	17,600	44,100
	8–12	6–9	3,500	6,100	8,200	10,500	12,200	16,400	51,500
	13–17	10–13	3,300	5,700	7,700	9,900	11,400	15,400	38,800
	18–25	14–18	3,100	5,400	7,200	9,300	10,800	14,400	36,000
	26–35	19–25	2,700	4,800	6,400	8,300	9,600	12,800	32,000
	36–45	26–35	2,400	4,300	5,700	7,300	8,600	11,400	28,200
	46–55	36–45	2,100	3,800	5,000	6,400	7,600	10,000	24,700
	.....	46–55	1,700	3,000	3,900	5,100	6,000	7,800	19,500
Combination Number			41	42	43	44	45	46	47
Stacks[b]: No. (in parenthesis) and Size			(1) 10x3¼	(1) 10x3¼	(1) 12x3¼	(1) 14x3¼	(2) 10x3¼	(2) 12x3¼	—
Rectangular Branch Size Round Branch Size (Diam)			3x8 5	4x8 6	5x8 7	6x8 8	8x8 9	10x8 10	15x8 12
Number of Joist Spaces and Minimum Depth-Return Air			1 at 3	1 at 3	1 at 4	1 at 5	1 at 6 or 2 at 3	1 at 7 or 2 at 4	1 at 9 or 2 at 5
Registers: Low Wall, High Wall, or Baseboard			10x6	10x6	12x6	14x6	(2)10x6 or (1) 24x6	(2)12x6 or (1) 30x6	—
Registers: Floor-Warm Air			8x10	8x10	9x12	9x12 or longer	10x12	12x14	—
Registers: Floor-Return Air			6x10 or 4x14	6x10 or 4x14	6x12 or 6x14	6x14	6x30	6x30	8x30
Trunk Duct Increase, Inches			1	2	3	4	5	7	12

[a] Uninsulated ducts in heated spaces and insulated ducts in unheated spaces.

[b] Note: For return air a 14x2¾ in. stud space may be used instead of 10x3¼ in. stack. Where 12x3¼ in. or 14x3¼ in. stack is required, a 14x2¾ in. stud spaces may be used only when inside of stud space is smooth and without protruding plaster keys. The number of joist spaces for combination 47 is based on 2x8 in. joists. One space may be used with 2x10 in. joists.

The limiting factor for a duct combination is the stack capacity.

Table 10.... Capacity and Sizing Table—Second Story

For Warm Air and Return Air Stack, Branch, and Registers

Equiv. Length of Fittings and Registers	Horizontal Length Furnace to Register		Second Story Room Heat Loss—Btuh						
	Ducts[a]	Insulated Attic Ducts							
Col. a	Col. b	Col. c	Col. d	Col. e	Col. f	Col. g	Col. h	Col. i	Col. j
Up to 70 Eq Ft	Up to 7	Up to 5	6,300	10,900	14,000	17,000	21,800	28,000	70,000
	8–12	6–9	5,700	10,000	13,000	15,900	20,000	26,000	65,000
	13–17	10–13	5,200	9,200	12,100	14,900	18,400	24,200	60,500
	18–25	14–18	4,800	8,500	11,400	13,900	17,000	22,400	57,000
	26–35	19–25	4,100	7,300	10,000	12,400	14,600	20,000	50,000
	36–45	26–35	3,500	6,400	8,800	11,100	12,800	17,600	44,000
	46–55	36–45	3,100	5,600	7,800	9,900	11,200	15,600	39,000
	.....	46–55	2,800	5,000	6,800	9,000	10,000	13,600	34,000
71 to 100 Eq Ft	Up to 7	Up to 5	5,000	9,000	11,900	14,800	18,000	23,900	59,500
	8–12	6–9	4,600	8,200	11,000	13,900	16,400	22,000	55,000
	13–17	10–13	4,300	7,600	10,300	13,000	15,200	20,600	51,500
	18–25	14–18	4,000	7,100	9,600	12,200	14,200	19,200	48,000
	26–35	19–25	3,400	6,200	8,400	10,800	12,400	16,800	42,000
	36–45	26–35	3,000	5,400	7,500	9,600	10,800	15,000	37,500
	46–55	36–45	4,700	4,700	6,700	8,500	9,400	13,400	33,500
	.....	46–55	2,300	4,200	6,000	7,500	8,400	12,000	30,000
101 to 130 Eq Ft	Up to 7	Up to 5	4,200	7,700	10,400	13,000	15,400	21,700	52,200
	8–12	6–9	3,900	7,200	9,700	12,100	14,400	19,400	48,400
	13–17	10–13	3,700	6,700	9,000	11,300	13,400	18,000	45,300
	18–25	14–18	3,500	6,200	8,400	10,600	12,400	16,800	42,300
	26–35	19–25	3,000	5,400	7,400	9,400	10,800	14,800	37,000
	36–45	26–35	2,600	4,700	6,500	8,300	9,400	13,000	32,500
	46–55	36–45	2,400	4,100	5,800	7,400	8,200	11,600	29,000
	.....	46–55	2,000	3,600	4,800	6,500	7,200	9,600	24,000
131 to 165 Eq Ft	Up to 7	Up to 5	3,800	6,800	9,100	11,400	13,600	18,200	45,500
	8–12	6–9	3,500	6,300	8,400	10,500	12,600	16,800	42,000
	13–17	10–13	3,200	5,800	7,900	9,800	11,600	15,800	39,600
	18–25	14–18	3,000	5,500	7,400	9,200	11,000	14,800	36,800
	26–35	19–25	2,700	4,800	6,300	8,100	9,600	12,600	32,200
	36–45	26–35	2,300	4,200	5,700	7,200	8,400	11,400	28,200
	46–55	36–45	1,900	3,400	4,500	5,800	6,800	9,000	22,800
	.....	46–55	1,800	3,300	4,300	5,400	6,600	8,600	21,500
166 to 200 Eq Ft	Up to 7	Up to 5	3,500	6,100	8,200	10,300	12,200	16,400	41,000
	8–12	6–9	3,200	5,700	7,600	9,500	11,400	15,200	38,300
	13–17	10–13	2,900	5,300	7,100	8,900	10,600	14,200	35,800
	18–25	14–18	2,700	5,000	6,700	8,300	10,000	13,400	33,400
	26–35	19–25	2,500	4,400	5,700	7,400	8,800	11,400	29,000
	36–45	26–35	2,100	3,900	5,100	6,500	7,800	10,200	25,800
	46–55	36–45	1,900	3,400	4,500	5,800	6,800	9,000	22,800
	.....	46–55	1,500	2,800	3,500	4,700	5,600	7,000	17,500
Combination Number			41	42	43	44	45	46	47
Stacks[b]: No. (in paren.) and Size			(1) 10x3¼	(1) 10x3¼	(1) 12x3¼	(1) 14x3¼	(2) 10x3¼	(2) 12x3¼	—
Rectangular Branch Size Round Branch Size (Diam)			3x8 5	4x8 6	5x8 7	6x8 8	8x8 9	10x8 10	15x8 12
Number of Joist Spaces and Minimum Depth-Return Air			1 at 3	1 at 3	1 at 4	1 at 5	1 at 6 or 2 at 3	1 at 7 or 2 at 4	1 at 9 or 2 at 5
Registers: Low Wall, High Wall or Baseboard:			10x6	10x6	12x6	14x6	(2)10x6 or (1) 24x6	(2)12x6 or (1) 30x6	—
Registers: Floor, Warm Air			8x10	8x10	9x12	9x12 or longer	10x12	12x14	—
Registers: Floor, Return Air			6x10 or 4x14	6x10 or 4x14	6x12 or 6x14	6x14	6x30	6x30	8x30
Trunk Duct Increase, Inches			1	2	3	4	5	7	12

[a] Uninsulated ducts in heated spaces and insulated ducts in unheated spaces.

[b] Note: For return air a 14x3¼ in. stud space may be used instead of 10x3¼ in. stack. Where 12x3¼ in. or 14x3¼ in. stack is required, a 14x3¼ in. stud space may be used only when inside of stud space is smooth and without protruding plaster keys. The number of joist spaces for combination 47 is based on 2x8 joists. One space may be used with 2x10 joists.

The limiting factor for a duct combination is the stack capacity.

Table 11.... Selection of Bonnet Temperature, Register Temperature and Register Delivery (CFM)[a, b]

(For Buildings Having a Heat Loss between 120,000 and 350,000 BTU/Hr.)

Register Delivery Volume in CFM per 1000 BTU per Hr.

Temp. of Bonnet °F	Linear Distance from Bonnet to Register, in Feet																							
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	210	220	230	240
110	108[a]	107	106	104	103	102	101	100	99	98	97	96	95	94	93	92	91	90	90	89	89	88	88	87
22.1	23.1[b]	23.6	24.2	25.4	26.0	26.6	27.3	28.0	28.8	29.6	30.5	31.5	32.3	33.3	34.6	35.9	37.3	38.6	38.6	39.9	39.9	41.3	41.3	42.7
120	118	116	114	112	111	109	108	106	105	103	102	101	100	99	98	97	96	95	94	93	92	91	90	90
18.4	19.0	19.6	20.3	21.1	21.6	22.5	23.1	24.2	24.8	26.0	26.6	27.3	28.0	28.8	29.6	30.5	31.5	32.3	33.3	34.6	35.9	37.3	38.6	38.6
130	128	126	123	121	119	117	115	113	111	110	108	107	105	104	103	102	101	99	98	97	96	95	94	93
15.9	16.4	16.9	17.6	18.1	18.7	19.3	20.0	20.7	21.6	22.1	23.1	23.6	24.8	25.4	26.0	26.6	27.3	28.8	29.6	30.5	31.5	32.3	33.3	34.6
140	137	134	131	129	126	124	122	120	118	116	114	112	110	109	107	106	105	103	102	101	100	99	98	97
14.0	14.6	15.1	15.7	16.2	16.9	17.3	17.9	18.4	19.0	19.6	20.3	21.1	22.1	22.5	23.6	24.2	24.8	26.0	26.6	27.3	28.0	28.8	29.6	30.5
150	147	143	140	137	134	131	129	127	124	122	120	118	116	114	112	110	109	107	106	105	103	102	101	100
12.6	12.9	13.5	14.0	14.6	15.1	15.7	16.2	16.6	17.3	17.9	18.4	19.0	19.6	20.3	21.1	22.1	22.5	23.6	24.2	24.8	26.0	26.6	27.3	28.0
160	156	152	149	145	142	139	136	133	130	128	126	123	121	119	117	115	113	112	110	108	107	106	104	103
11.4	11.8	12.3	12.7	13.2	13.7	14.2	14.8	15.3	15.9	16.4	16.9	17.6	18.1	18.7	19.3	20.0	20.7	21.1	22.1	23.1	23.6	24.2	25.4	26.0
170	166	161	157	153	150	146	143	140	137	134	131	128	126	124	122	120	118	116	114	112	110	109	107	106
10.5	10.8	11.3	11.7	12.2	12.6	13.1	13.5	14.0	14.6	15.1	15.7	16.4	16.9	17.3	17.9	18.4	19.0	19.6	20.3	21.1	22.1	22.5	23.6	24.2
180	175	170	166	161	157	153	150	146	143	140	137	135	131	128	126	124	122	120	118	116	115	113	111	110
9.7	10.0	10.5	10.8	11.3	11.7	12.2	12.6	13.1	13.5	14.0	14.6	14.9	15.7	16.4	16.9	17.3	17.9	18.4	19.0	19.6	20.0	20.7	21.6	22.1
190	184	179	174	169	165	160	156	153	149	145	142	139	136	133	130	128	126	124	121	119	117	115	113	112
9.1	9.4	9.8	10.1	10.6	10.9	11.4	11.8	12.2	12.7	13.2	13.7	14.2	14.7	15.3	15.9	16.4	16.9	17.3	18.1	18.7	19.3	20.0	20.7	21.1
200	193	187	182	176	171	167	163	158	154	151	147	144	140	138	135	132	129	127	125	122	120	118	116	114
8.6	8.9	9.2	9.6	9.9	10.4	10.7	11.1	11.6	12.1	12.4	12.9	13.4	14.0	14.4	14.9	15.5	16.2	16.6	17.2	17.9	18.4	19.0	19.6	20.3

[a] Register temperature, Fahrenheit. (Use upper value in each group.)

[b] Register delivery volume in cubic feet per minute per 1000 Btu. (Use lower value in each group.)

[c] Register delivery volumes in column 1 are for zero length of duct.

B. *Return air branches.*
   a. Select a low value of actual duct loss obtained from step b under item A for the suction loss of return duct system.
   b. Proceed in sizing return air branches by the same method described for the warm air branches.

C. *Trunk ducts for warm air and return air sides of system.*
   a. Add air volumes of branches to be handled by each trunk duct.
   b. The friction loss per 100 ft of trunk duct is determined by taking the smaller of the two values for friction loss for the two ducts meeting at the junction.
   c. Determine trunk duct size by using an air friction chart, volume (cfm), and pressure drop per 100 ft of trunk duct.

8. Selection of Blower.
   A. Determine total cfm air delivery (the sum of all branch cfm values).
   B. Determine static pressure requirement.
      a. For furnace-blower combination units it is the sum of bonnet pressure and suction pressure.
      b. For blowers separately selected from the furnace it is the sum of bonnet pressure, suction pressure, filter loss, casing loss, losses through air washers, coils, and other devices.

9. Selection of Furnace.
   A. Determine register delivery (the sum of room Btu losses).
   B. Determine bonnet capacity.
      Bonnet capacity = (total cfm) × (temperature rise) × 1.089.
   C. Determine allowance for pick-up load. For buildings which are heated intermittently, such as churches and auditoriums, it is customary to add from 10 to 25 percent extra furnace capacity for warming of the structure.

AUTOMATIC CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in a properly designed system, can be largely eliminated through proper care in the planning and installation of the control system.[29] Best results are obtained when the fan is operated as continuously as possible, coupled with frequent, short cycles of burner operation. Controls usually employed are;

1. A *room thermostat* located in the occupied space at a point where there is good air circulation and where the temperature is representative of the area being controlled. It is the

Table 12.... Selection of Bonnet Temperature, Register Temperature and Register Delivery (CFM)[a,b]

(For Buildings Having a Heat Loss Greater than 350,000 BTU/Hr.)

Register Delivery Volume in CFM per 1000 BTU per Hr.

Bonnet Temp. °F	Linear Distance from Bonnet to Register, in Feet																							
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	210	220	230	240
140	138[a]	136	134	132	130	128	127	126	125	124	123	121	120	119	118	117	116	115	114	113	112	111	110	109
14.0[c]	14.4[b]	14.8	15.1	15.5	15.9	16.4	16.6	16.9	17.2	17.3	17.6	18.1	18.4	18.7	19.0	19.3	19.6	20.0	20.3	20.7	21.1	21.6	22.1	22.5
150	148	146	143	141	139	137	135	133	131	129	128	127	126	124	123	122	120	119	118	117	116	115	114	113
12.6	12.8	13.1	13.5	13.9	14.2	14.6	14.9	15.3	15.7	16.2	16.4	16.6	16.9	17.3	17.6	17.9	18.4	18.7	19.0	19.3	19.6	20.0	20.3	20.7
160	158	155	152	150	147	145	143	140	138	136	134	132	130	128	127	126	125	124	123	121	120	119	118	117
11.4	11.6	12.0	12.3	12.6	12.9	13.2	13.5	14.0	14.4	14.8	15.1	15.5	15.9	16.4	16.6	16.9	17.2	17.3	17.6	18.1	18.4	18.7	19.0	19.3
170	167	164	161	158	155	152	150	147	145	143	140	138	136	134	132	130	128	127	126	125	124	123	121	120
10.5	10.7	11.0	11.3	11.6	12.0	12.3	12.6	12.9	13.2	13.5	14.0	14.4	14.8	15.1	15.5	15.9	16.4	16.6	16.9	17.2	17.3	17.6	18.1	18.4
180	177	173	170	167	164	161	158	155	152	150	147	145	143	140	138	136	134	132	130	128	127	126	125	124
9.7	9.9	10.2	10.5	10.7	11.0	11.3	11.6	12.0	12.3	12.6	12.9	13.2	13.5	14.0	14.4	14.8	15.1	15.5	15.9	16.4	16.6	16.9	17.2	17.3
190	186	182	178	175	172	168	165	162	159	156	153	151	148	146	143	141	139	137	135	133	131	129	128	127
9.1	9.3	9.6	9.8	10.0	10.3	10.7	10.9	11.2	11.5	11.8	12.2	12.4	12.8	13.1	13.5	13.9	14.2	14.6	14.9	15.3	15.7	16.2	16.4	16.6
200	196	191	187	183	179	176	172	169	166	163	160	157	154	151	149	147	144	142	140	138	135	133	131	130
8.6	8.8	9.0	9.2	9.5	9.8	9.9	10.3	10.6	10.8	11.1	11.4	11.7	12.1	12.4	12.7	12.9	13.4	13.7	14.0	14.4	14.9	15.3	15.7	15.9

[a] Register temperature, Fahrenheit. (Use upper value in each group.)
[b] Register delivery volume in cubic feet per minute per 1000 Btu. (Use lower value in each group.)
[c] Register delivery volumes in column 1 are for zero length of duct.

operating control which initiates the starting and stopping of the burner. It is frequently of the timed, two-position type. Refer to Chapter 43 for more information on types of room thermostats.

2. A *fan thermostat* located in the bonnet of the furnace to start the fan at a bonnet temperature beween 110 and 130 F and stop the fan at about 15 deg below the cut-in point. The lower settings are used for high side-wall register installations and the higher settings for baseboard register installations. For most satisfactory performance these settings should be as low as is feasible without resulting in drafts.

3. A *high limit control*, also in the bonnet of the furnace, to stop the burner independently of the room thermostat if the air temperature exceeds 175 F. The fan thermostat and high limit control are sometimes combined into a single unit.

4. A *humidistat* to regulate the moisture supplied to the rooms on systems which are equipped with humidifying means. This may be of the room type, usually located adjacent to the room thermostat, or it may be of the insertion type located in the main return air duct.

5. Primary and limit controls for the various types of gas burners, oil burners and coal burners (stokers) should be provided as required, and are discussed in Chapter 34.

ADJUSTMENT OF SYSTEM

More even room temperatures will result if the controls are adjusted to produce long period fan operation. This is accomplished in automatically-fired forced warm air heating systems when the control arrangement is of the type where the room thermostat controls the fire, and the blower control (fan switch) controls the blower operation. This procedure as outlined in detail in Manual 6 of the National Warm Air Heating and Air Conditioning Association, is as follows:[a]

1. Adjust the fuel input in proper relation to the heat loss of the structure.
2. Determine the temperature rise through the furnace.
3. Adjust the air volume to produce a temperature rise through the furnace of about 90 deg.
4. If adjustable, set the fan switch differential to a minimum of about 15 deg.
5. Adjust the fan switch cut-out point as low as practicable.
6. Balance the system by adjusting dampers to produce even temperature distribution between rooms.
7. Set the room thermostat at the desired room temperature.

Close control of room temperature is facilitated by the use of a room thermostat having a narrow operating differential. The heat anticipating thermostat is an example of such a device.

WARM AIR CEILING PANEL SYSTEMS

Warm air ceiling panel heating systems utilize a false ceiling suspended 3¼ in. from the ceiling joists which have previously been covered on the bottom with sheets of plaster-

(Continued on p. 258)

Table 13.... Determination of Free Area and Total Pressure Loss of Register for 22 Deg Deflection of Air[a, b, c, d, e]

Register Free Area in Square Inches (Upper value in each group)

Pressure Loss in Inches of Water (Lower value in each group)

Cfm	Residential Use		Cfm	Residential Use		Cfm	Distance from Register to Opposite Wall						
	Reg. Size or Equivalent	Pressure Loss		Reg. Size or Equivalent	Pressure Loss		31–34	35–39	40–49 A′	50–59 B′	60–69	70–79	80–89
Up to 59	10 x 4 10 x 6	.01 .01	100–119	12 x 6 14 x 4	.02 .02	700–739	243 0.02	186 0.03	127 0.06	85 0.11			
60–69	10 x 6 12 x 4	.01 .02	120–129	12 x 6 14 x 6	.02 .02	740–779	271 0.02	208 0.03	141 0.05	94 0.10			
70–99	10 x 6 12 x 6	.02 .02	130–169	14 x 6	.02	780–819		230 0.02	157 0.05	105 0.09			
			170–189	14 x 8	.02	820–859		254 0.02	173 0.04	115 0.08			

Cfm	Distance From Register to Opposite Wall									Cfm	Distance from Register to Opposite Wall								
	Up to 18	19–21	22–24	25–27	28–30 A	31–34	35–39	40–49	50–59		31–34	35–39	40–49 A′	50–59 B′	60–69	70–79	80–89
190–209	68 0.02	49 0.03	38 0.05	29 0.08	24 0.11	19 0.16				860–899		278 0.02	190 0.04	126 0.08			
210–229	82 0.02	60 0.03	45 0.04	35 0.06	28 0.10	23 0.13				900–930		304 0.02	207 0.04	138 0.07			
230–249		71 0.02	54 0.04	42 0.06	34 0.08	28 0.11	22 0.17	B		940–979		332 0.02	226 0.03	151 0.07	108 0.12		
250–269		84 0.02	63 0.03	49 0.05	40 0.07	32 0.10	25 0.16			980–1019		360 0.02	245 0.03	163 0.06	118 0.11		
270–299		100 0.02	76 0.03	60 0.04	48 0.06	38 0.09	30 0.13			1020–1059		390 0.02	265 0.03	177 0.06	127 0.11		
300–339			96 0.02	73 0.04	60 0.05	48 0.07	37 0.11			1060–1099			285 0.03	190 0.06	137 0.10		
340–379			122 0.02	95 0.03	78 0.04	61 0.06	47 0.09	33 0.18		1100–1139			307 0.03	204 0.05	147 0.09		
380–419				117 0.02	94 0.03	75 0.05	58 0.08	39 0.15		1140–1179			330 0.02	220 0.05	158 0.09		
420–459				142 0.02	113 0.03	91 0.04	70 0.06	47 0.13		1180–1219			353 0.02	235 0.04	169 0.08		
460–499				168 0.02	135 0.03	108 0.04	83 0.06	56 0.11		1220–1259			376 0.02	251 0.04	180 0.08		
500–539					159 0.02	126 0.03	97 0.05	66 0.10		1260–1299			401 0.02	267 0.04	192 0.07	136 0.13	
540–579					185 0.02	147 0.03	113 0.04	77 0.08		1300–1339			426 0.02	284 0.04	204 0.07	144 0.13	
580–619					212 0.02	169 0.03	130 0.04	88 0.08	51 0.18	1340–1379			452 0.02	301 0.04	217 0.06	154 0.12	109 0.22
620–659						192 0.02	147 0.03	100 0.07	59 0.15	1380–1419			479 0.02	319 0.04	230 0.06	163 0.12	116 0.21
660–699						217 0.02	166 0.03	113 0.06	75 0.13	1420–1459			508 0.02	338 0.03	244 0.06	172 0.11	122 0.20
						A		B		1460–1500			536 0.02	356 0.03	256 0.06	182 0.10	129 0.19
														A′	B′		

[a] If register selected based on distance from register to opposite wall is unsatisfactory on account of size or pressure loss, it is permissible to shift one or more spaces left or right in the tables to obtain a more suitable register. If requirements fall in blank space, select two registers in place of one and divide CFM capacity between the two registers.

[b] Total pressure (static plus velocity) loss is based on FLAT FACE ADJUSTABLE BAR TYPE and does NOT include stackhead.

[c] Values on the right of line A and A′ should not be used in applications such as churches, auditoriums, and concert halls.

[d] Values on right of line B and B′ should not be used in applications such as residential work, motion picture theaters, court rooms, and schools.

[e] For floor and baseboard registers where a velocity of approximately 300 FPM is used, the free area = $\frac{\text{CFM} \times 144}{300}$ or approximately, $\frac{\text{CFM}}{2}$. Assume a pressure loss of 0.01.

Table 14 . . . Determination of Free Area and Total Pressure Loss of Register for No Deflection of Air[a, b, c, d, e]

Register Free Air in Square Inches (Upper value in each group)
Pressure Loss in Inches of Water Column (Lower value in each group)

CFM	Distance from Register to Opposite Wall								
	19–21	22–24	25–27	28–30	31–34	35–39	40–49	50–59	60–69
					A	B			
190[f]–209	62 0.02	47 0.03	37 0.04	30 0.06	24 0.09	18 0.15			
210[f]–229	76 0.02	58 0.02	45 0.04	36 0.05	29 0.08	22 0.12			
230[f]–249		69 0.02	53 0.03	43 0.04	34 0.07	26 0.11			
250[f]–269		81 0.02	63 0.03	50 0.04	40 0.06	31 0.09	21 0.18		
270[f]–299		93 0.02	73 0.02	59 0.03	47 0.05	36 0.08	24 0.16		
300[f]–339			95 0.02	77 0.03	61 0.04	46 0.06	32 0.12		
340[f]–379			120 0.02	97 0.02	77 0.03	59 0.05	40 0.10		
380[f]–419				119 0.02	95 0.03	73 0.04	50 0.08		
420[f]–459				149 0.02	114 0.03	88 0.04	60 0.07		
460[f]–499				171 0.02	136 0.02	105 0.03	71 0.06		
500[f]–539					160 0.02	123 0.03	84 0.05		
540[f]–579					186 0.02	143 0.02	97 0.05	65 0.09	47 0.17
580[f]–619					213 0.02	164 0.02	111 0.04	74 0.08	54 0.15
620[f]–659						180 0.02	127 0.04	85 0.07	61 0.14
								A	B

CFM	Distance from Register to Opposite Wall					
	35–39	40–49	50–59	60–69	70–79	80–89
			A′	B′		
660–699	210 0.02	143 0.03	95 0.07	69 0.12	49 0.22	
700–739	236 0.02	160 0.03	107 0.06	77 0.11	54 0.21	
740–779	262 0.02	179 0.03	119 0.06	86 0.10	61 0.18	
780–819	291 0.02	198 0.03	132 0.05	95 0.09	67 0.17	
820–859	320 0.02	218 0.03	145 0.05	105 0.08	74 0.15	
860–899	352 0.02	240 0.02	160 0.04	115 0.08	81 0.14	
900–939	385 0.01	262 0.02	175 0.04	126 0.07	89 0.13	
940–979	419 0.01	285 0.02	190 0.04	137 0.07	97 0.12	
980–1019	455 0.01	309 0.02	206 0.04	148 0.06	105 0.11	
1020–1059	493 0.01	335 0.02	223 0.03	161 0.06	113 0.11	
1060–1099	530 0.01	361 0.02	241 0.03	173 0.06	123 0.10	
1100–1139	571 0.01	388 0.02	258 0.03	186 0.05	132 0.09	93 0.17
1140–1179		416 0.02	277 0.03	199 0.05	141 0.08	100 0.16
1180–1219		446 0.02	297 0.03	214 0.04	151 0.08	107 0.15
1220–1259		476 0.02	317 0.03	228 0.04	161 0.07	116 0.14
1260–1299		507 0.01	338 0.02	243 0.04	172 0.07	122 0.13
1300–1339		539 0.01	359 0.02	258 0.04	183 0.07	130 0.12
1340–1379			382 0.02	274 0.04	195 0.07	138 0.12
1380–1419			403 0.02	290 0.03	206 0.06	146 0.12
1420–1459				308 0.03	218 0.06	155 0.10
1460–1500				324 0.03	230 0.06	163 0.10
				A′	B′	

Footnotes for Table 14

[a] If register selected based on distance from register to opposite wall is unsatisfactory on account of size or pressure loss, it is permissible to shift one or more spaces left or right in the tables to obtain a more suitable register. If requirements fall in blank space, select two registers in place of one and divide CFM capacity between the two registers.

[b] Total pressure (static plus velocity) loss is based on FLAT FACE ADJUSTABLE BAR TYPE and does NOT include stackhead.

[c] Values on the right of line A and A′ should not be used in applications such as churches, auditoriums, and concert halls.

[d] Values on right of line B and B′ should not be used in applications such as residential work, motion picture theaters, court rooms, and schools.

[e] For floor and baseboard registers where a velocity of approximately 300 FPM is used, the free area = $\frac{\text{CFM} \times 144}{300}$ or approximately, $\frac{\text{CFM}}{2}$. Assume a pressure loss of 0.01.

[f] For any volume under 190 CFM use Table 13.

Table 15 . . . Pressure Drop in Duct

Inches of Water per 100 Feet of Duct Length

Equivalent Length of Duct (Ft)	Total Pressure Drop in Duct (In. of Water)																	
	0.04	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17	0.18	0.19	0.20
35–44	0.10	0.13	0.15	0.18	0.20	0.23	0.25	0.28	0.30	0.33	0.35	0.38	0.40	0.43	0.45	0.48	0.50
45–54	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34	0.36	0.38	0.40
55–64	0.07	0.08	0.10	0.12	0.13	0.15	0.17	0.18	0.20	0.22	0.23	0.25	0.27	0.28	0.30	0.32	0.33
65–74	0.06	0.07	0.09	0.10	0.11	0.13	0.14	0.16	0.17	0.19	0.20	0.21	0.23	0.24	0.26	0.28	0.29
75–84	0.05	0.06	0.08	0.09	0.10	0.11	0.13	0.14	0.15	0.16	0.18	0.19	0.20	0.21	0.23	0.24	0.25
85–94	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.16	0.17	0.18	0.19	0.20	0.21	0.22
95–104	0.04	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17	0.18	0.19	0.20
105–114	0.04	0.05	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.15	0.16	0.17	0.18
115–129	0.03	0.04	0.05	0.06	0.07	0.08	0.08	0.09	0.10	0.11	0.12	0.12	0.13	0.14	0.15	0.16	0.17
130–149	0.03	0.04	0.04	0.05	0.06	0.07	0.07	0.08	0.09	0.09	0.10	0.11	0.11	0.12	0.13	0.14	0.14
150–169	0.03	0.03	0.04	0.04	0.05	0.06	0.06	0.07	0.08	0.08	0.09	0.09	0.10	0.11	0.11	0.12	0.13
170–189	0.02	0.03	0.03	0.04	0.04	0.05	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.09	0.10	0.11	0.11
190–214	0.02	0.03	0.03	0.04	0.04	0.05	0.05	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.09	0.10	0.10
215–239	0.02	0.02	0.03	0.03	0.04	0.04	0.05	0.05	0.05	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.09
240–264	0.02	0.02	0.02	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.06	0.06	0.06	0.07	0.07	0.08	0.08
265–289	0.01	0.02	0.02	0.03	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.05	0.06	0.06	0.07	0.07	0.07
290–324	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.05	0.06	0.06	0.06	0.07
325–374	0.01	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.06
375–424	0.01	0.01	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.04	0.05	0.05	0.05
425–474	0.01	0.01	0.01	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.04	0.05
475–524	0.01	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.04	0.04	0.04
525–574	0.01	0.01	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03	0.04
575–625	0.01	0.01	0.01	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03

Equivalent Length of Duct (Ft)	Total Pressure Drop in Duct (In. of Water)																
	0.21	0.22	0.23	0.24	0.25	0.26	0.27	0.28	0.29	0.30	0.32	0.34	0.36	0.38	0.40	0.45	0.50
35–44	0.53	0.55	0.58	0.60	0.63	0.65	0.68	0.70	0.73	0.75	0.80	0.85	0.90	0.95	1.00	1.13	1.25
45–54	0.42	0.44	0.46	0.48	0.50	0.52	0.54	0.56	0.58	0.60	0.64	0.68	0.72	0.76	0.80	0.90	1.00
55–64	0.35	0.37	0.39	0.40	0.42	0.43	0.45	0.47	0.48	0.50	0.53	0.57	0.60	0.64	0.67	0.75	0.83
65–74	0.30	0.32	0.33	0.34	0.36	0.37	0.39	0.40	0.42	0.43	0.46	0.49	0.52	0.54	0.57	0.64	0.72
75–84	0.26	0.28	0.29	0.30	0.31	0.33	0.34	0.35	0.36	0.38	0.40	0.43	0.45	0.48	0.50	0.56	0.63
85–94	0.23	0.25	0.26	0.27	0.28	0.29	0.30	0.31	0.32	0.33	0.36	0.38	0.40	0.42	0.45	0.50	0.56
95–104	0.21	0.22	0.23	0.24	0.25	0.26	0.27	0.28	0.29	0.30	0.32	0.33	0.36	0.38	0.40	0.45	0.50
105–114	0.19	0.20	0.21	0.22	0.23	0.24	0.25	0.26	0.27	0.28	0.29	0.31	0.33	0.35	0.37	0.41	0.46
115–129	0.18	0.18	0.19	0.20	0.21	0.22	0.23	0.23	0.24	0.25	0.27	0.28	0.30	0.32	0.33	0.38	0.47
130–149	0.15	0.16	0.16	0.17	0.18	0.19	0.19	0.20	0.21	0.21	0.23	0.24	0.26	0.27	0.29	0.32	0.36
150–169	0.13	0.14	0.14	0.15	0.16	0.16	0.17	0.18	0.18	0.19	0.20	0.21	0.23	0.24	0.25	0.28	0.31
170–189	0.12	0.12	0.13	0.13	0.14	0.15	0.15	0.16	0.16	0.17	0.18	0.19	0.20	0.21	0.22	0.25	0.28
190–214	0.11	0.11	0.12	0.12	0.13	0.13	0.14	0.14	0.15	0.15	0.16	0.17	0.18	0.19	0.20	0.23	0.25
215–239	0.09	0.10	0.10	0.11	0.11	0.12	0.12	0.13	0.13	0.13	0.14	0.15	0.16	0.17	0.18	0.20	0.22
240–264	0.08	0.09	0.09	0.10	0.10	0.10	0.11	0.11	0.12	0.12	0.13	0.14	0.15	0.15	0.16	0.18	0.20
265–289	0.08	0.08	0.08	0.09	0.09	0.10	0.10	0.10	0.11	0.11	0.12	0.12	0.13	0.14	0.15	0.16	0.18
290–324	0.07	0.07	0.08	0.08	0.08	0.09	0.09	0.09	0.10	0.10	0.11	0.11	0.12	0.13	0.13	0.15	0.17
325–374	0.06	0.06	0.07	0.07	0.07	0.08	0.08	0.08	0.09	0.09	0.09	0.10	0.10	0.11	0.11	0.13	0.14
375–424	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.07	0.08	0.08	0.08	0.09	0.09	0.09	0.10	0.11	0.13
425–474	0.05	0.05	0.05	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.07	0.08	0.08	0.09	0.09	0.10	0.11
475–524	0.04	0.04	0.05	0.05	0.05	0.05	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.10
525–574	0.04	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.05	0.06	0.06	0.07	0.07	0.07	0.07	0.08	0.09
575–625	0.04	0.04	0.04	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.05	0.06	0.06	0.06	0.07	0.08	0.08

Table 16.... Suggested Bonnet Pressure
Inches of Water

Total CFM Through Any One Duct	Suggested Bonnet Pressure In. Water	Total CFM Through Any One Duct	Suggested Bonnet Pressure In. Water
800–1000	0.10	3500 to 5000	0.15
1000–1200	0.10	5000 to 7500	0.25
1200–1800	0.10	7500 to 10,000	0.375
1800–2400	0.13	10,000 to 12,000	0.500
2400 to 3500	0.14	12,000 to 14,000	0.750

board. Special hangers are used to suspend the false ceiling or heating panel from the joists. Steel supporting rods are installed through these hangers and metal lath is attached to the rods. The lath is then plastered to a thickness of ¾ in., making a completely sealed air space above the ceiling. Warm air is delivered to this sealed space through a standard warm air duct, installed in the usual manner. The air is then circulated over the entire ceiling being guided by sheet-metal baffles. After the air has passed over the ceiling, it is returned to the furnace through a return air duct for reheating. The system is closed, and no air is introduced into the heated space from the panel. (See also Chapter 30, Panel Heating.)

Because a warm air ceiling panel system involves a different type of ceiling construction, its installation is practically limited to new construction. Only automatically-fired and thermostatically-controlled furnace-blower units may be used with this system. The standard automatic heating controls consisting of a room thermostat, temperature limit control, blower control (fan switch), and primary control are all that are required. While warm air ceiling panel heating has particular advantages in one-story utility room houses, it can be just as effectively installed in one- and two-story houses with basements.

Complete design and installation procedure is given in Manual 7-A of the National Warm Air Heating and Air Conditioning Association.[11]

SUMMER OPERATION

During hot summer periods a slight cooling effect may sometimes be obtained by the circulation of basement air through the system. This beneficial effect is due to both the increased air movement in the occupied spaces and the small reduction in dry-bulb temperature brought about by passing warm air over the cooler basement walls and floor. In areas of high prevailing dew points, however, this practice may cause objectionable dampness to develop in the basement.

Complete summer air conditioning can be readily combined with forced warm air systems either by using factory-built units incorporating both refrigeration and heating equipment, or in *some cases* by adapting separate refrigeration equipment or well-water coils to *existing* warm air systems.

A more complete treatment of residential summer air conditioning is given in Chapter 46.

GRAVITY WARM AIR SYSTEMS

In *gravity* warm air heating systems the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the furnace casing and the cooled air entering the bonnet of the casing. Hence, the motive head is very small and the system must allow free flow.

A gravity warm air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal, which is placed in the basement of the building. The heated air, taken from the bonnet on the top of the furnace casing, is distributed to the various rooms of the building through sheet-metal warm air pipes. The warm air pipes in the basement are known as leaders, and the vertical warm air pipes which are installed in the inside partitions of the building are called stacks. The heated air is discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard.

The air supply to the furnace is usually taken entirely from inside the building through one or more recirculating ducts, although in some cases an outside air supply duct is provided.

Gravity warm air heating systems may be designed[12] by following the procedures given in Manual 5 of the National Warm Air Heating and Air Conditioning Association. This publication also contains information terminology, outlet and return locations, and standard sizes of pipe and fittings. The data underlying this information and the design procedure are given in a circular[13] issued by the University of Illinois, and are based on research conducted there.

REFERENCES

[1] *Application Guide for Residential Central Air Conditioning Systems—Winter and Year-Round* (National Warm Air Heating and Air Conditioning Association, Manual 8, Fifth Edition, 1958).

[2] A. P. Kratz and S. Konzo: Performance of a forced warm-air heating system as affected by volume and temperature of air recirculated (ASHVE Transactions, Vol. 48, 1942, p. 393).

[3] *Manual for Adjusting Winter Air Conditioning Systems for Maximum Comfort* (National Warm Air Heating and Air Conditioning Association, Manual 6, 1956).

[4] *Warm Air Perimeter Heating* (National Warm Air Heating and Air Conditioning Association, Manual 4, 1956).

[5] *Four-Inch Pipe Warm Air Perimeter Heating* (National Warm Air Heating and Air Conditioning Association, Manual 10, 1956).

[6] *Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems* (National Warm Air Heating and Air Conditioning Association, Manual 7, 1953).

[7] *Calculating Heat Losses* (National Warm Air Heating and Air Conditioning Association, Manual 3, 1956).

[8] S. Konzo, R. J. Martin, D. S. Levinson, and R. W. Roose: Proposed design procedures for large mechanical warm air heating systems (ASHVE Transactions, Vol. 53, 1947, p. 177).

[9] *Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems and Year 'Round Air Conditioning Systems* (National Warm Air Heating and Air Conditioning Association, Manual 9, 1956).

[10] S. Konzo and A. F. Hubbard: Automatic controls for forced-air heating systems (ASHVE Transactions, Vol. 40, 1934, p. 37).

[11] *Code and Manual for the Design and Installation of Warm Air Ceiling Panel Systems* (National Warm Air Heating and Air Conditioning Association, Manual 7-A, 1950, 3rd ed.).

[12] *Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems* (National Warm Air Heating and Air Conditioning Association, Manual 5, 1954).

[13] A. P. Kratz and S. Konzo: *Simplified Procedure for Selecting Capacities of Duct Systems for Gravity Warm Air Heating Plants* (University of Illinois, *Engineering Experiment Station Circular* 45, December 1942).

CHAPTER 19

CENTRAL SYSTEMS FOR AIR CONDITIONING

Features of Systems, Zoning, Humidity Control, Cooling Load, Heating Load, Air Quantity and Temperature Differential, Unitary-Central Systems, Low- and High-Pressure Induction Convectors, High-Velocity Systems, Fan and Coil Units, Evaporative Cooling, Precooling, Sensible Cooling with Unwetted Coils, Selection of System, Location of Apparatus

THE term, central, applied to an air conditioning system implies that the equipment such as fans, coils, filters and their encasement are designed for assembly in the field rather than in a factory as a unit. As a central system usually serves several different rooms, or spaces, individual controls are required for each room.

FEATURES OF CENTRAL SYSTEMS

One advantage of a central air-supply system is that one apparatus serving many rooms may involve a lower investment cost than that for a number of self-contained plants, each serving a single room. A central system may occupy basement or attic space that is relatively unimportant, whereas individual factory-assembled apparatus placed in each room may occupy otherwise valuable space. Another advantage of a central system is accessibility for servicing, since it is possible to provide doors in the encasement for cleaning and inspecting all of the component parts in a manner usually superior to that practicable with compact factory-assembled equipment. In addition service is concentrated in one or just a few places.

Central air-conditioning systems usually are connected by ducts with the various rooms served, and preferably have exhaust fans that may effect complete removal and disposal of any desired proportion of the air. The return-air fan may return air to the supply system for recirculation, as a measure of economy of fuel or refrigeration.

Central air-conditioning systems are served by heating and refrigerating equipment which may be located at some distance from the air-supply apparatus, and which may serve one or more central air-supply systems.

Fig. 1 is a sketch of a typical year-round central system. Outdoor air enters from an intake preferably on that side of the building least exposed to solar heat, and not close to the ground or to a sun-heated or dust-gathering roof. The outdoor-air damper is split into two sections, minimum outdoor-air and maximum outdoor-air, with the maximum outdoor-air damper interlocked with the return-air damper in such a manner that as the outdoor-air volume increases the return-air volume decreases. The return-air duct or connection could come from a return-air fan. All the air passes through filters which must have ample room on both sides of the filters for servicing. The filters may be of a mechanically-cleaned type, a replaceable-cell type, or may be electronic as described in Chapter 24.

The cleaned air passes to the equipment that changes its temperature and humidity. Except in very warm climates, heating or tempering coils are required to warm the entering outdoor air to a temperature above freezing, the heat being supplied by means of hot water or steam. Cooling coils provide the necessary sensible cooling and dehumidification. The coils may be chilled by direct expansion of an approved refrigerant within the tubes, or by a pump-circulated liquid such as water or brine. A water-tight drainage tank must be installed under the cooling coil and should extend for a distance toward the fan.

Reheater coils, utilizing steam or hot water, reheat the air in warm weather for control purposes or bring the air to its final temperature in cold weather. For humidification in dry weather, water sprays are used either separately as shown or combined with the cooling coils having the spray nozzles directed against the coil surfaces. The spray water may be circulated by a small pump from the water tank under the spray chamber.

The general requirements for the control of the various components of the central station apparatus are covered

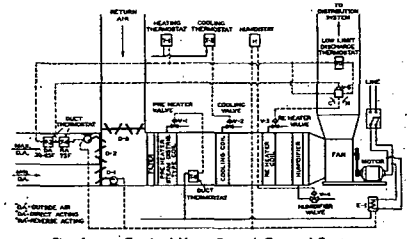


Fig. 1.... Typical Year-Round Central System

in Chapter 43. The functioning of a typical set of controls as illustrated in Fig. 1 is as follows:

Whenever the fan is started, solenoid air valve or relay E-1, actuated by the fan motor starter, opens minimum outdoor-air damper D-1, places humidistat H in service, and allows duct thermostats T-3 and T-4 to control the maximum outdoor-air damper D-2 and the return-air damper D-3.

When the fans stop, E-1 is de-energized to close the outdoor-air dampers and also to close humidifier valve V-4.

Thermostat T-1 positions steam valve V-3, on the reheater coil, to maintain a constant space temperature. As the space temperature rises, T-1 positions reheater valve V-3 to a closed or to a minimum open position, as determined by low limit discharge thermostat T-5. Duct thermostat T-6, in the preheater discharge, positions preheater coil valve V-1 to maintain a constant preheater discharge temperature.

On rising outdoor temperature, between 30 F and 65 F, duct thermostat T-3 located in the outdoor-air intake, moves maximum outdoor-air damper D-2 toward the open position

provided that T-1 is satisfied. At 65 F outdoor, D-2 will be fully open and return-air damper D-3 will be fully closed.

As the outdoor-air temperature rises above 65 F, duct thermostat T-3 positions V-5 in such a way as to bypass low limit thermostat T-5 so that reheater coil valve V-3 is operated directly from thermostat T-1. As the outdoor-air temperature rises from 65 F to 75 F, duct thermostat T-4 gradually closes maximum outdoor-air damper D-2 and opens return-air damper D-3.

Cooling thermostat T-2 positions cooling coil valve V-2 to admit more chilled water as the space temperature rises.

Humidistat H positions humidifier valve V-4 to maintain the desired humidity in the conditioned space.

ZONING AND ZONE CONTROL

It is apparent that while apparatus like that of Fig. 1 would be very desirable for any single room, since in that case the air could be delivered at optimum conditions, the cost of a complete individual system for each room and the space required for the equipment generally would be prohibitive. Economy is favored if the varying requirements of numerous rooms or zones can be simultaneously satisfied by air from a single central supply system.

Various methods are practicable for controlling the temperature, humidity and air movement in various rooms or zones. A measure of control is attainable merely by proportioning the flow of air to each room, though usually such control by throttling dampers is difficult to maintain and should be avoided when possible.

Another scheme is to install a properly proportioned coil in the branch air-supply duct serving each room or zone to warm the air to suit the occupants. For example, the air leaving the fan that serves several zones may be cooled, before entering the fan, to the condition favorable for one zone, and the air for each other zone may be reheated by the branch duct coil to the required temperature. It is also possible to circulate a heat absorbing medium in the branch duct coils to reduce the temperature of the air passing to rooms that would be overheated if they received air at the condition leaving the central air-supply system. When heat transfer devices are placed in branch ducts for improved temperature control, mechanically-circulated water gives excellent results as a heat carrier. The water usually is warmer than the air but it is possible to use water colder than the air.

It is practicable also to use single central air-conditioning equipment similar to that shown in Fig. 1, in conjunction with several fans, one for each room or zone. In such cases there may be a separate reheater on the suction side of each relatively small supply fan.

The designer must remember that the various supply fans will compete with each other for air, against the resistance interposed by the filters, coils, etc., that are used in common under such circumstances, and consequently, unless the fans are of backward-curved blade, non-overloading type, they may alternate in carrying more than their share of the air, and thereby cause the air distribution to be chaotic and unsatisfactory.

Another method of controlling temperature in various rooms served by a central air-supply system is shown in the sectional elevation, Fig. 2. The supply fan is placed immediately after the humidifier. When cooling the air in hot weather, the humidifier is not operated. The fan will deliver the air through the heating coil and through the cooling coil to the two air-pressure chambers *A* and *B* at the right of these coils. From these chambers many separate ducts, one of which is shown, each with a double-blade mixing damper, may convey the air to the various rooms. The mixing dampers, one of which is shown, are interlocked so that as the upper one closes, the lower one opens; selecting between them, air in the required quantity from either the warmer chamber *A* or the cooler one *B*. In cold weather no refrigerant is required in the cooling coil, and in hot weather no heating medium is circulated in the heating coil. With this scheme, the control of relative humidity in warm weather is not always sufficiently precise to meet requirements, since the untreated air delivered through the upper coil may be so high in relative humidity that it cannot compensate sufficiently for the nearly saturated air leaving the lower coil. A reheater could be placed if desired, at the right of the lower coil to bring the air in the lower chamber to the desired relative humidity. The simple arrangement of Fig. 2 is admirable in winter and, except where close control of relative humidity is important, may be acceptable in summer.

Another method of attaining temperature control in individual rooms with a year-round central air-supply system, is to install a booster fan between the main air-supply duct and the air-delivery opening to each zone or room, as shown in Fig. 3. Air can then be delivered from the central supply fan through the main duct at some desired condition, for instance, 60 F and 45 percent relative humidity. A double mixing damper near the intake opening of the booster fan, controlled by a thermostat in the room or zone that is served by the fan, is interlocked with an outlet exhaust damper in the spent-air opening, so that as more of the room air is recirculated, and as less new air from the main air-supply duct is delivered into the room, the spent-air outlet is throttled in proportion. In many large installations this principle is applied successfully for zoning different stories in multi-story office buildings, the main supply fan being on the roof, and each booster fan being used for supplying the rooms of one orientation of each story. In other cases the booster fans serve only single offices, and therefore are small enough to be concealed above ceilings alongside the main supply duct.

There may be installations in which the use of recirculated air for mixing with new refrigerated and nearly saturated air to control temperature and relative humidity is objectionable. In such cases the general recirculation arrangements of Fig. 1 may be omitted, and heat transfer coils located in the ducts may be used. In some cases where general recirculation is not acceptable, as for all the rooms in an entire building, use of the local circulation of Fig. 3 may solve the problem.

HUMIDITY CONTROL

In winter, room relative humidities in excess of 30 percent are seldom required in a system designed for comfort conditioning only, and a low saturating efficiency may be desirable, or even necessary, especially if the same volume of

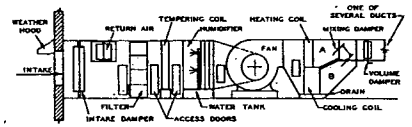


Fig. 2 . . . . Alternate Arrangement of Equipment for Controlling Air Conditions in Central-Air Supply System

air is handled as in summer. With a spray-type dehumidifier the main sprays may be shut off and only the eliminators need be flooded; which may give sufficient moisture. In other cases, such as those in which cooling coils are sprayed, the spray-water supply may be throttled. If the saturation efficiency of the sprays is too low, the spray water may be heated. The amount of heat put into the spray water by open or closed water heaters will be equal to that required to bring the dew-point temperature of the air entering the sprays up to that required before entering the preheater. It is possible, where clean steam is available, to introduce steam directly into the air stream to produce the desired dew-point temperature of supply air. However, the steam must be exceptionally clean, or objectionable odors will result.

It should be noted that the quantity of outdoor air to be introduced is affected by infiltration and leakage. Infiltration will reduce the quantity to be introduced by the system, while leakage may have to be offset by an increase in the quantity of outdoor air.

COOLING LOAD

The method of determining the cooling load for a conditioned space or spaces is outlined in Chapter 13. As pointed out therein, many of the items of heat gain are variable and do not reach their maximum values simultaneously. Proper consideration of these peaks and the avoidance of pyramiding these peaks in the cooling load calculations are stressed. Maximum solar heat gain on an east exposure is seldom coincident with the maximum outdoor wet-bulb temperature. A discussion of the factors affecting the actual instantaneous cooling load will be found in Chapter 13.

HEATING LOAD

Methods of calculating the heating load are shown in Chapter 12. Many of the factors outlined in Chapter 13, Cooling Load, such as zoning, non-simultaneous peaks, and diversity, apply in the reverse manner due to the heating requirements instead of the cooling requirements. However, these factors affect the heating load from the standpoint of control of inside conditions, overall performance, and economy of operation more than from a capacity of equipment standpoint. It is not only necessary to heat a building or space to its design conditions when there is but the merest fraction of normal occupancy, and when there are practically no lights, internal heat, or solar radiation, but it is also necessary to provide capacity to heat the building quickly when sudden cold follows relatively warm weather, as may

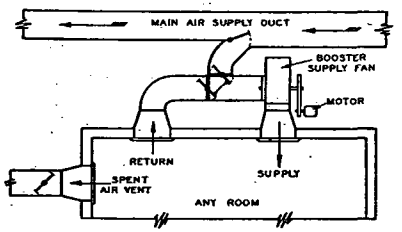


Fig. 3 . . . . Arrangement for Individual-Room Temperature Control with Central Air-Supply System

occur after a week-end or holiday shut-down. However, in normal operation during week-ends and holidays, buildings are usually kept at a holding temperature to prevent the freezing of services. In many cases, less fuel is required to operate the heating plant at a near-normal rate and maintain the building or space at a temperature of 50 to 65 F at such times, than to shut the system down and then bring the temperature back to normal through forced operation of the heat-generating equipment with a consequent loss in efficiency.

AIR QUANTITY AND TEMPERATURE DIFFERENTIAL

The difference between the room-air temperature and the supply-air temperature at the outlet to the room is known as the temperature differential. In the theoretical case of a dehumidifier having 100 percent saturating efficiency, and where this air is delivered directly to the room without temperature increases due to heat gain, then the temperature differential is the difference between room temperature and apparatus dew-point temperature. If duct heat gains are considered a part of the room load, this still holds true. The apparatus dew point, as outlined previously, is fixed by the latent and sensible loads of the space, but in many cases, it is desirable to deliver more air to the spaces than is indicated by the difference between the room temperature and the apparatus dew point.

It has been indicated that where a percentage of air is passed through the dehumidifier without being treated, the relationship is modified in direct proportion, and that if room air is passed through untreated, no effect on the heat balance results. Similarly, if room air is passed around the dehumidifier and mixed with the treated air, the heat balance is not adversely affected. Therefore, if the quantity of air passed through the dehumidifier is determined by the usual methods, room air can be passed around the dehumidifier and mixed with the dehumidified air, increasing the supply-air quantity and temperature and decreasing the temperature differential. Thus if the difference between the room temperature and the apparatus dew point indicates that 10,000 cfm at 30 deg below room temperature will be required to hold conditions, that quantity can be passed through the dehumidifier and cooled to 30 deg below the room temperature, then mixed with 10,000 cfm of room air, resulting in a supply-air quantity of 20,000 cfm and a temperature differential of 15 deg instead of 30 deg. Air-supply outlets and grilles having a high induction ratio are available, and through their induction effect cause a large amount of room air to be mixed with the supply air within a short distance of the grille. A proper selection of outlets may make it possible to introduce air at low temperatures and high velocities without causing objectionable drafts or cold spots, but care must be used to see that too little air motion is not a result. Small temperature differentials may be required for this reason. While the use of a large temperature differential results in a saving in initial cost of fans and ducts, and in the operating cost of fans, this differential should be carefully considered. If the sensible heat load of a space is subjected to substantial variations, small temperature differentials should be considered, since systems employing small temperature differentials require less precision in controls.

Reduction of air quantity by slowing down the fans for the winter season, and increasing the temperature differential, often is feasible. A saving in fan power can thus be effected, provided the air distribution remains adequate.

Extremes should be avoided in all cases. For summer air conditioning, low supply-air temperatures result in larger heat gains to the air passing through the ducts, as well as in poor control. Too high a supply-air temperature may result in excessive initial and operating costs. Suggested limits for the temperature differential are from 12 to 25 deg, the actual selection being based on the requirements of the particular case. For winter air conditioning, too high supply-air temperatures result in excessive heat losses from the ducts and stratification within the room unless thorough mixing is assured, while too low supply-air temperatures may cause drafts, high operating costs, etc. Suggested limits are from 15 to 35 deg. There can be no set rule, and each case should be judged according to its particular requirements of the installation.

Reference may be made to Chapter 20 for further discussion of the most satisfactory design difference between the entering air temperature and volume in relation to the desired room condition.

UNITARY-CENTRAL SYSTEMS

Many different types of central air-conditioning systems with room units of various designs have been developed for multi-room buildings, such as office buildings, hotels, and hospitals. The primary object in using these systems is to save space by reduction of duct sizes or by entirely eliminating ducts. In new buildings, small ducts may reduce the overall building height; in existing buildings, the use of small ducts is frequently imperative and may even play a decisive role in the acceptance of air conditioning for these buildings. Also, multi-room buildings frequently require a high degree of zoning or individual room control, which the systems must supply.

These buildings usually have a large perimeter relative to the floor area. Air-conditioning units are usually installed beneath the windows. Where the spaces to be conditioned extend a considerable distance from the outer wall into the interior of the building, as in some office buildings, a separate system or zone for the conditioning of the interior portions may be required. In these cases the interior system handles the relatively small and steady loads, due to such items as lights and people; while the exterior system must cope with the relatively large and variable loads imposed by sunshine and temperature difference through walls and glass. Descriptions of the four common unitary-central systems follow.

Induction Convectors—Low-Pressure Type

Induction convectors located in the room that is to be served, utilize a jet of primary conditioned air to mix with a stream of secondary room air as shown in Fig. 4. The mixture is discharged into the room through a grille at the top of the convector. Heating coils are located in the secondary-air stream. The output is controlled either by manually or automatically throttling the air jet. Heat may be supplied to the coil in summer as well as in winter. These induction convectors present several advantages. Since the secondary air stream is thoroughly mixed with the high-velocity low-temperature air stream before leaving the discharge outlet of the device, the resultant temperature of the mixture is satisfactory even though the primary air is introduced at a temperature too low for ordinary methods of distribution. One of these devices usually is provided under each window in place of the customary direct radiator, and combines the air-distribution system with the heating system. An air-conditioning system without induction convectors may require installation of direct radiation for maintenance of minimum temperatures during air-conditioning shut-down periods, but when induction convectors are used they may be selected with heating coils of sufficient capacity to maintain, by thermal circulation, a reasonable temperature when the primary-air supply system is shut off. The use of low-temperature, dehumidified air that has not been reheated or mixed with room air before delivery to the room, may permit a reduction in fan capacity and the use of smaller ducts. In some cases a bypass may be desirable in order to maintain the primary-air volume and to provide additional control. This system can provide a degree of zoning that is usually difficult with conventional design because the air delivered by each unit can be controlled individually. Selection of induction convectors should be made with due regard to noise level. The inductive capacity of the device increases with the jet velocity, but high jet velocities may result in objectionable noise.

The units are supplied with primary air from one or more central stations with the primary air providing all ventilation and handling the entire cooling load. The central apparatus is arranged in general conformity to Fig. 1 except that the central reheater is omitted since each unit has its own reheater. The controls are arranged to maintain a substantially-constant supply-air temperature equal to that required for maximum cooling. Final control is obtained by means of a room thermostat, or remote bulb thermostat in the unit having its measuring element in the recirculated-air stream. The controlling thermostat operates the coil supply valve and a damper operator on the primary-air damper in sequence. At maximum heating load the valve is wide open and the primary-air quantity is at a minimum. As the space temperature tends to rise, the valve is first gradually closed, then the primary-air quantity is gradually increased from minimum to maximum. Alternately the primary-air quantity can be kept constant while final control is achieved by the individual heating coils. If variable air quantities are to be handled by the primary-air fan, some form of fan-capacity control is necessary.

Induction Convectors—High-Pressure Type

Another type of induction convector, Fig. 5, employs nozzles that produce a high-velocity air jet without objectionable noise. The term, high pressure, is to some extent inaccurate, since the air pressure at the nozzles, while several times that used with a low-pressure induction convector, is still less than the total resistance pressure of a conventional central system. The high-velocity jet of primary air induces

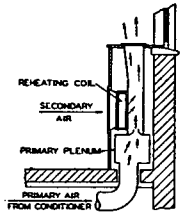


Fig. 4 .... Induction Unit (Low-Pressure Type)

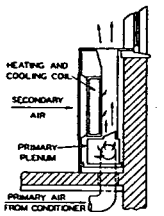


Fig. 5 .... Induction Unit (High-Pressure Type)

a flow of air from the room through coils located in the secondary-air stream and supplied with chilled water in summer and with hot water in winter. The chilled water removes a large portion of the sensible heat in summer and the hot water supplies the sensible heat loss in winter. The primary air is delivered at a sufficiently low dew point to compensate for the latent heat gain in summer. In winter the primary air is supplied at a sufficiently high dew point to take care of latent heat losses. Control of temperature is obtained by throttling the water quantity supplied to the secondary coil. The required flow of primary air is greatly reduced due to the fact that a portion of the sensible heat load is carried by the secondary air stream. Since the primary quantity is small, very high velocities can be maintained in the supply ducts without requiring fan power in excess of that for a conventional system. Therefore, the supply ducts or pipes can be very small and can be run in chases, or furred in at columns along with the water pipes. The primary air is treated in the usual manner to reach the required dew point and a surface or spray dehumidifier or a dehydrator may be used. The primary-air quantity is sufficient for ventilation purposes and frequently consists entirely of outdoor air.

Fig. 6 shows the general arrangement and control of the system. The operation of the controls is as follows:

The final temperatures in the spaces are controlled by individual valves, either automatic or manual, which throttle the flow of secondary water to the unit coils. Where automatic valves are used, the thermostats are of the heating-cooling type so that a rise in room temperature opens the valve when cold water is circulated and closes the valve when hot water is being supplied. Thermostat T1 controls the preheater steam valve V1 to maintain a temperature of about 45 F leaving the preheater. Thermostat T2 controls the maximum outdoor-air and return-air dampers so that the air entering the dehumidifier is maintained at 50 F whenever outdoor air conditions permit.

During the heating cycle, thermostat T3 and valve V2 are inoperative so that cool primary air is supplied to the units but at the same time thermostat T5 operates valve V4 so that the secondary water is heated. Thus the combination of cool primary air and hot secondary water can provide heating or cooling as required. The primary pump and water cooler are inoperative.

When the outdoor air is no longer suitable to provide sufficient cooling through the primary air system (typically about 50 F) the system is changed from heating to cooling operation. The primary pump is started and supplies chilled water to the dehumidifier. Thermostat T4 controls mixing valve V3 so as to maintain the secondary water at temperatures in the range of 50 to 55 F. At outdoor temperatures below 80 F thermostat T3 regulates reheater valve V2 so that warm primary air is supplied at the same time that cold

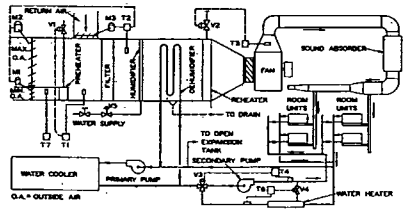


Fig. 6 .... Typical Induction Unit System

secondary water is fed to the units. The changeover from heating to cooling is usually manual and the exact time is not too important because both heating and cooling are available in either case. During the cooling cycle, thermostat T7 acting through thermostat T3 readjusts the primary-air temperature downward as the outdoor-air temperature increases from the changeover point to about 80 F so that above this temperature the primary air is at the apparatus dew point.

In some buildings the primary-air system and water circuits are zoned according to exposure, but because both heating and cooling are available at the units whenever either is needed, such zoning is not always used.

All-Air High-Velocity Systems

All-air high-velocity systems are air-conditioning systems in which the duct velocities and static pressures are such that special control and acoustic equipment is required for proper introduction of the conditioned air into the space to be served.

All-air high-velocity duct systems are of either single-duct or dual-duct types.

The *single-duct system* is usually zone controlled. The single supply temperature is established for each zone and therefore variations in individual rooms must be compensated by throttling the air supply volume. Throttling is limited to the lowest air quantity required for ventilation and for satisfactory air distribution which depends to a large degree on the performance of the type of air outlet used.

Dual-duct systems deliver the entire air supply to cold- and warm-air supply ducts, from which it is distributed into air-mixing valves or acoustic terminal devices. (A mixing valve is a mechanical device that serves to proportion air from a cold-air duct and a warm-air duct into a common outlet, or duct. An acoustic terminal device is an air distribution unit consisting of an air valve, acoustical attenuation chamber, and an air outlet.)

The proportion of cold air and warm air delivered to the air-mixing valves or acoustic terminal device is thermostatically controlled. This control, together with controls for maintaining fixed volumetric delivery, results in a constant volume of air supply at a temperature that satisfies the load.

Mixing valves may be used to deliver large quantities of mixed or blended air at high velocity to a single-duct system equipped with acoustic terminal devices supplying the needs of an entire zone or area. The air coming from the mixing valve may also be reduced in pressure (either in the mixing valve itself or by means of an additional valve or damper) so that it can be distributed by a conventional low-velocity duct system. In this case, noise attenuation is accomplished by a length of lined duct, or by a separate noise attenuator installed after the mixing valve.

For greater flexibility, acoustic terminal devices which usually are equipped with one air outlet only, may discharge into a low-velocity rigid or flexible duct equipped with several air outlets.

Refer to Chapter 21 on Air Duct Design for a discussion of the design of high-velocity ducting.

The general arrangement of a typical dual-duct system is shown in Fig. 7. In some arrangements, two supply-air fans are used with one fan supplying the cold-air duct and the other the hot-air duct. The warm-air fan handles only recirculated air so that operating economy is slightly improved.

The maximum outdoor-air damper and the preheater are controlled as in Fig. 1 so that when the temperature of the

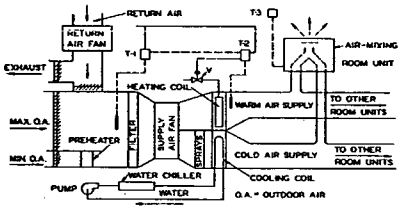


Fig. 7 .... Schematic Arrangement of Dual-Duct System

outdoor air is favorable it is used increasingly instead of return air.

The warm-air supply-duct temperature is controlled by a thermostat, T2, Fig. 7, in the air leaving the heating coil. Thermostat T2 is reset by thermostat T1 in accordance with variations in outdoor-air temperature. Whenever outdoor-air temperature is not low enough to accomplish the required cooling, the cold-air supply-duct temperature is regulated by the cooling coil. The final temperature in the conditioned areas is controlled by the individual air-mixing units.

Air-mixing room units under control of the room thermostat draw warm and cold air from the ducts in accordance with the room requirements, affording individual room control without zoning. Fig. 8 shows schematically one type of room unit designed for dual-duct systems. The warm and cold air pass through the three-way mixing damper which delivers air at the temperature required by the room. Some units are placed under windows, while others are mounted in the ceilings from which air may be delivered to the room through ceiling, wall, or window outlets. Some units are also provided with volume regulators to maintain a steady flow condition.

This system is best suited for multi-room or multi-zone installations, and may be of either high- or low-velocity and high- or low-pressure type with substantially the same control methods. It is often applied to high-velocity systems with long distribution mains, and consequently high-static fan pressures; although extremely high static pressures are seldom justified. With two supply mains, the system static pressures vary as a result of changing demands for heating or cooling. Thus volume regulation often becomes necessary. Extreme variations of flow to different rooms may be somewhat limited by static-pressure regulators and volume dampers at selected points along the mains. There are some installations where volume control at each room supply is desirable to maintain close control regardless of static-pressure changes in the ducts, and incidentally to facilitate balancing of the distribution system.

Air-capacity requirements are generally governed by summer cooling loads, and general practice is to supply the cold air to the cold-air duct at 25 to 30 deg below the established room temperature, this being satisfactory with suitably designed room outlets.

Fan and Coil Units

Another type of room convector used in connection with central cooling and heating plants is known as the *Fan and Coil* Unit (see Fig. 5, Chapter 16).

These units, like induction units, are located around the periphery of a building, usually under windows, and are equipped with fans and a water-type heating-cooling coil. They normally take air for ventilation directly from outdoors, and have a manual damper for adjusting the quantity of outdoor air within certain limits. Positive closing of this damper when the fan stops, can be obtained by adding a damper operator actuated by the fan-motor switch. Hot water supplied during the heating season frequently is varied in accordance with outdoor temperature. Cold water for use during the cooling season should be supplied at a fixed temperature low enough to provide the proper amount of dehumidification.

These units sometimes are used with a primary-air system that supplies the outdoor ventilation air and handles the latent load. In such cases the primary air unit is controlled as explained under the section Unitary-Central Systems, and may be of either the low- or high-pressure type.

The combined heating-cooling coil of each unit may be controlled by a valve in the water supply line to each unit, actuated by a thermostat in the room or recirculated air of the unit. The thermostat is of the heating-cooling type.

EVAPORATIVE COOLING

In climates where, on the hottest days, the outdoor wet-bulb depression is relatively great, it may be possible to replace mechanical refrigeration, or other cooling sources, and use the evaporative cooling effect. A well-designed air washer using recirculating sprays will reduce the entering dry-bulb temperature to within a degree or two of the entering wet-bulb condition. Thus, it may be possible that, with air entering at 100 F dry-bulb, 60 F wet-bulb, a leaving condition of 62 F dry-bulb, nearly saturated, can be obtained. Under some conditions of latent and sensible heat load, this evaporative cooling may be adequate.

At times when the outdoor wet-bulb temperature is not low enough to permit the use of straight evaporative cooling, it is still possible to use precooling convectors with refrigeration, well water, or a cooling tower, as the basic source of sensible heat removal to reduce the wet-bulb temperature of the air before it enters the spray chamber. Where internal heat loads are high, this scheme may be more economical than one using return air. Where the required supply-air dew point is too low to permit straight evaporative cooling, and where the sensible heat load is not too great, intentional partial saturation may be employed. In this case the low dew point of the outdoor air is utilized by permitting some of this air to pass through the humidifying sprays untreated, or to bypass the humidifier. All of these

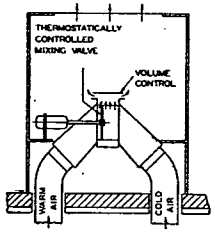


Fig. 8 .... Room Air-Mixing Unit for Dual-Duct System

remarks with regard to evaporative cooling are based on the assumption that all of the supply air will be taken from outdoors. Provision should be made in most cases for the return of some air from the conditioned spaces for control purposes, as well as for economy of fuel in winter.

PRECOOLING

Where sufficiently cold water from wells or streams is available, a saving in refrigeration may be obtained by the use, in location ahead of the dehumidifier, of precooling coils through which the cold water is circulated. The resultant cooling of the air decreases the load to be carried by the dehumidifier and refrigeration plant. In normal practice the water, after passing through the precooling coils, may be further utilized in the refrigeration plant condenser. The economic advantages of this scheme are apparent, and it is frequently used.

SENSIBLE COOLING WITH UNWETTED COILS

Under favorable atmospheric conditions where a large wet-bulb depression exists and the dew point of the outdoor air is sufficiently low at all times, acceptable cooling may be obtained by removing only the sensible heat from the outdoor air delivered to the rooms. Under this condition of a great wet-bulb depression, a temperature-reducing coil may be located in the air stream and supplied with water from a cooling tower. When humidity control is desired, sprays to saturate or partially saturate the air may be used downstream from the unwetted coil. Saturation or partial saturation after the coil will reduce further the dry-bulb temperature and the air quantity required. This system has very definite application in hot dry climates.

SELECTION OF TYPE OF SYSTEM

Low buildings with large floor areas may be divided into sections or zones with separate central air-supply systems to facilitate temperature control. In the case of large department stores it may be possible to provide a single conditioner, with a fan delivering the conditioned air to local mixing fans which supply the various departments or spaces. This application is limited by the practicability of running the large conditioned-air ducts to the various recirculating fans. Each vertical section of the building also may be supplied by a separate fan delivering conditioned air to local mixing fans. In many cases the most economical and satisfactory scheme may be to employ a hot water or steam reheater in each branch duct. Where vertical sectionalizing is not indicated, the building may be divided into horizontal groups, each handled by a central system and adequately zoned. In some large buildings, apparatus rooms for the systems may be located in the basement and attic and on intermediate floors.

In high buildings the necessity for horizontal sectionalizing may be suggested by the size of air-supply and return risers, and by the extent to which they encroach upon usable space. Each story should be cut off by doors from other stories, as otherwise the cool air tends to collect in the lower story and the warm air is forced to the upper story.

Balconies and large lobbies in theaters and similar high rooms frequently justify the use of separate zoning fans, to counteract the tendency of the heavier, cooler air to collect at the lower levels of these spaces.

Fans operate at full capacity continuously in many systems, and therefore should be selected for good efficiencies. In winter when higher temperature differentials are used, it is sometimes practicable to deliver smaller air quantities than when cooling.

In climates where winter temperatures fall below freezing, the tempering coils should be of the steam-distributing type; or if they are heated by a liquid, this liquid should contain some antifreezing substance such as ethylene glycol. If hot water is used in cold climates, the control of air temperature should be obtained by means of face and bypass dampers rather than by throttling the valves, to prevent damage due to freezing.

If zone reheaters placed in supply-duct branches are employed, they should be of such type as to be heated over the entire surface so that no temperature stratification can occur in the delivered air. Steam-distributing-tube coils or mechanically-circulated water coils are satisfactory in such cases, and throttling valves may be used.

Refrigeration equipment must be carefully selected to satisfy the particular requirements of each installation. For some small plants the evaporator may be placed in the air stream, when type of refrigerant and nature of occupancy permit. In many cases, chilled water coils are required by considerations of safety. Where low temperature and relative humidity are necessary, brine, often of calcium chloride, may be indicated.

Condensing requirements must have economic analysis. Wells, public water service, cooling towers, and evaporative condensers present possibilities for consideration. Condenser water may have a secondary use for roof sprays in hot weather, and is usually suitable for lawn sprinkling. Most health department rules in cities prohibit any connection from refrigerant condensers that might permit the water to be used for drinking or lavatory purposes.

Practically without exception, air cleaners should be provided for both outdoor and recirculated air.

Control of temperature and of relative humidity by automatic means is vital, if comfort and economical operation of air-conditioning equipment are to be attained.

The insulation of ductwork may be necessary to prevent condensation of moisture either inside or outside of ducts, housings, and fan encasements or to prevent a change of temperature in ducts passing through hot or cold spaces.

In general, the apparatus should be arranged to have a straight-line air flow. Each change in direction increases air resistance and, in addition, elbows and offsets may cause eddy currents resulting in stratification. The usual order of equipment location, beginning at the outdoor-air intake, is: weather hood or louvers, outdoor-air dampers, return-air connection, filters, tempering coils, cooling coils or sprays, bypass connection with or without preheaters, reheaters, fan and distributing ducts.

Screens at the intake prevent the entry of large foreign matter, birds, etc. A hood or louver at the outdoor-air intake prevents the entry of rain and snow. Since in most climates there are many days during which use of 100 percent outdoor air unheated or uncooled may be economical, the areas of all air-passing and treating apparatus should be large enough for such a volume, and the exhaust or spent-air equipment should be capable of discharging out of doors all of the air admitted.

The bypass connection normally connects the return-air duct system with the apparatus casing between the conditioner and the supply fan. Usually the bypass opening is sized to handle about 50 percent of the fan capacity where a variable bypass is used though extreme load variations

may require a larger size. It is at times good design to locate a reheating coil in the bypass connection to permit using some bypass air when heating is required. Since the relatively high resistance of the cooling coil or spray is to be balanced by the heating coil and bypass connection, enough heating surface can be provided to raise the temperature of the bypass air to the point where the mixture of bypassed air and conditioned air will have the required temperature. When a variable bypass is used, a damper working in opposition to the bypass damper should be placed across the face of the dehumidifier, for unless the resistances of the two are carefully balanced at all operating points, the proper mixtures of air will not be obtained. Outdoor air that has not been dehumidified should not be bypassed around a cooling coil or spray dehumidifier if accurate control of the delivered relative humidity is desired. Where the bypass is made a part of the dehumidifier or conditioner and is located on the top or side of it, the return-air connection should be arranged so that stratification of return air is insured, baffles being provided to accomplish this purpose if necessary. When return-air and bypass air connections are taken off a return-duct system, it may be necessary to install a back-draft damper between the return-air connection and the bypass connection. When the bypass damper is at maximum opening it may be much easier for outdoor air to pass through the return damper, into the return-duct connection and through the bypass, than for return air to pass through the bypass connection into the fan. Air tends to take the path of least resistance and, if the dehumidifier resistance is high, and if the return duct resistances are low, this situation is apt to occur. A recirculating air fan, instead of a back-draft damper, may be required for this case, if the failure of return air to reach the dehumidifier or conditioner is a serious matter under reduced load conditions.

LOCATION OF APPARATUS

In general, the outdoor-air intake, preheaters, and return-air connections precede the conditioner, while the bypass, reheaters, and fan follow it. In the case of a blow-through system, where the fan is located ahead of the conditioner, the leakage of air at the conditioner is outward, instead of inward, and may be accompanied by water leakage.

The location of the complete apparatus assembly, including the dehumidifier, will be dependent on the type of building, spaces available, structural characteristics, etc. The type of conditioner used may limit the location under certain conditions. Where cooling coils employing chilled water or brine as the cooling agent are used, there are few limitations with regard to location other than those of pumping power, working pressures, piping costs, etc. Where spray dehumidifiers are used, very definite limitations present themselves, and these may require certain extraneous equipment to make the system workable. If several spray-type dehumidifiers are located on different levels, thus involving different water pressures, a surge or storage tank, to which the return water from each dehumidifier can be taken, is required. Should the water level in the pan of the dehumidifiers be low in relation to that of the surge tank, return-water pumps will be required, and these pumps will have to be operated until the water supply lines are drained in order to prevent flooding of the lower dehumidifiers. Where spray dehumidifiers are on the same level, equalizing lines between the pans may be required if a storage tank is not provided. It is exceedingly important that watertight drained floors be provided under all overhead cooling systems, since water condensed out of the air generally will be present and may damage the interior finish of the rooms below the apparatus.

All of the various pieces of equipment from the outdoor-air intake through the fan usually are connected together by sheet-metal casings. Frequently, the building structure or specially constructed walls or partitions may be used to form a portion of the casing. In any case the casing or connection must be sufficiently sturdy for the required duty. Sheet-metal work must be well braced not only to prevent vibration under pulsations in air flow, but also to withstand the abuse of normal usage. Casings should be braced wherever access doors are installed, and all large panels should be adequately reinforced by structural steel.

Accessibility for Service

Each apparatus layout should be made with accessibility in mind. Where cooling convectors are used, space for removing and repairing or replacing them should be provided. Adequate space should be provided for the servicing and replacement of eliminators. Filters must be so located that the proper cleaning, replacement, or routine servicing can be accomplished without difficulty. Free access to the bearings of all moving machinery is a necessity. Provision should be made for the replacement of any parts that are subject to wear, deterioration, or damage, such as filters, fanwheels, motors, pump impellers, or heat transfer surface.

CHAPTER 20

AIR DISTRIBUTION

Definitions, Standards for Satisfactory Conditions, Principles of Air Distribution Ventilating Jets in Air Distribution, Outlet Performance, Outlet Types, Outlet Location and Selection, Directional and Volume Control, Return and Exhaust Intakes, Specific Applications.

CORRECT air distribution is essential in warm air heating, ventilating, and air-conditioning systems. Even though a system delivers the required quantity of conditioned air to a room, unsatisfactory conditions result if the air is poorly distributed.

This chapter deals with room air distribution only; the transmission of air through ducts is treated in Chapter 21, Air Duct Design. The objectives of this chapter are to:

1. Present the principles of room air distribution with a discussion of ventilating jets in air distribution.
2. Present general information on practical air distribution problems involving outlet performance, outlet types, location, selection, and outlet control.
3. Illustrate the use of the basic concepts and of the practical experience gained in air distribution in making specific applications.

Further pertinent information on air distribution will be found in Chapter 6, Physiological Principles; Chapter 12, Heating Load; Chapter 13, Cooling Load; Chapter 15, Unit Ventilators and Unit Heaters; Chapter 16, Unitary Air Conditioning Equipment; Chapter 18, Warm Air Heating Systems; Chapter 19, Central Systems for Air Conditioning; Chapter 25, Sound Control; Chapter 44, Instruments and Measurements; Chapter 48, Transportation Air Conditioning.

DEFINITIONS

The following definitions referring to air distribution equipment have gained general acceptance.

1. *Supply Opening or Outlet:* Any opening through which air is delivered into a space which is being heated, cooled, humidified, dehumidified, or ventilated.
2. *Exhaust Opening or Return:* Any opening through which air is removed from a space which is being heated, cooled, humidified, dehumidified, or ventilated.
3. *Outdoor Air Opening:* Any opening used as an entry for air from outdoors.
4. *Damper:* A device used to vary the volume of air passing through a confined cross-section by varying the cross-sectional area.
5. *Grille:* A covering for any opening and through which air passes.
6. *Register:* A grille equipped with a damper or control valve.
7. *Free Area:* The total minimum area of the openings in the air outlet or inlet through which air can pass.
8. *Core Area:* The total plane area of that portion of a grille, included within lines tangent to the outer edges of the outer openings through which air can pass.
9. *Free Area Ratio:* The ratio of the free area to the core area.
10. *Coefficient of Discharge:* Ratio of area at vena contracta to area of opening.
11. *Aspect Ratio:* The ratio of length to width of an opening or core of a grille.
12. *Vane Ratio:* The ratio of depth of vane to minimum width between two adjacent vanes.
13. *Plaque:* A ceiling outlet in which the supply air impinges against a plate or series of parallel plates, and is deflected horizontally in all directions.
14. *Diffuser:* An outlet discharging supply air in various directions and planes.
15. *Primary Air:* The air delivered to the outlet by the supply duct.
16. *Induction:* The induction of room air drawn into an outlet by the primary air stream (commonly called *aspiration*).
17. *Entrainment:* The entrainment of room air by the air stream discharged from the outlet (commonly called *secondary air motion*).
18. *Total Air:* The mixture of discharged air and entrained air.
19. *Entrainment Ratio:* The total air divided by the discharged air.
20. *Throw (Blow):* The horizontal or vertical axial distance an air stream travels on leaving the outlet (grille) before the maximum velocity is reduced to 50 fpm.
21. *Drop:* The vertical distance, the lower edge of a horizontally projected air stream drops between the outlet and the end of its throw.
22. *Rise:* The converse of drop.
23. *Envelope:* The outer boundary of an air stream moving at a perceptible velocity.
24. *Spread:* The divergence of the air stream in a horizontal or vertical plane after it leaves the outlet.
25. *Diffusion:* Distribution of air within a space by an outlet discharging supply air in various directions and planes.
26. *Radius of Diffusion:* The horizontal distance from the diffuser outlet to the perimeter of the space, within which air motion in the occupied zone is reduced to 50 fpm maximum.
27. *Outlet Velocity:* The average velocity of air emerging from the outlet, measured in the plane of the opening.
28. *Terminal Velocity:* The average air stream velocity at the end of the throw.
29. *Temperature Differential:* Temperature difference between primary and room air.
30. *Temperature Variation:* Temperature difference between points of the same space.

STANDARDS FOR SATISFACTORY CONDITIONS

The object of air distribution in warm air heating, ventilating, and air-conditioning systems is to create in the occupied zone of the conditioned room (floor to 6 ft above floor level) the proper combination of temperature, humidity, and air motion. To obtain comfort conditions within this zone, standard limits have been established as acceptable *effective temperatures*. This term comprises air temperature, motion, humidity, and their physiological effect on the surface of the human body. Any variation from accepted standards of one of these elements may result in discomfort to the occupants. The same effect may be caused by lack of uniformity of conditions within the space or by excessive fluctuation of conditions in the same part of the space. Such discomfort may

arise due to excessive room air temperature variations (horizontally, vertically, or both), excessive air motion (draft), failure to deliver or distribute the air according to the load requirements at the different locations, or too rapid fluctuation of room temperature or air motion (gusts).

With reference to permissible room air motion it is not possible to establish a specific standard covering the entire complex problem of air distribution. Velocities less than 15 fpm generally cause a feeling of air stagnation, whereas velocities higher than 65 fpm may result in a sensation of draft. Air velocities of 25 to 35 fpm in the occupied zone are most satisfactory, but air motion of 20 to 50 fpm will usually be acceptable, with the lower values used in cooling applications, and the higher values in heating applications. The permissible air motion depends to some degree on the geographic location of the air-conditioning installation. In addition the noise level created by the introduction of supply air should be kept within acceptable limits, and streaking or smudging of walls or ceilings should be prevented.

Reference should be made to Chapter 6, Physiological Principles, for information on effective temperature and comfort lines, and to Chapter 25, Sound Control, for acceptable room noise levels and noise generated by air outlets. Material in Chapter 51, Control of the Industrial Environment, and Chapter 50, Process and Product Air Conditioning, deals with the health, safety, and efficiency of workers, and with temperature and humidity requirements for products and manufacturing processes, respectively.

PRINCIPLES OF AIR DISTRIBUTION

Conditioned air is supplied to air outlets at temperatures and velocities which differ greatly from those in the occupied zone of the room. Proper air distribution, therefore, calls for (1) entrainment of room air by the primary air stream outside of the zone of occupancy in order that air motion and temperature differences will be reduced to acceptable limits before the air enters the occupied zone, and (2) counteraction of the natural convection and radiation effects within the room.

The complexity of practical room air distribution problems is due to innumerable variations in building construction, system design, and operating requirements, and makes the exploration of the basic character of air distribution imperative. The theory of room air distribution is not complete but a considerable fund of knowledge supported by experimental guidance is available for the solution of many air distribution problems. In particular, research at the ASHAE Research Laboratory and at Case Institute of Technology have advanced considerably the fundamental knowledge on ventilating jets in air distribution. Also, a comprehensive survey of the literature relevant to air distribution problems was made at the ASHAE Research Laboratory and is an indispensable and timesaving tool for every researcher in air distribution. (Refer to Bibliography: ASHAE Research Laboratory.)

The cooperative research program in air distribution at Case Institute of Technology was started in 1937. The early research dealt with methods of measurements and the available instruments; next the basic flow patterns of free round and rectangular jets in an open space were studied. A third phase was concerned with diffusion rates and characteristics of free jets from the more complex types of outlets, such as grilles with guide vanes, narrow slots, perforated panels, plaques, and round ceiling diffusers. The latest phase has dealt with air movement produced by non-isothermal jets, especially in confined spaces, and has included experimental work in cold-wall rooms during heating and warm-wall rooms during cooling. (Refer to Bibliography: Cooperative Research at Case Institute of Technology.)

VENTILATING JETS IN AIR DISTRIBUTION

A surprising similarity between the shape of jets exists at a short distance from the outlet face, whether the outlet is round, rectangular, grille-like, or a perforated panel. The jet discharged from a round opening forms an expanding cone, whereas jets from rectangular outlets rapidly pass from rectangular to elliptical cross-sectional shape at a short distance from the outlet face, and then to circular shape, at a rate depending primarily on the aspect ratio. Even in the case of wide-angle grilles and angular outlets, the similarities are such as to permit the same analysis of performance.

For many conditions of jet discharge, therefore, it is possible to analyze jet performance, and to determine (1) the angle of divergence of the jet boundary, (2) the velocity patterns along the jet axis, (3) the velocity profile at any cross-section in the zone of maximum engineering importance, and (4) the entrainment ratios in the same zone.[1]

In using the data presented in this section, however, the following must be kept in mind:

1. The method of finding jet velocities is based upon several approximations and, therefore, the two recommended equations must be used with caution for extreme axial and radial distances.
2. The characteristics of the low-velocity regions of ventilating jets are not yet well understood. Neither for axial nor radial jets are the effects at various Reynolds number fully known.
3. The quantitative treatment of the forces that govern room air distribution problems has been limited, and non-isothermal conditions involving buoyant forces have not yet been fully explored.

For equations describing the performance of non-isothermal jets resulting from the air distribution research at Case Institute of Technology see References 2 and 3. (Refer also to Bibliography: ASHAE Cooperative Research at Kansas State College.)

4. Most investigations have been concerned with *free* jets, whereas air streams in practical room air distribution are not free streams but are influenced by walls, ceilings, floors, and obstructions. (Refer to sections on Effect of Walls and Ceilings and Room Air Motion.)

Angle of Divergence

The angle of divergence is very definite close to the outlet face, but the boundary contours are somewhat billowy and are easily affected by external influences. Here, as in air distribution generally, room air movement is replete with local eddies, vortices, and surges, which are manifestations of unbalance in the forces acting within the air stream. These internal forces govern the air motion, yet they are extremely delicate.[4]

Measured angles of divergence (spread) for discharge into large open spaces have usually ranged from 20 to 24 deg with an average of 22 deg. Coalescing jets for closely spaced multiple outlets expand at somewhat smaller angles, averaging 18 deg, and jets discharging into relatively small spaces show even smaller angles of expansion.[5] Tests indicate that in cases where the outlet area itself is small compared to the dimensions of the space normal to the jet, the jet may be considered free as long as

$$X \leq 1.5\sqrt{A_R}$$

where

X = distance from face of outlet, feet.
A_R = cross-sectional area of the confined space, square feet.

Four Zones in Jet Expansion

In analyzing the performance of jets, four major zones can be distinguished. They may be roughly defined in terms of the maximum or centerline velocity existing at the cross-section being considered:

Zone 1: A short zone, extending about 4 diameters or widths from the outlet face (or vena contracta for orifice discharge), in which the maximum velocity of the air stream remains practically unchanged.

Zone 2: A transition zone, extending to about 8 diameters for round outlets, or for rectangular outlets of small aspect ratio, over most of which maximum velocities vary inversely as the square root of the distance from the outlet. For rectangular outlets of large aspect ratio, this zone is elongated and extends from about 4 widths to a distance approximately equal to the width multiplied by four times the aspect ratio.

Zone 3: A long zone, of major engineering importance, in which the maximum velocity varies inversely as the distance from the outlet. This zone is often called the zone of fully *established turbulent flow* and may be 25 to 100 diameters long (or equivalent diameters of equal areas), depending on the shape and area of the outlet, the initial velocity, and the dimensions of the space into which the outlet discharges.

Zone 4: A terminal zone in which, in the case of confined spaces, the maximum velocity decreases at an increasing rate, or, in the case of large spaces free from wall effects, the maximum velocity decreases rapidly in a few diameters to the velocity range below 50 fpm which is usually regarded as *still air*.

Centerline Velocity in Zone 3

Research has shown that maximum or centerline velocities in Zone 3 of straight flow isothermal jets can be determined with good engineering accuracy from

$$\frac{V_x}{V_o} = \frac{KD_o}{X} = \frac{K'\sqrt{A_o}}{X} \qquad (1)$$

$$V_x = \frac{K'V_o\sqrt{A_o}}{X} = \frac{K'Q}{X\sqrt{A_o}} \qquad (2)$$

$$= \frac{K'Q}{X\sqrt{A_c \times C_d \times R_{fo}}} \qquad (2a)$$

Table 1. . . . Recommended Values of the Centerline Velocity Constant K or K'
(See Equation 1)

Type of Outlet	K		K'	
	V_o = 500 to 1000	V_o = 2000 to 10,000	V_o = 500 to 1000	V_o = 2000 to 10,000
Free Openings				
Round or Square	5.0	6.2	5.7	7.0
Rectangular, large aspect ratio (<40)	4.3	5.3	4.9	6.0
Annular slots axial or radial[a]	—	—	3.9	4.8
Grilles and Grids				
Free area 40% or more	4.1	5.0	4.7	5.7
Perforated Panels				
Free area 3 to 5%	2.7	3.3	3.0	3.7
Free area 10 to 20%	3.5	4.3	4.0	4.9

[a] For radial slots use X/H instead of $X/(A)^{1/2}$. H is the height or width of the slot.

Note: K and K' are indexes of loss in axial kinetic energy. Interpolate as required. Departures from maximum value indicate losses in first and second zones when compared with the jet from a rounded-entrance, circular nozzle.

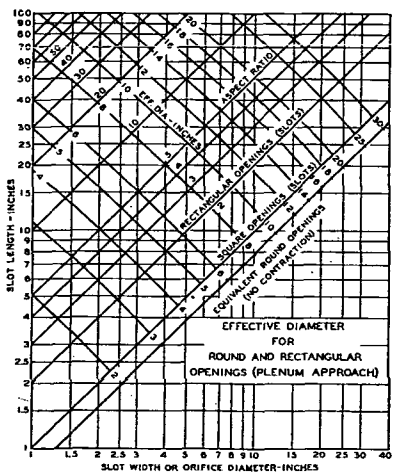


Fig. 1. . . . Effective Diameters for Round and Rectangular Openings (Plenum Approach)

where

V_x = centerline velocity, feet per minute.
$V_o = \frac{V_c}{C_d \times R_{fo}}$ = average initial velocity at discharge from open-end duct or across contracted stream at vena contracta of orifice or multiple-opening outlet, feet per minute.
V_c = nominal velocity of discharge based on the core area, feet per minute.
C_d = coefficient of discharge (usually between 0.65 and 0.90).
R_{fo} = ratio of free area to gross (core) area.
X = distance from face of outlet, feet.
K and K' = porportionality constants, with $K' = 1.13\ K$.
D_o = effective or equivalent diameter of stream at discharge from open-end duct or at a contracted section, feet.
$A_o = A_c \times C_d \times R_{fo}$ = effective area of stream at discharge from an open-end duct or at a contracted section, square feet.
A_c = measured gross (core) area of outlet, square feet.
Q = discharge from outlet, cubic feet per minute.

Equation 1 is nondimensional and requires only that consistent units be used, as in the above nomenclature. Values of K and K' are listed in Table 1.[1, 6]

Low velocity test results, in the range $V_x < 150$ fpm, indicate that the normal values of K and K' should be reduced about 20 percent for $V_x = 50$ fpm, as used in later Equation 4b for throw. Fig. 1 gives the effective diameter D_o in inches for single openings and includes the coefficient of discharge.[7] Similar values for grilles are available in the literature or can

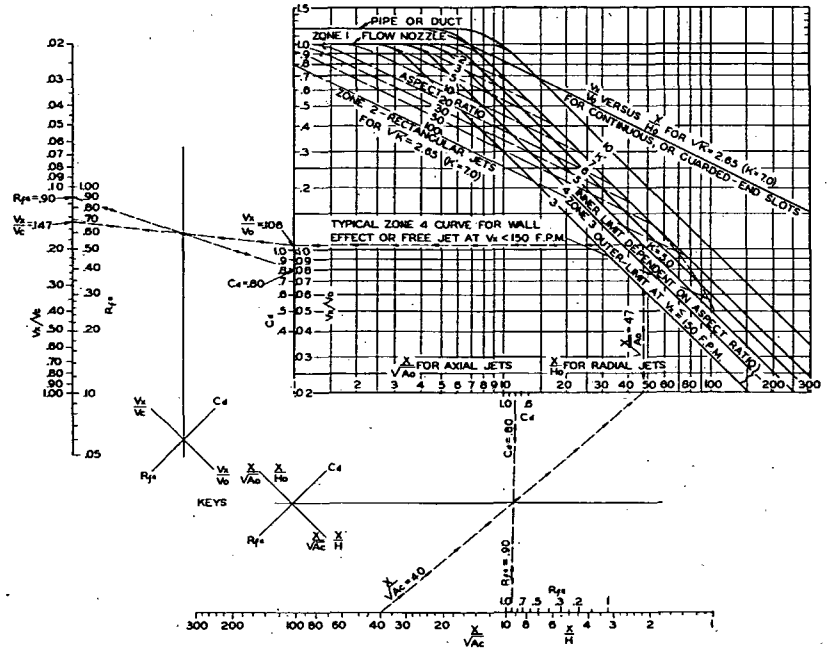


Fig. 2. . . . Chart for Determining Centerline Velocities of Axial and Radial Jets

be experimentally determined. Values of C_d and R_{fe} are required for determining V_0, D_0, and A_0 for orifice-type and multiple-opening outlets, and should, therefore, be included in catalog data of these outlets.

In the case of multiple-opening outlets and annular-ring outlets, the streams coalesce into a solid jet before actual jet expansion takes place. This coalescence affects the experimentally determined proportionality constants K or K' and accounts for some of the divergence in reported values for similar outlets.

Centerline Velocities in Zones 1 and 2

Experimental evidence indicates that in Zone 2

$$\frac{V_x}{V_0} = \sqrt{\frac{K'H_0}{X}} \qquad (3)$$

where

H_0 = width of jet at outlet or at vena contracta.

Approximately the same values of K' apply as for Zone 3 expansion from axial outlets.

In Zone 1, the ratio V_x/V_0 is constant and equal to the ratio of the center velocity of the jet at start of expansion to the average velocity, ranging from approximately 1.0 for rounded-entrance nozzles to about 1.2 for straight pipe discharge, but with much higher values for diverging-discharge outlets.

Determining Centerline Velocities

To permit correlation of data from all four zones, centerline velocity ratios are plotted against distance from outlet in Fig. 2 in accordance with the basic relation of Equation 2, and a nomogram for calculating the parameters

$$\frac{X}{A_0} \quad \text{and} \quad \frac{V_x}{V_c} \quad \text{from} \quad \frac{X}{\sqrt{A_c}} \quad \text{and} \quad \frac{V_x}{V_0}$$

through R_{fe} and C_d is given in the same illustration.

The variation of the centerline velocity ratio with distance from outlet, or more properly, from start of jet expansion, is also shown on Fig. 2 for Zones 1 and 2. V_x/V_0 is plotted against X/H_0 and, for a range of aspect ratios, against $X/\sqrt{A}$ for the single value of $K' = 7.0$. Values of V_x/V_0 for other values of K' may be obtained by direct proportioning of $\sqrt{K'}$ to $\sqrt{7.0}$.

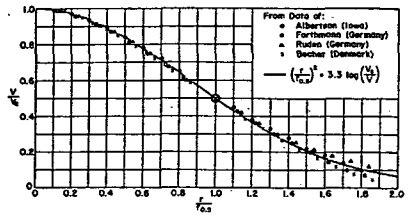


Fig. 3. . . . Cross-Sectional Velocity Profiles for Straightflow Turbulent Jets

The following *Example 1* which is solved on Fig. 2 will illustrate the use of the chart.

Example 1: A grille has a core area 12 in. x 18.75 in., $R_{fe} = 0.90$, $C_d = 0.80$, and $K' = 5.0$. Find V_x (velocity through core area) when V_x is 50 fpm for throw of 50 feet ($X = 50$).

Solution:

$$A_c = \frac{12 \times 18.75}{144} = 1.56 \text{ sq ft}$$

$$\frac{X}{\sqrt{A_c}} = \frac{50}{1.25} = 40$$

$$A_0 = 1.56 \times 0.80 \times 0.90 = 1.123$$

$$\frac{X}{\sqrt{A_0}} = \frac{50}{1.06} = 47.2$$

$$\frac{V_x}{V_0} = \frac{K'\sqrt{A_0}}{X} = \frac{5\sqrt{1.123}}{50} = 0.106$$

$$\frac{V_x}{V_c} = \frac{V_x}{V_0(C_d R_{fe})} = \frac{0.106}{0.80 \times 0.90} = 0.147$$

For $V_x = 50$,

$$V_c = \frac{50}{0.147} = 340 \text{ fpm.}$$

The quantity of air discharged is then,

$$Q = V_c A_c = 340 \times 1.56 = 530 \text{ cfm.}$$

Throw

Equation 2a can be used to determine the *throw* X of an outlet, if the discharge volume and the center velocity are known.

$$X = \frac{K'}{V_x} \cdot \frac{Q}{\sqrt{A_c \times C_d \times R_{fe}}} \qquad (4)$$

or, if

$$Z = \sqrt{C_d \times R_{fe}}$$

$$X = \frac{K'}{V_x} \cdot \frac{Q}{Z\sqrt{A_c}} \qquad (4a)$$

The maximum throw L is usually defined as the distance from the outlet face where the centerline velocity is 50 fpm. Therefore, for $V_x = 50$ fpm,

$$L = X = \frac{K'}{50} \cdot \frac{Q}{Z\sqrt{A_c}} \qquad (4b)$$

Velocity Profiles of Jets

In Zone 3 of both axial and radial jets, the velocity distribution may be expressed by a single curve (Fig. 3) in terms of dimensionless coordinates, and this same curve can be used as a good approximation for adjacent portions of Zones 2 and 4. Experiments have shown that temperature and density differences have but a small effect on cross-sectional velocity profiles.

Velocity distribution in Zone 3 can be expressed by the Gauss error-function or probability curve which is approximated by a simple equation using common logarithms

$$\left(\frac{r}{r_{0.5}}\right)^2 = 3.3 \log \frac{V_x}{V} \qquad (5)$$

where

r = the radial distance of the point under consideration from the centerline of the jet.

$r_{0.5}$ = the radial distance in the same cross-sectional plane from the axis to the point where the velocity is half the centerline velocity. ($V = 0.5\ V_x$.)

V_x = the centerline velocity in the same cross-sectional plane, feet per minute.

V = the actual velocity at the point being considered, feet per minute.

Experiments show that the conical angle for $0.5\ V_x$ and $r_{0.5}$ is approximately one-half of the total angle of divergence of a jet. The velocity profile curve for one-half of a straight-flow turbulent jet (the other half being a symmetrical duplicate) is shown in Fig. 3. For multiple-opening outlets, such as grilles, or perforated panels, the velocity profiles are similar, but the angles of divergence are smaller.

Radial Jets

In the radial jet the cross-sectional area at any distance from the outlet varies as the square of this distance, the same as for an axial jet. Experiments have shown that the centerline velocity gradients and the cross-sectional velocity profiles are similar to those of Zone 3 of axial jets and that the angles of divergence are about the same. In using Fig. 2, X/H should be used as abscissa instead of $X/\sqrt{A}$.

Jets from ceiling plaques have the same form as one-half of a free radial jet. The jet is wider and longer than a free jet, with the maximum velocity close to the wall. This is demonstrated in Fig. 4 which also indicates that under the conditions shown the width of the slot between ceiling and plaque has

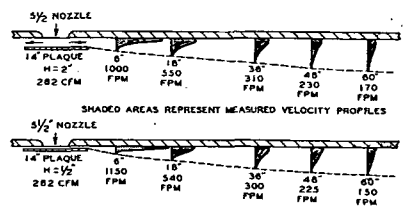


Fig. 4. . . . Air Jets from a 14-in. Ceiling Plaque for Two Slot Widths with Same Rate of Flow

little effect on the jet pattern or velocities at some distance from the plaque.[1]

For a basic equation for radial flow outlets resulting from the air distribution research at Case Institute of Technology see Jet Velocities from Radial Flow Outlets, by Alfred Koestel (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 505).

Discharge from a Long Slot

When a long slot receives its air supply from one end only, the important design factor is the ratio of the area of the slot to the area of the supply duct, and both the air stream profiles and the duct pressure requirements are determined by this ratio.[8] Fig. 5 shows the changing profile as the slot area is increased for a rounded entrance slot (C_d = 0.93) with constant duct cross-section.

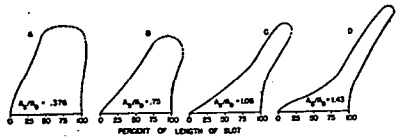


Fig. 5.... Shape of Air-Stream Envelopes as Slot Area is Increased

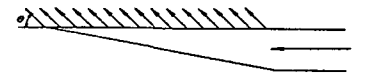
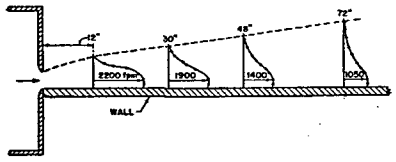


Fig. 6.... Uniform Air Flow from a Slot Supplied by a Tapered Duct



(V_o = 2260, A = 0.10, C_d = 0.78)

Fig. 7.... Jet Velocity Patterns for a Square Outlet Discharging along Wall or Ceiling

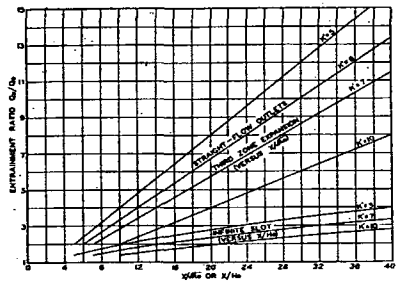


Fig. 8.... Entrainment Ratios for Straight-Flow Outlets

The air discharge from a slot in a tapered duct will be uniform (Fig. 6) for a relationship between discharge angle and slot-duct dimensions[8]

$$\cot \theta = \frac{A_s C_d}{A_d} \tag{6}$$

where

θ = discharge angle, degrees.
A_s = slot area, square feet.
A_d = duct cross-sectional area at upstream end, square feet.
C_d = coefficient of discharge.

Perforated Panels

When air is discharged from perforated panels of relatively large sizes the constant velocity core formed by the coalescence of the individual jets extends a considerable distance from the panel face. In this Zone 1 region the proportionality constants K and K' do not apply. Therefore, the proportionality constants given in Table 1 should be used only when the ratio (Distance from panel/$\sqrt{\text{Panel area}}$) is larger than 5. When the ratio is less than 5, the equation

$$V_x = V_o \cdot 1.2\sqrt{C_d \times R_f} \tag{7}$$

should be used for estimating centerline velocities.[12]

Effect of Walls and Ceilings

Jets discharging parallel to a wall with one edge of the outlet coinciding with the wall, take the form of one-half of an axial jet discharging from an outlet twice as large, similar to radial jets from ceiling plaques. Entrainment takes place practically only along the surface of a half cone and the maximum velocity remains close to the wall.[1] (See Fig. 7.) Values of K and K' are approximately those for a free jet multiplied by $\sqrt{2}$, that is to say, the normal maximum of 7.0 for K' for free jets becomes 9.9 for a similar jet adjacent to, and discharged parallel to, a wall, and in Fig. 2, $\frac{X}{\sqrt{2A_o}}$ should be used in place of $\frac{X}{\sqrt{A_o}}$

When a jet is discharged parallel to, but at some distance from a wall, its expansion in the direction of the wall is reduced and entrained air must be obtained by recirculation from the jet itself instead of from ambient air.[8, 11] The jet expands normally to within a short distance from the wall, which nearer to the outlet only affects velocity distribution in the outer shell of the jet, but further on also affects centerline velocities. This happens at some distance from a plane where the jet outline becomes parallel to the wall, and where the jet enters its fourth or terminal zone. Few engineering data are available on this important phase of jet expansion.[8]

Effect of Resistance on Return Path of Jet Air

Laboratory experiments on jets usually involve recirculated air with negligible resistance to flow on the return path of the jet air. Experience and experiments in tunnels of small cross-sectional area in mines, where considerable resistance is offered to the return flow of jet air to outlets, show that the expansion of the jet terminates abruptly at a distance that is independent of velocity of discharge and but slightly affected by size of outlet. Such distances are determined primarily by size and

length of return path. In a long 5 ft x 6 ft tunnel, a jet may not travel more than 25 ft whereas in a relatively short opening, 25 by 60 ft, it may travel more than 250 ft. Few engineering data are available on this phase of jet expansion but it can be of great practical importance.[8]

Entrainment Ratios

Equations for the entrainment of circular jets and of jets from long slots have been mathematically derived.[12] They are:

For third zone expansion of circular jets,

$$\frac{Q_x}{Q_o} = \frac{2}{K'} \cdot \frac{X}{\sqrt{A_o}} \tag{8}$$

For a long slot

$$\frac{Q_x}{Q_o} = \sqrt{\frac{2}{K'}} \cdot \sqrt{\frac{X}{H_o}} \tag{8a}$$

where

Q_x = total volume flow rate at distance X from face of outlet, cubic feet per minute.
Q_o = discharge from outlet, cubic feet per minute.
X = distance from face of outlet, feet.
K' = proportionality constant.
A_o = effective area of stream at discharge from an open-end duct or at a contracted section, square feet.
H_o = width of slot, feet.

Equations 8 and 8a have been used in plotting Fig. 8 for entrainment ratios.

OUTLET PERFORMANCE

The air distribution research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS has proved that the phenomena of room distribution are amenable to scientific research and rational interpretation, and has now reached a stage where a definite technique of experimentation in combination with semi-empirical theories can be of great help in the design of air outlets.

Practical room air distribution, however, is still predominantly an art, where decisions depend on isolated tests and judgments must be based on the accumulated experience of the behavior of outlets in the laboratory and in the field. Manufacturers' literature is available to provide data for the performance and application of various outlets. In designing a satisfactory air distribution system, the engineer must make use of this material, and weigh the different factors of the outlet performance which place considerable limitations on his proceedings. A short discussion follows regarding some of these factors which are: (1) jet pattern, (2) capacity, (3) temperature differential, (4) room air motion, (5) permissible noise level, and (6) smudging.

Jet Pattern

Refer to section Ventilating Jets in Air Distribution for a general discussion of the jet pattern. In selecting the *throw* and *drop* of outlets the following considerations are important: (1) throw, (2) effect of vanes, (3) effect of type of outlet, and (4) drop.

1. *Throw.* The throw of a wall outlet must be sufficient to produce satisfactory conditions over the area to be conditioned. Underblowing may cause heated air to rise too rapidly above the occupied zone and thus create excessive vertical temperature variation (stratification); in cooling operation it may cause cold air to drop into the occupied zone before supply and room air are mixed sufficiently, and thereby create a condition of acute discomfort (draft). On the other hand, overblowing will result in objectionable downdrafts from any surface the primary air stream may strike.

On the average, it is considered most practicable to select a throw which is three-fourths of the distance toward an exposed wall or window, as shown in diagram A of Fig. 9. However, structural characteristics, mounting height of outlet, temperature differential, and resultant drop or rise, or location of greatest heating or cooling loads strongly affect the selection of the optimum throw. In spaces with beamed ceilings, the outlets should be located below the bottom of the lowest beam level, and preferably low enough so that an upward or arched blow may be employed. The blow should be arched sufficiently to miss the beams and, at the same time, in such a manner as to prevent the primary or induced air stream from striking furniture and obstacles, and producing objectionable drafts.

2. *Effect of Vanes.* Vanes affect grille performance if their *depth corresponds at least to the distance between the vanes.* If the *vane ratio* is less than unity, effective control of the air stream discharged from the grille by means of the vanes is impossible. If it is higher than two, further improvement is slight.

A grille discharging air uniformly forward (vertical vanes in straight position), will have a *spread* of 14 to 24 degrees, depending on type of outlet, duct approach, and discharge velocity. Turning of the vanes will influence the direction and the throw of the discharged air stream.

A grille with *diverging vanes* (vertical vanes with uniformly increasing angular deflection to a maximum at each end of 45 deg) will have a spread of approximately 60 deg and the throw will be reduced considerably. With increasing divergence the quantity of air handled by the grille for a given duct *static pressure will decrease.*

A grille with *converging vanes* (vertical vanes with uniformly decreasing angular deflection) will have a slightly higher throw than a grille with straight vanes, but the spread will be the same for both settings. The air stream will converge slightly for a short distance in front of the outlet, and then spread somewhat more than the air discharged from a grille with straight vanes.

In addition to vertical vanes which normally spread the air horizontally, horizontal vanes may be used to spread the air vertically. However, spreading the air vertically entails the risk of hitting beams or other obstructions, or of blowing primary air at excessive velocities into the occupied zone.

3. *Effect of Type of Outlet.* Ceiling diffusers distribute the air with a horizontal spread of 360 deg and also have a downward air motion. Therefore, both throw (radius of diffusion) and mounting height are important and interdependent factors. Due to the 360 deg spread, the rate of induction is higher and the throw shorter than that for a wall grille handling the same air quantity at the same outlet velocity. Therefore, ceiling diffusers will frequently permit the use of higher air velocities than wall outlets, and consequently may be sized smaller to handle the same air volumes. If ceiling outlets are installed flush with the ceiling, impingement of the air stream along the ceiling surface restricts induction of secondary air, and the throw is increased approximately 40 percent above that of an unrestricted air stream. (Refer to Radial Jets and Effect of Walls and Ceilings in section Ventilating Jets in Air Distribution.)

Slotted outlets, due to their high aspect ratio, have higher induction than comparable round or square vaned outlets of

equal area handling the same air quantity, and their throw is shorter. (Refer to Discharge From a Long Slot in section Ventilating Jets in Air Distribution.)

The perforated panel is an excellent device for producing a large diameter stream with a uniform velocity across its entire area. Although this type of outlet can handle the greatest amount of air in proportion to room size, jet velocities must be kept low. (Refer to Perforated Panels in section Ventilating Jets in Air Distribution.)

For all types of ceiling outlets the following considerations should be noted: (1) if the temperature of the supply air is below that of the room, it should be thoroughly mixed with the room air before it enters the occupied zone; (2) air slightly above room temperature will usually be properly distributed by outlets selected for cooling; and (3) if the temperature of the supply air is substantially higher than that of the room, it should be projected downward in a broad diverging pattern to avoid stratification and to obtain proper mixing and controlled air flow.

4. *Drop.* The air discharged from a wall outlet should not reach the occupied zone until the velocity has fallen to about 50 fpm. Therefore, the outlets should be located high enough

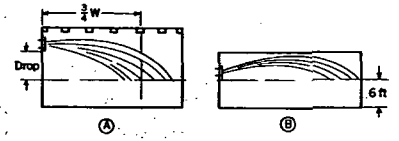


Fig. 9. . . . Throw of Wall Outlets

for the air stream at the termination of the throw to be not less than 5 or 6 ft above the floor level, or in other words, the drop should not be more than the difference between mounting height and zone of occupancy (see Diagram A of Fig. 9). As illustrated in Diagram B of Fig. 9 the maximum permissible throw for a given ceiling height may be obtained by locating the outlet low on the wall, arching the blow, and sweeping the air across the flat area. The air, as it traverses, will adhere to the ceiling. The objection to this method is the possible streaking of the ceiling with dirt.

Capacity

The quantity of air to be handled is determined by the heating, cooling, or ventilating requirements as given in Chapters 11 Infiltration and Ventilation, 12 Heating Load, and 13 Cooling Load. Manufacturers' rating sheets are usually consulted for selection of the proper number, size, and type of outlets for a given air quantity. The basis of rating should be carefully noted to make certain that resulting velocities are suitable for the application.

Temperature Differential

The temperature difference between supply and room air is highly important, because temperature control in the conditioned room is strongly affected by this difference. The greater the temperature differential, the greater the change in room temperature for a given change in the heating or cooling load. The use of outlets which cause a large amount of room air to be mixed with the supply air, permits the use of greater temperature differentials. (Refer to section Air Quantity and Temperature Differential, Chapter 19, Central Systems for Air Conditioning.)

Room Air Motion

To achieve a constant air motion in the occupied zone without exceeding acceptable velocity limits is one of the most important problems in air distribution. Some of the factors which may cause air motion to exceed these limits are: excessive air discharge velocities; high air volume per cubic foot of space (often referred to as number of air changes per hour); premature drop of cold air into the occupied zone; overblow causing spilling of high velocity air into the occupied zone; heating in severe climates by means of downward projection of hot air. These factors will not affect equally the various designs and types of air outlets, and their effect will also depend on the temperature differential, the mounting height of the outlet, and other circumstances. For instance, certain outlets will handle safely more air per cubic foot of space at higher discharge velocities than others, and downward projection of air may sometimes even be necessary if the supply air temperature is substantially higher than the room temperature. (Refer to section on Throw.)

Sometimes excessive air motion may be encountered in correctly designed air distribution systems due to drafts occurring, if stairways or exit doors are left open, or if windows are opened when design of system requires that they be closed. Discomfort is frequently attributed to drafts when it is actually due to too low room temperature. This may occur during the heating as well as the cooling season. Systems designed for heating and ventilation only may easily cause complaints of drafts when cooling is added.

The following papers give useful information with regard to room air motion: Room Air Distribution Research for Year 'Round Air Conditioning, Part I—Supply Outlets at One High Sidewall Location, by S. F. Gilman, H. E. Straub, A. E. Hershey, and R. B. Engdahl (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 151); Room Air Distribution Research For Year 'Round Air Conditioning, Part II—Supply Outlets at Three Floor Locations, by H. E. Straub and S. F. Gilman (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 249); Performance and Evaluation of Room Air Distribution Systems, by Alfred Koestel and G. L. Tuve (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 533); Air Velocities in Two Parallel Ventilating Jets, by Alfred Koestel and J. B. Austin, Jr. (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 425).

Permissible Noise Level

The increase of the noise level caused by an outlet is primarily a function of its discharge velocity and its size. The maximum acceptable noise level in a space may dictate completely the selection of the permissible outlet velocity. In addition, however, noise may be caused by excessive restriction of the free outlet area due to outlet design, by unnecessary turbulence due to one-sided air flow through the outlet, or by impingement of high velocity air on sharp edges.

The origin of excessive noise may be spotted by removing the outlet without stopping the air supply. If noise is still evident, other elements of the air-conditioning system or entirely different sources are at fault. High-pitched, rushing noise indicates that the duct system or excessive velocities in the duct are the cause, low-pitched rumbling noise points to the mechanical equipment of the air-conditioning plant.

When very high duct velocities are used, sound-absorbing lining or special acoustical filters (sound traps) must be

provided. High-pressure or high-velocity outlets for multi-room and other buildings are equipped with a special pressure reduction and sound attenuation box. (Refer to section Types of Air Outlets.)

For a general discussion of permissible room noise levels and air supply outlet noises, refer to Chapter 25, Sound Control.

Smudging

All air outlets have their particular *smudging* problems which have been studied by various investigators but cannot yet be considered solved. Smudging is due to dirt in the room air and in the air discharged by the outlets. Although the primary air may be carefully filtered, small particles of dirt and dust will not be captured by mechanical filters, and may finally be deposited on the walls or ceiling. With ceiling outlets, dirt streaking may be minimized by carefully controlling the discharge of the outlets, or by the use of special deflecting plates or anti-smudge rings. With wall outlets, dirt streaking may be minimized by preventing direct impingement of the air on any ceiling or room surface. Floor outlets may be objectionable as dirt collectors.

TYPES OF AIR OUTLETS

Air supply outlets for commercial, industrial, and residential applications are either side-wall- or ceiling-type outlets. In homes, baseboard and floor type outlets are also used. (Refer to Chapter 18, Warm Air Heating Systems.) As a great variety of outlets of various design are available in these four types, the final selection of a supply air outlet depends to a large degree upon the specific problems of the air distribution system to be used.

In addition to the comments which follow on use and application of outlets, reference should also be made to sections Outlet Location and Selection and Specific Applications.

Wall Outlets

Side wall outlets in general use are (1) perforated grilles, (2) fixed and adjustable bar and vane grilles, (3) registers, (4) wall diffusers, (5) slotted wall outlets, and (6) high-pressure or high-velocity wall outlets.

1. *Perforated Grilles.* These grilles have a small vane ratio and are not adjustable. They are useful primarily where directional air flow is unnecessary, and for applications where the grille size can be increased in order to provide the required free area.

2. *Fixed and Adjustable Bar and Vane Grilles.* Grilles with fixed bars or vanes are used for wall, floor, or baseboard applications when the performance is not critical or can be adequately predicted. Adjustable grilles are equipped with either vertical or horizontal adjustable bars or vanes or both, and are used where proper control of air motion is essential. Vanes should be designed for quiet operation.

3. *Registers.* Grilles equipped with dampers or adjustable valves for controlling volume distribution of the supply air are called registers.

4. *Wall Diffusers.* These outlets incorporate design features originally developed for ceiling outlets, and therefore make use of semi-conical or semi-pyramidal guide vanes instead of the straight vanes of the conventional side-wall outlet. In warm air perimeter systems they are used to blanket the outside wall with the warm supply air.

5. *Slotted Outlets.* These outlets are available in a great variety of designs, consisting either of elongated outlets with perforations, or flat outlets with a number of long narrow slots or a single narrow slot, or straight long outlets with one long slot or diffusing vanes. Due to their high aspect ratio, slotted outlets have higher entrainment than comparable round or square vaned outlets of equal area and consequently, the throw is reduced. (Refer to Discharge from a Long Slot in section Ventilating Jets in Air Distribution.)

Slotted outlets are particularly useful in connection with the linear design motif in architecture, for applications where it is desired to merge air outlets unobtrusively with the room decoration. They may be used for either ceiling or wall air distribution, including the so-called under-the-window application. In all applications, capacity limitations, proper duct approaches, and correct balancing of slotted outlets must be carefully considered, as correction after installation is difficult.

6. *High-Pressure or High-Velocity Outlets. Ejector nozzles* operate at high static pressure and convert this pressure into velocity pressure with resulting high induction of room air. They are mainly used for industrial process installations, particularly drying. Another type of ejector nozzle is sometimes referred to as a *louver nozzle* and has a 45 to 90 deg elbow, which can be rotated similarly to a universal joint about an axis perpendicular to the surface to which it is fastened. These outlets give a considerable degree of adjustability and are, therefore, useful for spotcooling in confined spaces.

High-pressure or high-velocity air-conditioning systems installed in multi-room and other buildings, use very high velocity and pressure so that the size of the ducts can be reduced. This in turn requires the use of *high-pressure or high-velocity air outlets* which reduce the velocity and pressure and attenuate the sound to predetermined values at the point where the air is discharged. It is also possible, of course, to use high velocity only in part of the system, for example in the main trunk ducts, and conventional velocities in the branch ducts. Special high-pressure control units are then installed at the end of the main duct, and the air is discharged through conventional air outlets in the various spaces of the building.

Various high-pressure or high-velocity outlets of different design, but usually consisting of a pressure reduction and sound attenuation box and a grille or diffuser, have been developed for high-velocity systems and are described in the literature of the manufacturers. These outlets may be used in the ceiling or in the wall of the air-conditioned space, including under-the-window installation, or in the ceilings of corridors. Refer to Unitary-Central Systems for multi-room buildings in Chapter 19, Central Systems for Air Conditioning, and to Design of High-Velocity Ducts in Chapter 21, Air Duct Design.

Ceiling Outlets

Ceiling outlets in general use are (1) ceiling plaques, (2) ceiling diffusers, (3) high-pressure or high-velocity ceiling outlets, and (4) perforated ceilings and panels.

1. *Ceiling Plaques.* Plaques are of simple design. The air from the supply opening impinges on a plate and is discharged horizontally. Plaques are inexpensive, but as the air is not always uniformly discharged in all directions, proper control is difficult.

2. *Ceiling Diffusers.* Ceiling diffusers are round, square, rectangular, or slotted outlets of various designs installed on, or parallel to, the ceiling. Performance of the different designs varies according to the principle employed. Some

have no internal induction, but hasten external induction by supplying air in multiple layers. Others have internal induction and distribute air over an entire hemisphere. The induction effect is greatest in the direction of the axis of the outlet, and least in the plane perpendicular to the axis and located at the ceiling level. Thus the induction is greatest in the vertical direction where the least throw can be tolerated, and least in the horizontal plane at the ceiling where the greatest throw is desired. (Refer to section *Slotted Outlets* under Wall Outlets.)

3. *High-pressure or High-velocity Outlets.* Refer to text in section Wall Outlets.

4. *Perforated Ceilings and Perforated Panels.* These devices discharge air through perforations in the ceiling or part of the ceiling, and share with slotted outlets the advantage of unobtrusive appearance. In addition, sound-absorbing materials may be used for the panels, or the panels may provide a luminous ceiling.

Outlet air velocities may be kept low by using large outlet areas, particularly in spaces with high room loads, high ventilating requirements, or low ceilings. Some perforated panels have a control plate frame which is inserted in the conventional ceiling duct. Supply air enters the plenum above the distribution plates through an adjustable air valve which can be set for varying air quantities and velocities. (Refer to Perforated Panels in section Ventilating Jets in Air Distribution.)

OUTLET LOCATION AND SELECTION

The design of the air-conditioning plant for a building depends on the use to which the building is put, on its size, and its construction type. The design of the air transmission and air distribution system is influenced by the same factors. Therefore, these factors must be considered in designing the air transmission system, and in selecting the type and location of the supply outlets. The location and selection of the supply outlets is further influenced by the interior design of the building, local sources of heat gain or loss, and outlet performance and design.

Use, Size, Construction of Building

The use, size, and construction of a building are three factors which may influence or even govern the selection of a particular type of air distribution system, as is shown by the development of special high-pressure or high-velocity transmission and distribution systems for multi-room and other buildings. Generally, the physical details of the individual rooms of a building, such as height and shape, and the location of beams and other obstructions, influence the possible location of the supply ducts which in turn influences the outlet locations. The location of the supply ducts—above the ceiling or within the walls, in a furred space above corridors, or in the conditioned room itself—has an important bearing, therefore, upon the design of the air distribution system. A particular method of air distribution may be highly desirable but it may be impossible to use it, due to the location of beams and masonry walls.

Refer to later section Specific Applications for a discussion of air distribution methods in various buildings.

Heat Gain or Loss

Local sources of heat gain or loss promote convection currents or cause stratification, and may, therefore, determine both the type and location of the supply outlets.

Generally, the outlets should be located to neutralize any undesirable convection currents set up by a concentrated heating load. If a concentrated source of heat is located at the occupancy level of the room, the heating effect may be counteracted by blowing the supply air toward the heat source, or by locating an exhaust or return grille adjacent to the heat source. The second method is more economical, as heat will be withdrawn at its source rather than be dissipated into the conditioned space. Where lighting loads are heavy and ceilings relatively high (five watts per sq ft and above 15 ft), the outlets should be located below the lighting load, and the stratified warm air removed by an exhaust or return fan. An exhaust fan is recommended if the wet-bulb temperature of the air is above that of the outdoors, a return fan, if it is below this temperature. Either method reduces the requirements for supply air. If the lights are exposed, less saving can be realized than if they are enclosed, as a considerable portion of the energy is radiant.

An important function of any supply outlet during *winter heating* is to maintain the temperature difference between the floor and ceiling at the lowest possible value. Since the air within a room is never so perfectly mixed that all of it is at exactly the same temperature, a certain amount of stratification is inevitable. Unless the supply outlets are properly selected and located, the temperature difference will be large enough to result in discomfort to the occupants.

The proper selection and location of outlets is particularly difficult for rooms with large windows and exposed wall areas. Heat is lost primarily through these areas, and room air in contact with these surfaces is chilled and flows downward, causing drafts near the exposed surfaces and floor.

This condition can best be prevented by locating the air outlets under the windows, by discharging warm air across the exposed surfaces, or by providing supplementary radiation under the windows, in addition to the wall or ceiling air distribution outlets. This solution must be used for comfort installations in northern latitudes (outdoor temperatures below 40 F), when the building walls are uninsulated or single-glass windows are used.

In buildings located in regions where winter heating is a minor problem, high induction by ceiling or wall outlets may be employed to mix properly the entire room air with the supply air and to prevent downdrafts along the exposed surfaces. In northern latitudes, this solution can only be used if the buildings have double-glass windows and insulated walls, as otherwise the induction effort required for neutralization of the downdrafts is so great that the air motion in the room exceeds comfort limits. Where comfort conditions are not critical as in factories for heavy maunfacturing, warehouses, etc., satisfactory results with high induction outlets can be obtained even in cold climates. For uninsulated walls and glass areas in these buildings some supplementary heating is often valuable. Wall diffusers, direct radiation, or warm panels will satisfy these requirements for supplementary heating.

The location of exhaust or recirculating air openings at the base of large areas of glass is sometimes effective in reducing downdrafts into the occupied space.

Great care should be exercised in the use of supply-exhaust air outlets for heating in northern latitudes when considerable exposed wall and window surfaces are involved. In such cases additional radiation is usually required. When the system is used for ventilating only during the heating season, direct radiation should be regulated to provide heat until room and outside temperature are balanced. To avoid

stratification, it is advisable to operate the fan continuously and to limit the temperature differential by supplying air at a high rate of change per hour.

Outlet Performance and Design

Outlets should perform efficiently and conform to the esthetic appearance of the room. The physical appearance of air outlets has been highly influenced by modern interior design, and to meet the demands of architects and engineers. Outlets to be installed flush with the ceiling and outlets with built-in lighting as well as square and linear-type outlets have been developed in the past years. Linear-type outlets and perforated ceilings or panels have the advantage of unobtrusive appearance.

Procedure for Outlet Location and Selection

In determining outlet location and selecting the type of outlets, it is customary to proceed as follows:

1. Study the plan of the building and note the amount of air to be supplied to each room. (For computation of the *required air quantity* for heating and cooling refer to the formulas given in Chapters 12, Heating Load, and 13, Cooling Load. The amount of outdoor air introduced into the room must be checked against the ventilation requirements of state or local codes.)

2. Select number of outlets for each room, considering air quantity required and distance available for throw or as radius of diffusion. The same factors, as well as distance from floor level available as mounting height, structural characteristics of the space, and consideration of appearance, will determine the type of outlet used.

3. Arrange location of outlets in the room. Generally, the outlets must be evenly spaced to distribute the air uniformly throughout the room. More air, however, must be supplied and directed towards areas with exceptional heat gain or loss. An important point to consider is the combination of proper outlet location and efficient duct design (see Chapter 21). Consult manufacturers' tables for recommended location and spacing of outlets.

Refer to Chapter 19 for a discussion of zone control.

4. Select size of outlets according to air quantities handled, permissible throat or discharge velocities, or effective throw, taking into consideration other factors, such as permissible noise level, etc. In order to determine whether the selection made will satisfy the requirements of the job, the type, size, and location of the outlet must be checked against manufacturers' ratings. The most important questions to be considered are:

   a. Will drafts be created because of divergence between rated throw (radius of diffusion) and distance between outlet and nearest obstacle (wall, beam, column, etc.) in the air stream.
   b. Will drafts be created because of excessive cooling temperature differential or too low mounting height of the outlet?
   c. Will drafts created because of too low velocity energy of the air stream cause excessive downward air flow in cooling installations?
   d. Will the air pass through the outlet at too high a velocity and thereby cause an excessive increase in noise level?

Balancing the System

In designing a system, the engineer aims to size ducts and outlets in such a manner that the supply of air is properly distributed. He may, however, feel it necessary to oversize certain trunks, branches, or outlets to allow flexibility or to permit future redistribution, so that the system as designed may not be self-balancing. Even if ducts are designed for self-balancing and outlets are properly selected, all air systems must be balanced, that is, the amount of air through supply ducts, supply outlets, outdoor air intakes, exhausts, return outlets, and ducts must be properly adjusted so that the air quantities correspond closely with the design quantities. Balancing, therefore, is part of the field test procedure to which each air-conditioning system should be subjected in order to determine whether the capacity and performance of the equipment correspond with the design.

For balancing, air systems may be equipped with dampers in supply, return, and outdoor air ducts, splitter dampers, or dampers in outlets. In selecting the desired type of damper or balancing method, the following points should be kept in mind:

1. Unfavorable effect on air stream and noise level should be avoided. This will often rule out blank-offs or dampers in outlets, unless such dampers are of special design.

2. It should be possible to alter outlet volume without altering the direction of discharged air, and to measure the amount of air handled without difficulty. Blanking off part of the area of supply outlets makes this difficult.

Generally speaking, it is most satisfactory to install dampers at some distance back of outlets so as to avoid disturbing the air flow. Dampers in both supply and return ducts form the most flexible means of controlling supply of air to the room and static pressure within the room. (Refer to following section Directional and Volume Control for discussion of various air distribution control devices.)

Instruments for balancing include instruments for measuring air quantities, static pressure, temperature and humidity. Commonly used instruments are the rotating vane anemometer, deflecting vane anemometer, thermal anemometer, smoke gun, clamp type ammeter-voltmeter, sling psychrometer and thermometers. Pressure, humidity and temperature recorders are sometimes useful. The Pitot tube and a manometer may be used on higher velocities, but only with the technique prescribed for its use by the manufacturer for the tube used. (See Air Flow Measurement in Chapter 44.)

The following balancing procedure is recommended for air systems:

1. Open all duct and outlet dampers.
2. Measure fan capacity. This can most often be done with a rotating vane anemometer inside the fan inlet plenum. For exhaust fans the discharge outlet will often serve.
3. Measure fan motor input.
4. Determine approximate flow at each outlet.
5. Adjust outlets or branches so that outlet deviation (from design quantity) is approximately the same (percentagewise) as fan capacity deviation (from total design quantity).
6. Recheck fan capacity.
7. Adjust fan speed or main trunk damper to make fan capacity equal to design. (The fan laws apply when changing speeds; quantity varies directly with speed,—power input as the cube of speed.) Use power input readings to check possibility of overloading the motor.

The procedure outlined may be shortened depending on the type of installation. For instance, where many outlets serve one room, exact balancing of each outlet may not be required, so long as drafts, noise and temperature differences are checked. Experience and judgment are valuable in determining how carefully or thoroughly a system must be tested and adjusted.

DIRECTIONAL AND VOLUME CONTROL

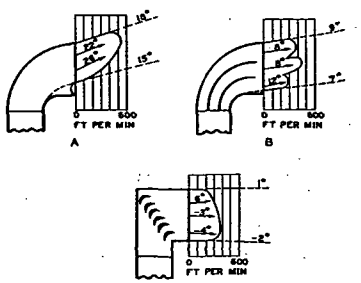
Duct Approaches to Outlets

The importance of the manner in which the air stream is introduced into the outlet cannot be overestimated. To obtain correct air distribution, the velocity of the air stream must be as uniform as possible over the entire connection to

the duct, and perpendicular to the outlet face. No air outlet can compensate for improper duct approach.

A wall grille installed at the end of a horizontal duct and a ceiling outlet at the end of a vertical duct receive the air perpendicularly and at (for practical purposes) uniform velocity over the entire duct cross-section, provided the system is carefully designed. However, very few outlets are installed in this manner. Most side wall outlets are installed either at the end of vertical ducts or in the side of horizontal ducts, and most ceiling outlets are attached either directly to the bottom of horizontal ducts or to special vertical take-off ducts which connect the outlet with the horizontal duct. In all these cases, special devices for directing and equalizing the air flow are necessary to obtain proper direction and distribution of the air.

The influence of the duct approach on outlet performance has been investigated for vertical stack heads with plain



Stack 14 in. x 6 in. Outlets 14 in. x 9 in. Stack Velocity 500 fpm
A. Rounded Throat and Rounded Back
B. Rounded Throat and Back and 2 Splitters.
C. Square Throat and Back and 6 Guide Vanes.

Fig. 10. . . . Outlet Velocity and Air Direction Diagrams for Stack Heads with Expanding Outlets

openings[13] or equipped with grilles[14] and for side outlets on horizontal ducts.[14] In the tests conducted with the stack heads it was found that it is necessary to provide splitters or guide vanes in the elbows at the top of the vertical stacks regardless of the shape of the elbows, whether of rounded, square or expanding types. Cushion chambers at the top of the stack heads have no beneficial effect. Fig. 10 shows the direction of flow, distribution and velocity (measured 12 in. from opening) of the air for various types of stack heads tested, expanding from a 14 x 6 in. stack to a 14 x 9 in. opening, without grille. The air velocity for each was 500 fpm in the stack below the elbow, but the direction of flow and the distribution pattern are generally indicative of performance obtainable with nonexpanding elbows of similar shapes for a range of velocities 200 to 1400 fpm.

In tests conducted with 3 x 10 in., 4 x 9 in., and 6 x 6 in. side outlets in a 6 x 20 in. horizontal duct at duct velocities of 200 to 1400 fpm (in the 6 x 20 in. section) it was found that multiple curved deflectors produced the best flow characteristics. Vertical guide strips in the outlet were not so effective as curved deflectors. A single scoop type deflector at the outlet did not improve the flow pattern obtained from a plain outlet, and was therefore not found to be desirable.

Directional Control

Many devices for directing and equalizing air flow in side wall and ceiling outlets are now commercially available, and should be used whenever necessary. They are indispensable for all side outlets in horizontal ducts and for most ceiling outlets.

Ceiling outlets are usually installed below horizontal supply ducts so that the supply air has to make a 90-deg turn before entering the outlet itself. The shorter the connection between bottom of duct and outlet, the greater is the need for directive devices to obtain uniformity of flow. Generally speaking, conditions and remedy in such cases strongly resemble those for side outlets in horizontal air ducts. Ceiling ducts often have a rectangular cross-section, while the connections to the ceiling outlets are circular. It will then be quite difficult to install turning vanes successfully, particularly if the ducts are shallow and the connection areas are comparatively large. This will be the case when more than one outlet is installed on one duct run, and restrictions of duct areas must be avoided. In such cases good results have been obtained by using a series of vertical guide strips, installed at right angles to the direction of air approach in the outlet connection where it leaves the horizontal air duct.

Volume Control

Various methods are used to regulate volume of supply and return (exhaust) air. Some of these accomplish only minor changes in volume; most of them, however, permit a range of adjustment from maximum air supply to complete shut-off.

When selecting type and location of such dampers, the following points must be considered, especially when the volume control feature is to be located near the air outlet itself: (1) deflection of air stream by the damper; (2) need and feasibility of directional control; (3) increase of noise level due to irregular and localized high air velocities caused by damper operation.

The following types of volume control are most frequently encountered:

1. *Slide Damper.* A single plate which can be pushed across the duct. Since its operation changes the free area of air passage in a one-sided manner, it should not be located near any air outlet, and its use is practicable only where no intermediate setting between full open and closed is required.

2. *Hit-and-Miss Damper.* Two slotted plates or discs, closely adjacent; by moving one of the two plates the respective slots may be opened or closed. This type of volume control may be installed close to an air outlet and it is easy to operate, but its main disadvantage is that even in the open position the air passage area is blocked by at least 50 percent. This requires oversizing of the air outlet in order to avoid excessive increase of noise level.

3. *Splitter Damper.* A single blade sheet metal plate hinged at one edge, usually located at the branch connection of a duct or outlet. It is easy to operate, but often causes irregular air flow in the duct. When used in connection with, and near an outlet, additional directional control is required.

4. *Butterfly Damper.* A single blade sheet metal plate hinged in the middle, usually located in a straight duct run.

Table 2. . . . Recommended Return Intake Face Velocities

Intake Location	Velocity Over Gross Area Fpm
Above occupied zone	800 up
Within occupied zone, not near seats	600–800
Within occupied zone, near seats	400–600
Door or wall louvers	200–300
Undercutting of doors (through undercut area)	200–300

It is easier to handle than a splitter damper, since only half the motion is necessary to change its setting. However, if located too close to an air outlet, it is objectionable because its operation frequently results in a condition whereby two high velocity jets are created along the sides of the duct, or the air spills immediately downward into the occupied zone.

5. *Louver Dampers.* Numerous designs have been developed incorporating a series of splitter or butterfly dampers across the duct or air outlet. Their main advantage consists in retaining greater uniformity of air flow, and in requiring less depth for installation. Some designs provide for louver blades moving in opposite directions, and while decreasing free air passage area, retain a constant air flow direction along the axis of the duct air outlet connection.

RETURN AND EXHAUST INTAKES

The *selection of return and exhaust intakes* depends on: (1) velocity in occupied zone near intake; (2) permissible pressure drop through intake; and (3) noise.

1. *Velocity.* The control of the room air motion for the maintenance of comfort conditions depends on the proper selection of the *supply* outlets. The effect of air flow through return intakes upon air movement in the room is slight. Air handled by the intake approaches the opening from all directions and its velocity decreases rapidly as the distance from the opening increases. Therefore, drafty conditions will rarely be encountered near return intakes. Recommended return intake face velocities are given in Table 2.

2. *Permissible Pressure Drop.* The permissible pressure drop will depend on the choice of the designer. Table 3 gives pressure drop through plain lattice intakes as a function of free area and face velocity.

Proper pressure drop allowance should be made for control or directive devices.

3. *Noise.* The problem of *noise* generated by return intakes is the same as that for supply outlets. In computing resultant room noise levels from the operation of an air-conditioning system, the return intake must be included as a part of the total grille area. The major difference between the supply outlets and return intakes is the frequent installation of the latter at ear level. When so located, it is recommended that the return intake velocity be not in excess of 75 percent of the maximum permissible outlet velocity.

The *location of return and exhaust intakes* does not critically affect air motion, unless room air velocities in the occupied zone near the intake exceed comfort limits. The locations of return or exhaust intakes are, however, important for obtaining the desired room temperature equalization.

Ceiling locations for exhaust outlets are recommended for

Table 3. . . . Approximate Pressure Drops for Lattice Return Intakes
Inches Water Gage—Standard Air

Percent Free Area	Face Velocity, Fpm						
	400	500	600	700	800	900	1000
50	0.06	0.09	0.13	0.17	0.22	0.28	0.35
60	0.04	0.06	0.09	0.12	0.16	0.20	0.24
70	0.03	0.05	0.07	0.09	0.12	0.15	0.18
80	0.02	0.03	0.05	0.07	0.09	0.11	0.14

bars, kitchens, lavatories, dining rooms, club rooms, etc. where warm air will rise to the ceiling level. In heating installations, location of the return grilles in the ceiling or high on the wall is not recommended, as it may result in stratification of the conditioned air, and—depending on the relative location of supply and return outlets—in short-circuiting. (Refer to previous section Outlet Location and Selection.)

Some ceiling outlets combine the supply and return openings in a single unit. This method is used for heating as well as for cooling applications. However, the application for heating is more critical and requires consideration of ceiling height, amount of outside wall area, and number of air changes required. In some cases, stratification of warm air may cause short-circuiting. (Refer to previous section Outlet Location and Selection.)

Floor locations of returns are used in heating installations for ceiling or side-wall supply. When located so that air is drawn across exposed walls, the performance of the system may be somewhat improved. In general, floor locations tend *to collect dirt and refuse.*

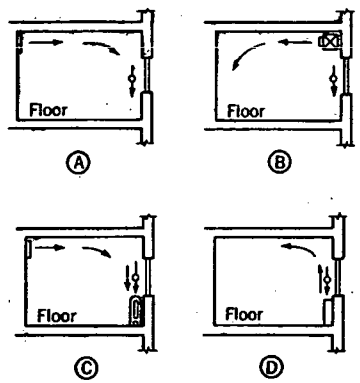
Wall and door locations of exhaust outlets depending on their elevation, have the characteristics of either floor or ceiling returns. In large buildings with many small rooms, the return air may be brought through door grilles or door undercuts into the corridors, and then to a common return or exhaust. The pressure drop through door returns should not be excessive; otherwise the air distribution to the room may be seriously unbalanced with the opening or closing of the doors. Outward leakage through doors or windows cannot be counted upon for dependable results.

SPECIFIC APPLICATIONS

For *theaters and auditoriums* the air distribution methods used are the downward distribution system with ceiling diffusers, and the horizontal distribution system with ejector nozzles or wall diffusers. Fig. 11 shows both methods. Ceiling distribution is accomplished by ceiling outlets under main ceiling and balcony. It is indicated when main ceiling or



Fig. 11. . . . Air Distribution Methods for Theaters, Churches, and Auditoriums



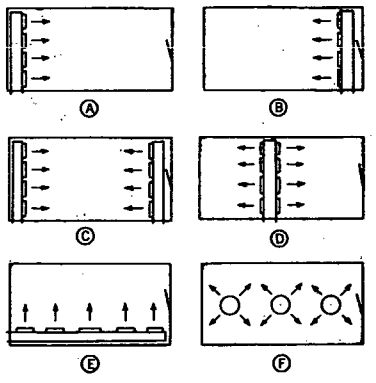
A. Satisfactory for cooling. Unsatisfactory for heating in severe climates where the outside temperature is consistently below 40 F, and single glass and uninsulated walls are prevalent.

B. Performance approximately that of A when small diffusers are used in bottom of the duct.

C. Satisfactory for cooling. Satisfactory for heating if direct radiation is properly controlled.

D. Satisfactory for both cooling and heating. The air should be discharged slightly away from the wall and for low velocities, should be fanned out parallel to the wall.

Fig. 12.... Distribution Methods for Small Rooms



A. *Rear Wall.* High outlet velocity, satisfactory if properly designed; possibility of excessive air motion and drafts if used for wrong application.

B. *Front Wall.* High outlet velocity, results same as A.

C. *Front and Rear Walls.* Moderate room air motion, outlet blows should not impinge giving rise to downdrafts in center.

D. *Center.* Moderate air motion, no impingement of air streams. Good results.

E. *One Side.* Moderate room air motion; should blow toward exposed wall. Good results.

F. *Ceiling.* Low room air motion. Good results. Outlets should be selected of sufficient size to allow for blocking when not located in perfect squares.

Fig. 13.... Small Store Cooling Distribution

balcony is cut up by architectural treatment or beams. The only critical points are under the balcony, and (occasionally) above the very rear of the balcony, where ceiling heights are low and where direct impingement of air is sometimes a hazard.

Wall or ejector distribution is particularly applicable for relatively long and narrow theaters. It is essential with this type of distribution that there be no interference with the movement of air throughout its entire path from the high velocity nozzles to the front of the theater. The ceiling should be smooth, without projecting beams or obstructing ornamentation. For large theaters, relatively high velocities can be used. These will work satisfactorily if adjustable outlets are used to avoid areas of local turbulence.

In small or medium size theaters, it is sometimes practicable to use side-wall or front-wall distribution. For the satisfactory operation of such a system during the winter heating period, the returns should preferably be located at the floor level and near the front of the theater to prevent cold spots resulting from exposed wall convection or infiltration from exits.

For *multi-room buildings* diagrams shown in Fig. 12 illustrate distribution methods for small rooms with exposed wall, such as offices, hotel (guest) rooms, hospital (patients) rooms, apartments, etc.

For *a small store* the cooling performance of various distribution methods is illustrated in Fig. 13.

REFERENCES

[1] G. L. Tuve: ASHVE Research Report No. 1476—Air velocities in ventilating jets (ASHVE Transactions, Vol. 59, 1953, p. 261).

[2] Alfred Koestel: ASHVE Research Report No. 1512—Computing temperatures and velocities in vertical jets of hot or cold air (ASHVE Transactions, Vol. 60, 1954, p. 385).

[3] Alfred Koestel: ASHAE Research Report No. 1534—Paths of horizontally projected heated and chilled air jets (ASHAE Transactions, Vol. 61, 1955, p. 213).

[4] H. B. Nottage, J. G. Slaby, and W. P. Gojsza: ASHVE Research Report No. 1458—Outlet turbulence intensity as a factor in isothermal-jet flow (ASHVE Transactions, Vol. 58, 1952, p. 343).

[5] G. E. McElroy: *Air Flow at Discharge of Fan-Pipe Lines in Mines (U. S. Bureau of Mines Report of Investigations R. I.* 3730, November 1943, p. 19).

[6] Alfred Koestel, Philip Hermann, and G. L. Tuve: ASHVE Research Report No. 1404—Comparative study of ventilating jets from various types of outlets (ASHVE Transactions, Vol. 56, 1950, p. 459).

[7] R. D. Madison and W. R. Elliot: Throw of air from slots and jets (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, November 1946, p. 108).

[8] Alfred Koestel and G. L. Tuve: ASHVE Research Report No. 1328—The discharge of air from a long slot (ASHVE Transactions, Vol. 54, 1948, p. 87).

[9] Alfred Koestel and C. Y. Young: ASHVE Research Report No. 1429—The control of air streams from a long slot (ASHVE Transactions, Vol. 57, 1951, p. 407).

[10] Alfred Koestel, Philip Hermann, and G. L. Tuve: ASHVE Research Report No. 1366—Air streams from perforated panels (ASHVE Transactions, Vol. 55, 1949, p. 283).

[11] H. B. Nottage, J. G. Slaby, and W. P. Gojsza: ASHVE Research Report No. 1443—Isothermal ventilation—jet fundamentals (ASHVE Transactions, Vol. 58, 1952, p. 107).

[12] Entrainment in Ventilating Jets (Case Institute of Technology, unpublished paper).

[13] D. W. Nelson, D. H. Krans, and A. F. Tuthill: ASHVE Research Report No. 1155—The performance of stack heads (ASHVE Transactions, Vol. 46, 1940, p. 205).

[14] D. W. Nelson, D. H. Lamb, and G. E. Smedberg: ASHVE Research Report No. 1206—Performance of stack heads equipped with grilles (ASHVE Transactions, Vol. 48, 1942, p. 279).

[15] D. W. Nelson and G. E. Smedberg: ASHVE Research Report No. 1226—Performance of side outlets on horizontal ducts (ASHVE Transactions, Vol. 49, 1943, p. 58).

BIBLIOGRAPHY

Note: This guide to the literature on air distribution is purposely restricted to papers resulting from research sponsored by The American Society of Heating and Air-Conditioning Engineers and other easily accessible material.

ASHAE Research Laboratory

Cyril Tasker: ASHVE research in air distribution and air duct friction (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, April 1948, p. 125).

H. B. Nottage: ASHVE Research Report—Ventilation jets in air distribution (See Reference 1). The following papers based on this report have been published:

H. B. Nottage: ASHVE Research Report No. 1360—Turbulence—a fundamental frontier in air distribution (ASHVE Transactions, Vol. 55, 1949, p. 193).

H. B. Nottage: ASHVE Research Report No. 1402—A simple heated-thermocouple anemometer (ASHVE Transactions, Vol. 56, 1950, p. 431).

H. B. Nottage, J. G. Slaby, and W. P. Gojsza: ASHVE Research Report No. 1443—Isothermal ventilation-jet fundamentals (ASHVE Transactions, Vol. 58, 1952, p. 107).

H. B. Nottage, J. G. Slaby, and W. P. Gojsza: ASHVE Research Report No. 1441—A V-wire direction probe (ASHVE Transaction, Vol. 58, 1952, p. 79).

H. B. Nottage, J. G. Slaby, and W. P. Gojsza: ASHVE Research Report No. 1458—Outlet turbulence intensity as a factor in isothermal-jet flow (ASHVE Transactions, Vol. 58, 1952, p. 343).

H. B. Nottage, J. G. Slaby, and W. P. Gojsza: ASHVE Research Report No. 1461—A smoke-filament technique for experimental research in room air distribution (ASHVE Transactions, Vol. 58, 1952, p. 399).

H. B. Nottage, J. G. Slaby, and W. P. Gojsza: ASHVE Research Report No. 1459—Exploration of a chilled jet (ASHVE Transactions, Vol. 58, 1952, p. 357).

H. B. Nottage: *Ventilation Jets in Room Air Distribution.* Originally submitted as a Ph.D. thesis to Case Institute of Technology (3 volumes). Loan copies available from ASHAE Research Laboratory.

Volume I: *Data and Interpretations of Tests for Ventilation Jets Discharging Horizontally into a Large Confining Space* (131 pages + 67 figs. + 9 tables).

Volume II: Appendix A. *Survey of the Literature on the Problems of Ventilation Jets and the Related Fundamentals of Turbulent Flow* (499 pages incl. figs. and tables).

Volume III: Appendixes B to H. *General Features of the Experimental Installations, Instrumentation, and Operations, Boundary-layer Theory and Work-energy Treatment* (142 pages incl. figs.).

ASHAE Cooperative Research at Case Institute of Technology

G. L. Tuve, D. K. Wright, Jr., and L. J. Seigel: ASHVE Research Report No. 1140—The use of air velocity meters (ASHVE Transactions, Vol. 45, 1939, p. 645).

G. L. Tuve and D. K. Wright, Jr.: ASHVE Research Report No. 1162—Air flow measurements at intake and discharge openings and grilles (ASHVE Transactions, Vol. 46, 1940, p. 313).

G. L. Tuve, G. B. Pricster, and D. K. Wright, Jr.: ASHVE Research Report No. 1204—Entrainment and jet-pump action of air streams (ASHVE Transactions, Vol. 48, 1942, p. 241).

G. L. Tuve and G. B. Priester: ASHVE Research Report No. 1248—Control of air streams in large spaces (ASHVE Transactions, Vol. 50, 1944, p. 153).

Alfred Koestel and G. L. Tuve: ASHVE Research Report No. 1328—The discharge of air from a long slot (ASHVE Transactions, Vol. 54, 1948, p. 87).

Alfred Koestel, Philip Hermann, and G. L. Tuve: ASHVE Research Report No. 1366—Air streams from perforated panels (ASHVE Transactions, Vol. 55, 1949, p. 283).

Alfred Koestel, Philip Hermann, and G. L. Tuve: ASHVE Research Report No. 1404—Comparative study of ventilating jets from various types of outlets (ASHVE Transactions, Vol. 56, 1950, p. 459).

Alfred Koestel and C. Y. Young: ASHVE Research Report No. 1429—The control of air streams from a long slot (ASHVE Transactions, Vol. 57, 1951, p. 407).

G. L. Tuve: ASHVE Research Report No. 1476—Air velocities in ventilating jets (ASHVE Transactions, Vol. 59, 1953, p. 261).

Alfred Koestel and G. L. Tuve: ASHAE Research Report No. 1553—Performance and evaluation of room air distribution systems (ASHAE Transactions, Vol. 61, 1955, p. 533).

Alfred Koestel and J. B. Austin, Jr.: ASHAE Research Report No. 1580—Air velocities in two parallel ventilating jets (ASHAE Transactions, Vol. 62, 1956, p. 425).

Alfred Koestel: ASHAE Research Report No. 1618—Jet velocities from radial flow outlets (ASHAE Transactions, Vol. 63, 1957, p. 505).

ASHVE Cooperative Research at Kansas State College

Linn Helander and C. V. Jakowats: ASHVE Research Report No. 1327—Downward projection of heated air (ASHVE Transactions, Vol. 54, 1948, p. 71).

Linn Helander, S. M. Yen, and R. E. Crank: ASHVE Research Report No. 1475 (In cooperation with the Industrial Unit Heater Association)—Maximum downward travel of heated jets from standard long radius ASME nozzles (ASHVE Transactions, Vol. 59, 1953, p. 241).

Linn Helander, S. M. Yen, and L. B. Knee: ASHVE Research Report No. 1511—Characteristics of downward jets of heated air from a vertical delivery discharge unit heater (ASHVE Transactions, Vol. 60, 1954, p. 359).

S. M. Yen, Linn Helander, and L. B. Knee: ASHAE Research Report No. 1562—Characteristics of downward jets from a vertical discharge unit heater (ASHAE Transactions, Vol. 62, 1956, p. 123).

Linn Helander, S. M. Yen, and Wilson Tripp: ASHAE Research Report No. 1601—Outlet characteristics that affect the downthrow of heated air jets (ASHAE Transactions, Vol. 63, 1957, p. 255).

Rüdiger Knaak: ASHAE Research Report No. 1619—Velocities and temperatures on axis of downward heated jet from 4-inch long-radius ASME nozzle (ASHAE Transactions, Vol. 63, 1957, p. 527).

ASHVE Cooperative Research—General

E. L. Davies: ASHVE Research Reports No. 857, 911, and 966—Measurement of the flow of air through registers and grilles (ASHVE Transactions, Vol. 36, 1930, p. 201; Vol. 37, 1931, p. 619; and Vol. 39, 1933, p. 373).

D. W. Nelson and D. J. Stewart: ASHVE Research Report No. 1076—Air distribution from side wall outlets (ASHVE Transactions, Vol. 44, 1938, p. 77).

A. M. Greene, Jr. and M. H. Dean: ASHVE Research Report No. 1092—The flow of air through exhaust grilles (ASHVE Transactions, Vol. 44, 1938, p. 387).

D. W. Nelson, D. H. Krans, and A. F. Tuthill: ASHVE Research Report No. 1155—The performance of stack heads (ASHVE Transactions, Vol. 46, 1940, p. 205).

A. P. Krats, A. E. Hershey, and R. B. Engdahl: ASHVE Research Report No. 1165—Development of instruments for the study of air distribution in rooms (ASHVE Transactions, Vol. 46, 1940, p. 351).

D. W. Nelson, D. H. Lamb, and G. E. Smedberg: ASHVE Research Report No. 1206—Performance of stack heads equipped with grilles (ASHVE Transactions, Vol. 48, 1942, p. 279).

D. W. Nelson and G. E. Smedberg: ASHVE Research Report No. 1226—Performance of side outlets on horizontal ducts (ASHVE Transactions, Vol. 49, 1943, p. 58).

Other Papers Presented at ASHVE Meetings

D. J. Stewart and G. F. Drake: ASHVE Research Report No. 1051—The noise characteristics of air supply outlets (ASHVE Transactions, Vol. 43, 1937, p. 81).

R. A. Nielsen: ASHVE Research Report No. 1158—Dirt patterns on walls (ASHVE Transactions, Vol. 46, 1940, p. 247).

G. S. Dauphinee and Peter Argentieri: ASHVE Research Report No. 1361—Balancing air delivery of a system of manifold air diffusers (ASHVE Transactions, Vol. 55, 1949, p. 213).

John Rydberg and Per Norbäck: ASHVE Research Report No. 1362—Air distribution and draft (ASHVE Transactions, Vol. 55, 1949, p. 225).

S. F. Gilman, H. E. Straub, A. E. Hershey, and R. B. Engdahl: ASHVE Research Report No. 1471—Room air distribution research for year 'round air conditioning, Part I—Supply outlets at one high sidewall location (ASHVE Transactions, Vol. 59, 1953, p. 151).

H. E. Straub and S. F. Gilman: ASHVE Research Report No. 1504—Room air distribution research for year 'round air conditioning, Part II—Supply outlets at three floor locations (ASHVE Transactions, Vol. 60, 1954, p. 249).

H. G. Elrod, Jr.: ASHVE Research Report No. 1514—Computation charts and theory for rectangular and circular jets (ASHVE Transactions, Vol. 60, 1954, p. 431).

Miscellaneous Articles

G. L. Tuve: Measuring air distribution and grille performance in air conditioning (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, November 1937, p. 700).

F. F. Stevenson: Air discharge from narrow slots (*Heating, Piping and Air Conditioning*, May 1941, p. 308, and June 1941, p. 368. Discussed by J. R. Fellows and D. W. Nelson, *loc. cit.*, September 1941, p. 558 and 559).

Sturm: Vertical air distribution in tall buildings (*Heating, Piping and Air Conditioning*, June 1947, p. 69 and September 1947, p. 93).

Efficient air distribution in theatre (*Heating, Piping and Air Conditioning*, August 1947, p. 107; September 1947, p. 113; November 1947, p. 103).

L. T. Avery and J. Black: Test every air conditioning system (*Heating and Ventilating*, June 1948, p. 96).

L. R. Phillips: Balancing an air conditioning system (*Refrigerating Engineering*, July 1950, p. 111).

What is a draft? (*Heating, Piping and Air Conditioning*, May 1951, p. 67; July 1951, p. 71; September 1951, p. 69; January 1952, p. 113; March 1952, p. 73).

Seven questions on ducts and grilles (*Heating, Piping and Air Conditioning*, May 1952, p. 106 and July 1952, p. 111).

R. V. Martelli and H. O. Scarlett: Air distribution studies (*Heating, Piping and Air Conditioning*, November 1952, p. 77. Discussed by W. O. Huebner, *loc. cit.*, January 1953, p. 119 and by the authors, March 1953, p. 85).

CHAPTER 21

AIR DUCT DESIGN

Pressure Changes, Friction Losses, Circular Equivalents of Rectangular Ducts, Dynamic Losses, Pressure Losses in Elbows, Pressure Losses in Divided-Flow Fittings, Losses Due to Area Changes, Duct Design Methods and Examples, High-Velocity Duct Design, Duct Construction Details, Heat Losses from Ducts

IN THE design of air duct systems, the objective is to provide a system that, within prescribed limits of noise and space available for ducts, efficiently transmits the required flow rate of air (cfm) to each space while maintaining a proper balance between investment and operating cost. When the heating, cooling, or ventilation load is established, the total flow rate of air required can be determined by methods shown in Chapters 12 and 13. The problem is then to distribute the supply air properly among the spaces.

PRESSURE CHANGES

Ductwork imposes resistance to air flow which must be overcome by the expenditure of mechanical energy; this energy is ordinarily supplied by a fan. The portion of the airflow rate from the fan that is transmitted to a particular space is governed by the law that the loss in total pressure from the fan outlet must be the same along each air path. This also holds for the return-air system except that the loss in total pressure refers to the fan inlet. In air-conditioning and ventilating work the pressure differences are ordinarily so small that the equations for incompressible flow can be applied. Additional simplicity is obtained by considering the air to be at the standard density of 0.075 lb per cu ft.

At any cross-section in a duct the total pressure H is the sum of the static pressure P and the velocity pressure H_v. Thus,

$$H = P + H_v \tag{1}$$

Pressures are considered in units of inches of water. The velocity pressure is then given by

$$H_v = \left(\frac{V}{4005}\right)^2 \tag{2}$$

where V is defined by the equation

$$V = Q/A \tag{3}$$

where:

V = mean velocity of flow, feet per minute.
Q = air-flow rate, cubic feet per minute.
A = cross-sectional area of duct, square feet.

The total pressure H is a measure of the total mechanical energy at a cross-section. In any duct system, the total pressure always decreases in the direction of the air flow. Static and velocity pressure are mutually convertible and either can increase or decrease in the direction of flow.

Typical pressure changes in a duct system are illustrated in Fig. 1. Between Sections A and B, the static and total pressures decrease uniformly because of friction between air molecules along the sides of the duct. Pressure losses in straight pipes are termed *friction losses* and are characterized by dependence upon Reynolds number. (See Chapter 4.)

Section BC is a converging section. The duct area is reduced, and consequently the flow is accelerated. This is a stable and efficient process and the loss in total pressure is seen to be small. The marked decrease in static pressure is caused by conversion of static to velocity pressure in accordance with Equation 2.

Section CD presents the friction loss in the section of smaller duct. Since friction losses increase with nearly the square of the velocity, the pressure line falls much more rapidly than in Section AB.

An abrupt expansion occurs at DE. The duct area is abruptly increased and the flow decelerated. The process is an inefficient one and the loss in total pressure is therefore large. The velocity pressure is decreased at E in accordance with Equations 2 and 3, and the static pressure rises. This and the pressure loss at BC are termed *dynamic losses*, and are characterized by being essentially independent of Reynolds number. Dynamic losses occur because of changes in the direction or velocity of the air, and hence occur at duct transitions, bends (elbows), and obstructions such as dampers. As will be shown later, dynamic losses can be specified in terms of a constant times the velocity pressure at a cross-section under consideration.

Beginning at Section E the velocity pressure is constant, and the static and total pressures again decrease uniformly due to wall friction. The static pressure at F is zero, as referred to the atmosphere. The total pressure is the velocity pressure and is a measure of the kinetic energy of the stream as it discharges from the duct.

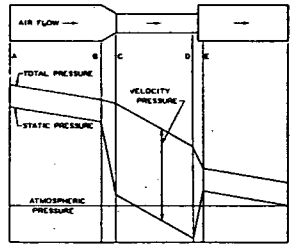
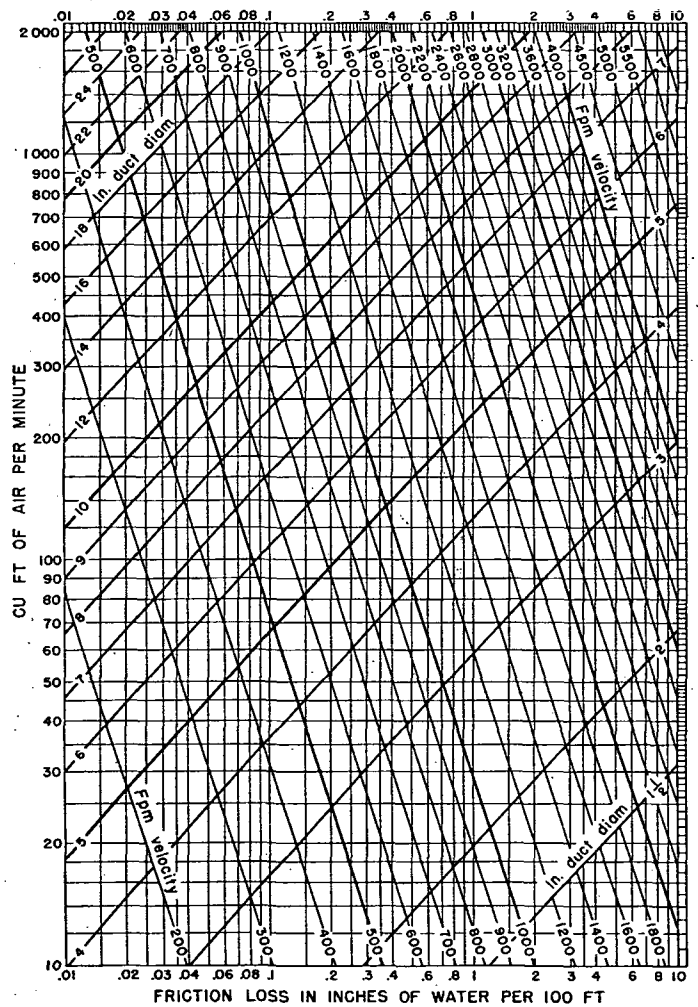
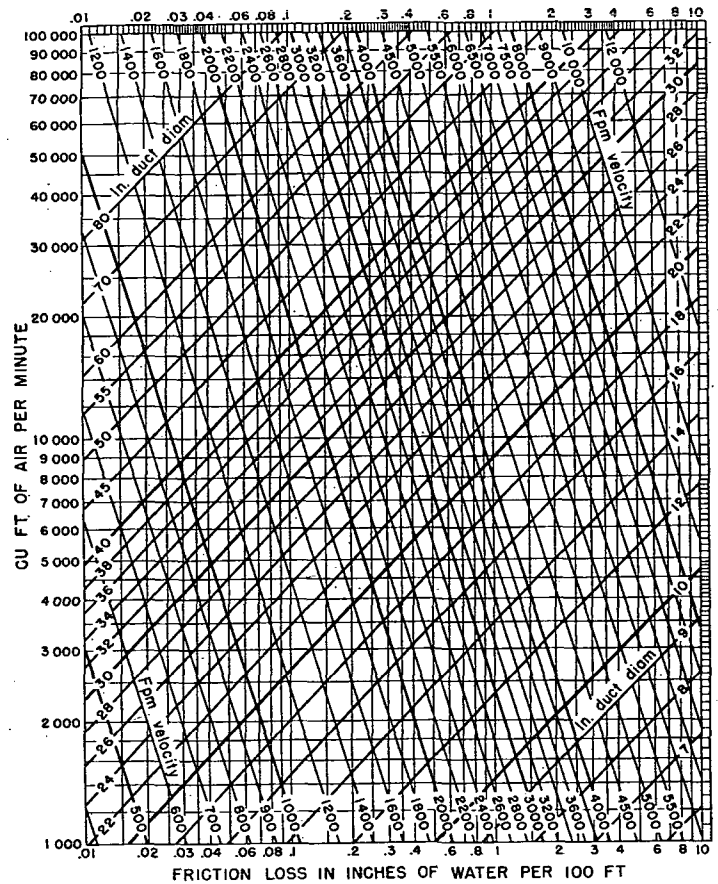


Fig. 1 . . . . Pressure Changes During Flow in Ducts



(Based on Standard Air of 0.075 lb per cu ft density flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.) Caution: Do not extrapolate below chart.

Fig. 2 .... Friction of Air in Straight Ducts for Volumes of 10 to 2000 Cfm



(Based on Standard Air of 0.075 lb per cu ft density flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.)

Fig. 3 .... Friction of Air in Straight Ducts for Volumes of 1000 to 100,000 Cfm

The distinction between *static* and *total pressure* is important, because the former is conventionally used as the *basis for system design*, but the latter determines the actual mechanical energy that must be supplied to the system. Note in Fig. 1 that the static pressure decreases and then increases in the direction of flow. Moreover, it even becomes negative (below atmospheric). Therefore, in dealing with static pressures, distinction must always be made between static pressure *loss* (Section *AB*) and static pressure *change* as a result of conversion of velocity pressure (Section *BC*).

FRICTION LOSSES

Pressure drop in a *straight* duct is caused by surface friction, and this friction loss is most readily calculated by means of the Air Friction Charts, Figs. 2 and 3, covering volume ranges of 10 to 2000 cfm, and 1000 to 100,000 cfm,

respectively. These charts were developed by the ASHAE Research Laboratory.[3]

The charts, Figs. 2 and 3, were constructed from the basic flow equation for the pressure loss in circular ducts (see Chapter 4):

$$H_f = f\left(\frac{l}{D}\right)H_v \qquad (4)$$

where

H_f = head loss due to friction, inches of water.
f = a nondimensional friction coefficient, which for air-conditioning work depends upon Reynolds number and the relative roughness of the conduit. Approximate values of f were taken from the work of Moody[2] where $\epsilon = 0.0005$ ft. (See Chapter 4, Fig. 4, Relation Between Friction Factor and Reynolds number.) It is numerically equal to the reciprocal of the number of duct diameters required to cause a pressure loss equivalent to one velocity pressure.
l = length of conduit, feet.
D = inside diameter of conduit, feet.
H_v = velocity pressure of mean velocity, inches of water.

The air friction chart is based on standard air with a density of 0.075 lb per cu ft, flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft. Fig. 2 should not be used to obtain values below the charts by extrapolation, because critical flow would occur in this region and values so obtained would be unreliable.

Variations in air temperature of the order of ±20 deg from 70 F affect duct friction very little. Figs. 2 and 3 can therefore be used for all air systems with temperatures from 50 F to 90 F.[1] For systems carrying air at much higher temperatures, however, the values found in the charts must be corrected.[3] To determine the friction loss in such systems, the *actual* flow rate or velocity existing at the nonstandard conditions must be used.

Changes in humidity, or the usual changes in barometric pressure or in the pressures required to circulate the air through the systems, have little influence on duct friction and can be disregarded in calculating friction losses in ordinary air-flow systems.[1] For unusual conditions, the friction values obtained from Figs. 2 and 3 must be corrected.[3]

For ordinary applications, an excellent approximation for nonstandard temperatures is

$$h_s = Kh_a \qquad (5)$$

where

h_s = friction loss under actual operating conditions, inches of water.
h_a = friction loss obtained from Figs. 2 and 3 using the actual flow rate or velocity existing under the nonstandard conditions, inches of water.
K = friction loss correction factor, obtained from Fig. 4, dimensionless.

For ducts of other than standard sheet-metal construction, correction factors may be obtained from Figs. 5 and 6. The correction factors shown in Fig. 5 were computed for the value of ϵ, the roughness in feet, shown in Table 1.[3] Most ductwork today is fabricated from galvanized-iron or aluminum sheet metal, or flexible tubing. Fig. 6 presents in graphical form the recommended correction factors for aluminum

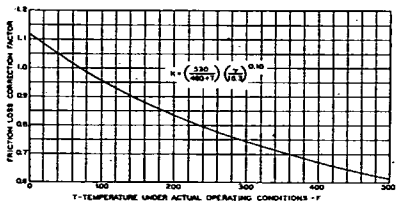


Fig. 4 .... Correction Factors for Density and Viscosity

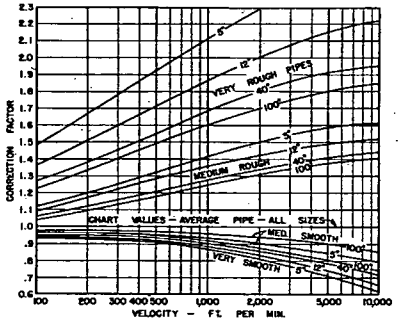
ductwork.[4] The correct friction loss for ducts of other than standard sheet-metal construction may be determined by multiplying the losses obtained from Figs. 2 and 3 by the factors found in Figs. 5 and 6.

Accurate experimental data on the absolute roughness for flexible tubing are not yet available. There is also little information on the friction loss in ducts lined with acoustical material. Until definite information is available, it is recommended that the correction factor for very rough pipes be used for uncovered soft material, and the factor for medium roughness be used for material behind perforated metal.

Example 1 illustrates the use of Fig. 3 to determine friction loss in a galvanized sheet-metal duct. *Examples 2, 3,* and *4* illustrate the use of Figs. 4, 5, and 6 in applying correction factors for ducts of other than sheet-metal construction and for air at high temperature.

Example 1: Determine the friction loss when circulating 10,000 cfm of air at standard density through 75 ft of 24-in. diameter galvanized duct.

Solution: Find 10,000 cfm on the left scale of Fig. 3 and move horisontally right to the diagonal line marked 24 in. The other intersecting diagonal shows that the velocity in the pipe is 3200 fpm. Directly below the intersection it is found that the friction per 100 ft is 0.50 in.; then for 75 ft the friction will be $0.75 \times 0.50 = 0.38$ in. In a like manner, any two variables may



To correct for pipe roughness multiply friction loss obtained from Figs. 2 and 3 by correction factor obtained from Fig. 5.

Fig. 5 .... Correction Factors for Pipe Roughness

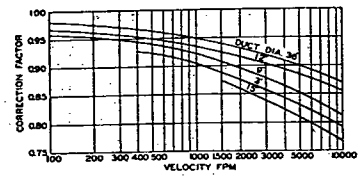


Fig. 6 .... Correction Factors for Aluminum Duct

be determined by the intersection of the lines representing the other two variables.

Example 2: If the duct in *Example 1* is very rough, instead of galvanized, with 40 joints per 100 ft, find the total friction.

Solution: On Fig. 5 find (by interpolation between 12-in. and 40-in. pipe) the intersection of the 24-in. *very rough pipe* line and the 3200 fpm velocity ordinate, and at the left margin read a correction factor of 2. The friction loss in the rough duct is therefore $2 \times 0.38 = 0.76$ in.

Example 3: If the duct in *Example 1* is made of aluminum, instead of galvanized iron, find the total friction.

Solution: On Fig. 6 find (by interpolation between 12-in. and 36-in. pipe) the intersection of the 24-in. line and the 3200-fpm velocity ordinate, and at the left margin read a correction factor of 0.91. The friction loss in the aluminum duct is therefore $0.91 \times 0.38 = 0.35$ in.

Example 4: Determine the friction loss in the duct of *Example 1*, if the temperature of the air transmitted through the duct is 200 F.

Solution: For $t = 200$ F, the density of the air is $\rho_s = 0.060$ lb per cu ft. Hence, the actual air-flow rate in the duct is

$$Q_s = \frac{10{,}000 \times 0.075}{0.060} = 12{,}500 \text{ cfm}$$

and the actual velocity

$$V = \frac{Q_s}{A} = \frac{12{,}500}{3.14} = 4{,}000 \text{ fpm}$$

With $Q_s = 12{,}500$ and $D = 24$ in., find $h_a = 0.77$ in. of water per 100 ft in Fig. 3 and $K = 0.835$ in Fig. 4. The friction loss under actual operating conditions is:

$$h_s = 0.835 \times 0.77 = 0.64 \text{ in. per 100 ft.}$$

The friction loss for 75 ft is then $0.75 \times 0.64 = 0.48$ in.

CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS

An air-handling system is usually sized first for round ducts. Then, if rectangular ducts are desired, their sizes are selected to provide flow rates equivalent to those of the round ducts originally selected.

A comprehensive study at the ASHAE Research Laboratory proved that for most practical purposes rectangular ducts of aspect ratios not exceeding 8:1 will have the same friction pressure loss for equal lengths and mean velocities of flow as a circular duct of the same hydraulic diameter. When duct sizes are expressed in terms of hydraulic diameter, and when equations for friction loss in round and rectangular ducts are equated for equal flow rate and equal length, Equation 6 giving the circular equivalent of a rectangular duct is obtained.[4]

Table 1 .... Values of Roughness ϵ for Different Pipes[a]

Pipe	Degree of Roughness	ϵ Roughness in Feet
Drawn Tubing	Very smooth	0.0000015
New Steel or Wrought Iron Pipe	Medium smooth	0.00015
Galvanized Iron	Average	0.0005
Average Concrete	Medium rough	0.003
Average Riveted Steel	Very rough	0.01

[a] Used in computing values for Fig. 5.

$$d_e = 1.30\frac{(ab)^{0.625}}{(a+b)^{0.250}} = 1.30\sqrt[8]{\frac{(ab)^5}{(a+b)^2}} \qquad (6)$$

where

a = length of one side of rectangular duct, inches.
b = length of adjacent side of rectangular duct, inches.
d_e = circular equivalent of a rectangular duct for equal friction and capacity, inches.

Table 2 gives the circular equivalents of rectangular ducts for equal friction and flow rate for aspect ratios not greater than 11.7:1 based on Equation 6.[4] Note that the mean velocity in a rectangular duct will be *less* than in its circular equivalent.

Multiplying or dividing the length of each side of a duct by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 x 24 in. duct is required, it will be twice that of a 40 x 12 in. duct, or $2 \times 23.0 = 46.0$ in.

DYNAMIC LOSSES

Wherever eddying flow is present, brought about by sudden changes in the direction or magnitude of the velocity of the air flowing, a greater loss in pressure takes place than would occur in a steady flow through a similar length of straight duct having a uniform cross-section. The amount of this loss, in excess of straight-duct friction, is termed *dynamic loss*. Although dynamic losses may be assumed to be caused by changes in area actually occupied by the air flow, for convenience they are divided into two general classes: (1) those caused by *changes in direction* of the duct and (2) those caused by *changes in cross-sectional area* of the duct.

Dynamic losses vary substantially as the square of the mean velocity of the air, and are therefore conveniently expressed as a fraction of the velocity head:

$$H_d = C\left(\frac{V}{4005}\right)^2 \qquad (7)$$

where

H_d = dynamic pressure loss, inches of water.
V = mean velocity of air flow, feet per minute.
C = an experimentally determined constant (dynamic loss coefficient).

Where different areas are involved, subscripts are used to denote the area to which the coefficient applies, as C_1 for inlet area, C_2 for outlet area, and C_o for orifice area.

The dynamic loss coefficient C is dimensionless and represents the number of velocity heads lost at the conduit transition or bend. Values of the dynamic loss coefficient for elbows

Table 2.... Circular Equivalents of Rectangular Ducts for Equal Friction and Capacity

Dimensions in Inches

Side Rectangular Duct	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	9.0	10.0	11.0	12.0	13.0	14.0	15.0	16.0
3.0	3.8	4.0	4.2	4.4	4.6	4.8	4.9	5.1	5.2	5.5	5.7	6.0	6.2	6.4	6.6	6.8	7.0
3.5	4.1	4.3	4.6	4.8	5.0	5.2	5.3	5.5	5.7	6.0	6.3	6.5	6.8	7.0	7.2	7.4	7.6
4.0	4.4	4.6	4.9	5.1	5.3	5.5	5.7	5.9	6.1	6.4	6.8	7.1	7.3	7.6	7.8	8.1	8.3
4.5	4.6	4.9	5.2	5.4	5.6	5.9	6.1	6.3	6.5	6.9	7.2	7.5	7.8	8.1	8.4	8.6	8.9
5.0	4.9	5.2	5.5	5.7	6.0	6.2	6.4	6.7	6.9	7.3	7.6	8.0	8.3	8.6	8.9	9.1	9.4
5.5	5.1	5.4	5.7	6.0	6.3	6.5	6.8	7.0	7.2	7.6	8.0	8.4	8.7	9.0	9.4	9.6	9.8

Side Rectangular Duct	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	22	24	26	28	30	Side Rectangular Duct
6	6.6																				6
7	7.1	7.7																			7
8	7.5	8.2	8.8																		8
9	8.0	8.6	9.3	9.9																	9
10	8.4	9.1	9.8	10.4	10.9																10
11	8.8	9.5	10.2	10.8	11.4	12.0															11
12	9.1	9.9	10.7	11.3	11.9	12.5	13.1														12
13	9.5	10.3	11.1	11.8	12.4	13.0	13.6	14.2													13
14	9.8	10.7	11.5	12.2	12.9	13.5	14.2	14.7	15.3												14
15	10.1	11.0	11.8	12.6	13.3	14.0	14.6	15.3	15.8	16.4											15
16	10.4	11.4	12.2	13.0	13.7	14.4	15.1	15.7	16.3	16.9	17.5										16
17	10.7	11.7	12.5	13.4	14.1	14.9	15.5	16.1	16.8	17.4	18.0	18.6									17
18	11.0	11.9	12.9	13.7	14.5	15.3	16.0	16.6	17.3	17.9	18.5	19.1	19.7								18
19	11.2	12.2	13.2	14.1	14.9	15.6	16.4	17.1	17.8	18.4	19.0	19.6	20.2	20.8							19
20	11.5	12.5	13.5	14.4	15.2	15.9	16.8	17.5	18.2	18.8	19.5	20.1	20.7	21.3	21.9						20
22	12.0	13.1	14.1	15.0	15.9	16.7	17.6	18.3	19.1	19.7	20.4	21.0	21.7	22.3	22.9	24.1					22
24	12.4	13.6	14.6	15.6	16.6	17.5	18.3	19.1	19.8	20.6	21.3	21.9	22.6	23.2	23.9	25.1	26.2				24
26	12.8	14.1	15.2	16.2	17.2	18.1	19.0	19.8	20.6	21.4	22.1	22.8	23.5	24.1	24.8	26.1	27.2	28.4			26
28	13.2	14.5	15.6	16.7	17.7	18.7	19.6	20.5	21.3	22.1	22.9	23.6	24.4	25.0	25.7	27.1	28.2	29.5	30.6		28
30	13.6	14.9	16.1	17.2	18.3	19.3	20.2	21.1	22.0	22.9	23.7	24.4	25.2	25.9	26.7	28.0	29.3	30.5	31.6	32.8	30
32	14.0	15.3	16.5	17.7	18.8	19.8	20.8	21.8	22.7	23.6	24.4	25.2	26.0	26.7	27.5	28.9	30.1	31.4	32.6	33.8	32
34	14.4	15.7	17.0	18.2	19.3	20.4	21.4	22.4	23.3	24.2	25.1	25.9	26.7	27.5	28.3	29.7	31.0	32.3	33.6	34.8	34
36	14.7	16.1	17.4	18.6	19.8	20.9	21.9	23.0	23.9	24.8	25.8	26.6	27.4	28.3	29.0	30.5	32.0	33.0	34.6	35.8	36
38	15.0	16.4	17.8	19.0	20.3	21.4	22.5	23.5	24.5	25.4	26.4	27.3	28.1	29.0	29.8	31.4	32.8	34.2	35.5	36.7	38
40	15.3	16.8	18.2	19.4	20.7	21.9	23.0	24.0	25.1	26.0	27.0	27.9	28.8	29.7	30.5	32.1	33.6	35.1	36.4	37.6	40
42	15.6	17.1	18.5	19.8	21.1	22.3	23.4	24.5	25.6	26.6	27.6	28.5	29.4	30.4	31.2	32.8	34.4	35.9	37.3	38.6	42
44	15.9	17.5	18.9	20.2	21.5	22.7	23.9	25.0	26.1	27.2	28.2	29.1	30.0	31.0	31.9	33.5	35.2	36.7	38.1	39.5	44
46	16.2	17.8	19.2	20.6	21.9	23.2	24.3	25.5	26.7	27.7	28.7	29.7	30.6	31.6	32.5	34.2	35.9	37.4	38.9	40.3	46
48	16.5	18.1	19.6	20.9	22.3	23.6	24.8	26.0	27.2	28.2	29.2	30.2	31.2	32.2	33.1	34.9	36.6	38.2	39.7	41.2	48
50	16.8	18.4	19.9	21.3	22.7	24.0	25.2	26.4	27.6	28.7	29.8	30.8	31.8	32.8	33.7	35.5	37.3	38.9	40.4	42.0	50
52	17.0	18.7	20.2	21.6	23.1	24.4	25.6	26.8	28.1	29.2	30.3	31.4	32.4	33.4	34.3	36.2	38.0	39.6	41.2	42.8	52
54	17.3	19.0	20.5	22.0	23.4	24.8	26.1	27.3	28.5	29.7	30.8	31.9	32.9	33.9	34.9	36.8	38.7	40.3	42.0	43.6	54
56	17.6	19.3	20.9	22.4	23.8	25.2	26.5	27.7	28.9	30.1	31.2	32.4	33.4	34.5	35.5	37.4	39.3	41.0	42.7	44.3	56
58	17.8	19.5	21.1	22.7	24.2	25.5	26.9	28.2	29.3	30.5	31.7	32.9	33.9	35.0	36.0	38.0	39.8	41.7	43.4	45.0	58
60	18.1	19.8	21.4	23.0	24.5	25.8	27.3	28.7	29.8	31.0	32.2	33.4	34.5	35.5	36.5	38.6	40.4	42.3	44.0	45.8	60
62	18.3	20.1	21.7	23.3	24.8	26.2	27.6	29.0	30.2	31.4	32.6	33.8	35.0	36.0	37.1	39.2	41.0	42.9	44.7	46.5	62
64	18.6	20.3	22.0	23.6	25.2	26.5	27.9	29.3	30.6	31.8	33.1	34.2	35.5	36.5	37.6	39.7	41.6	43.5	45.4	47.2	64
66	18.8	20.6	22.3	23.9	25.5	26.9	28.3	29.7	31.0	32.2	33.5	34.7	35.9	37.0	38.1	40.2	42.2	44.1	46.0	47.8	66
68	19.0	20.8	22.5	24.2	25.8	27.3	28.7	30.1	31.4	32.6	33.9	35.1	36.3	37.5	38.6	40.7	42.8	44.7	46.6	48.4	68
70	19.2	21.	22.8	24.5	26.1	27.6	29.1	30.4	31.8	33.1	34.3	35.6	36.8	37.9	39.1	41.3	43.3	45.3	47.2	49.0	70
72															39.6	41.8	43.8	45.9	47.8	49.7	72
74															40.0	42.3	44.4	46.4	48.4	50.3	74
76															40.5	42.8	44.9	47.0	49.0	50.8	76
78															40.9	43.3	45.5	47.5	49.5	51.5	78
80															41.3	43.8	46.0	48.0	50.1	52.0	80
82															41.8	44.2	46.4	48.6	50.6	52.6	82
84															42.2	44.6	46.9	49.2	51.1	53.2	84
86															42.6	45.0	47.4	49.6	51.6	53.7	86
88															43.0	45.4	47.9	50.1	52.2	54.3	88
90															43.4	45.9	48.3	50.6	52.8	54.8	90
92															43.8	46.3	48.7	51.1	53.4	55.4	92
96															44.6	47.2	49.5	52.0	54.4	56.3	96

Equation for Circular Equivalent of a Rectangular Duct[5]

$$d_e = 1.30\,\frac{(ab)^{0.625}}{(a+b)^{0.250}} = 1.30\,\sqrt[8]{\frac{(ab)^5}{(a+b)^2}}$$

where

a = length of one side of rectangular duct, inches.

b = length of adjacent side of rectangular duct, inches.

d_e = circular equivalent of a rectangular duct for equal friction and capacity, inches.

Table 2.... Circular Equivalents of Rectangular Ducts for Equal Friction and Capacity (Concluded)

Dimensions in Inches

Side Rectangular Duct	32	34	36	38	40	42	44	46	48	50	52	56	60	64	68	72	76	80	84	88	Side Rectangular Duct
32	35.0																				32
34	36.0	37.2																			34
36	37.0	38.2	39.4																		36
38	38.0	39.2	40.4	41.6																	38
40	39.0	40.2	41.4	42.6	43.8																40
42	39.9	41.1	42.4	43.6	44.8	45.9															42
44	40.8	42.0	43.4	44.6	45.8	46.9	48.1														44
46	41.7	43.0	44.3	45.6	46.8	47.9	49.1	50.3													46
48	42.6	43.9	45.2	46.5	47.8	48.9	50.2	51.3	52.6												48
50	43.5	44.8	46.1	47.4	48.8	49.8	51.2	52.3	53.6	54.7											50
52	44.3	45.7	47.1	48.3	49.7	50.8	52.2	53.3	54.6	55.8	56.9										52
54	45.0	46.5	48.0	49.2	50.6	51.8	53.2	54.3	55.6	56.8	57.9										54
56	45.8	47.3	48.8	50.1	51.5	52.7	54.1	55.3	56.5	57.8	58.9	61.3									56
58	46.6	48.1	49.6	51.0	52.4	53.7	55.0	56.2	57.5	58.8	60.0	62.3									58
60	47.3	48.9	50.4	51.8	53.3	54.6	55.9	57.1	58.5	59.8	61.0	63.3	65.7								60
62	48.0	49.7	51.2	52.6	54.2	55.5	56.8	58.0	59.4	60.7	62.0	64.3	66.7								62
64	48.7	50.4	52.0	53.4	55.0	56.4	57.7	59.0	60.3	61.6	62.9	65.3	67.7	70.0							64
66	49.5	51.1	52.8	54.2	55.8	57.2	58.6	59.9	61.2	62.5	63.9	66.3	68.7	71.1							66
68	50.2	51.8	53.5	55.0	56.6	58.0	59.5	60.8	62.1	63.4	64.8	67.3	69.7	72.1	74.4						68
70	50.9	52.5	54.2	55.8	57.3	58.8	60.3	61.7	63.0	64.3	65.7	68.3	70.7	73.1	75.4						70
72	51.5	53.2	54.9	56.5	58.0	59.6	61.1	62.6	63.9	65.2	66.6	69.2	71.7	74.1	76.4	78.8					72
74	52.1	53.9	55.6	57.2	58.8	60.4	61.9	63.3	64.8	66.1	67.5	70.1	72.7	75.1	77.4	79.9					74
76	52.7	54.6	56.3	57.9	59.5	61.2	62.7	64.1	65.6	67.0	68.4	71.0	73.6	76.1	78.4	80.9	83.2				76
78	53.3	55.2	57.0	58.6	60.3	62.0	63.4	64.9	66.4	67.9	69.3	71.8	74.5	77.1	79.4	81.8	84.2				78
80	53.9	55.8	57.6	59.3	61.0	62.7	64.1	65.7	67.2	68.7	70.1	72.7	75.4	78.1	80.4	82.8	85.2	87.5			80
82	54.5	56.4	58.2	60.0	61.7	63.4	64.9	66.5	68.0	69.5	71.0	73.6	76.3	79.0	81.4	83.8	86.2	88.6			82
84	55.1	57.0	58.9	60.7	62.4	64.1	65.7	67.3	68.8	70.3	71.8	74.5	77.2	79.9	82.4	84.8	87.2	89.6	91.9		84
86	55.7	57.6	59.5	61.3	63.0	64.8	66.4	68.0	69.5	71.1	72.6	75.4	78.1	80.8	83.3	85.8	88.2	90.6	92.9		86
88	56.3	58.2	60.1	62.0	63.7	65.4	67.0	68.7	70.3	71.8	73.4	76.3	79.0	81.6	84.2	86.8	89.2	91.6	93.9	96.3	88
90	56.9	58.8	60.7	62.6	64.4	66.0	67.8	69.4	71.1	72.6	74.2	77.1	79.9	82.5	85.1	87.8	90.2	92.6	94.9	97.3	90
92	57.4	59.4	61.3	63.2	65.0	66.8	68.5	70.1	71.8	73.3	74.9	77.9	80.8	83.4	86.0	88.7	91.2	93.6	95.9	98.3	92
94	57.9	60.0	61.9	63.8	65.6	67.5	69.2	70.8	72.5	74.1	75.6	78.7	81.7	84.3	86.9	89.6	92.1	94.6	96.9	99.3	94
96	58.4	60.5	62.4	64.4	66.2	68.2	69.8	71.5	73.2	74.3	76.3	79.4	82.6	85.2	87.8	90.5	93.0	95.6	97.9	100.3	96

and other duct elements have been determined experimentally and are given in Tables 3 and 4. It should be noted, however, that absolutely reliable dynamic loss coefficients are not yet available for all duct elements and that the information available for pressure losses due to area changes is generally restricted to symmetrical area changes.

Fig. 7, which shows the relation of velocity pressure to velocity for standard air ($V = 4005\sqrt{H_v}$), can be conveniently used to find the total dynamic pressure loss for any duct element with known *dynamic loss coefficient* C.

PRESSURE LOSSES IN ELBOWS

Dynamic-loss coefficients are nearly independent of the air velocity and are affected by the roughness of the duct walls only in the case of bends, for which the dynamic losses are often grouped with the friction losses to facilitate design calculations. An ASHAE survey[6] of available data has indicated that the method of expressing the combined dynamic and friction losses due to an elbow as equivalent to the loss in a length L of similar straight duct is justified for design purposes, owing to the relation of the loss to the corresponding friction factor f.

Fig. 8 gives the *additional equivalent length* of duct in terms of widths W for elbows in rectangular ducts; Fig. 9 gives the *equivalent length* of duct in terms of diameters D for round ducts. When these curves for *additional equivalent length* are used, the straight lengths of duct between elbows should be measured to the intersection of their center lines.

Example 5: Use of Fig. 8 for the calculation of elbow losses.

Given the portion of a duct system shown in Fig. 10, it is required to determine the pressure loss between points A and D. Air at standard conditions is being supplied at the rate of 2000 cfm in a 6 by 24 in. galvanized duct of average construction. Elbows No. 1 and 2 have center line radii of 18 and 19 in., respectively.

Solution: For elbow No. 1 the radius ratio is $R_1/W_1 = 18/24 = 0.75$ and the aspect ratio is $H_1/W_1 = 6/24 = 0.25$. The additional equivalent length for elbow No. 1 in terms of W is obtained from Fig. 8; $(L/W)_1 = 11.5$. Thus $L_1 = 11.5 \times 24/12 = 23$ additional equivalent feet. Similarly for elbow No. 2, the ratio radius is $R_2/W_2 = 9/6 = 1.5$ and the aspect ratio is $H_2/W_2 = 24/6 = 4.0$; Fig. 8 gives $(L/W)_2 = 6$, so $L_2 = 6 \times 6/12 = 3$ additional equivalent feet.

The total length of the straight runs from A to D is $l = l_{A\text{-}1} + l_{1\text{-}2} + l_{2\text{-}D} = 7 + 20 + 5 = 32$ ft and the additional equivalent length due to the elbows is $L = L_1 + L_2 = 23 + 3 = 26$ ft. Thus the equivalent length of the system from A to D is $l + L = 32 + 26 = 58$ ft of 6 by 24 in. duct.

The diameter of a circular duct, equivalent in friction and capacity to this rectangular duct, is 12.4 in. as given by the table of circular equivalents, Table 2. At a delivery rate of

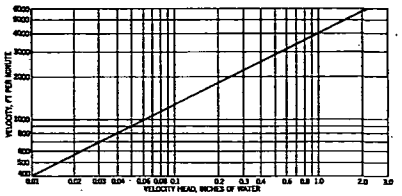


Fig. 7 .... Relation Between Velocity and Velocity Head for Standard Air

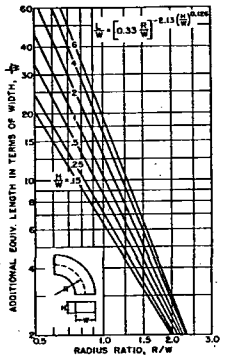


Fig. 8 .... Loss in 90-Deg Elbows of Rectangular Cross-Section

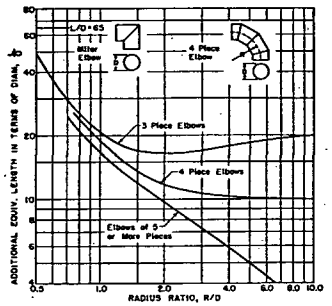


Fig. 9 .... Loss in 90-Deg Elbows of Round Cross-Section

Table 3 .... Pressure Losses Due to Elbows

(Additional Equivalent Losses in Excess of Friction to Intersection of Center Lines)

TYPE	ILLUSTRATION	CONDITIONS	PRESSURE LOSS C^a	L/D	L/W
MITER		RECTANGULAR OR ROUND; WITH OR WITHOUT VANES	$\frac{H}{90}$ TIMES VALUE FOR SIMILAR 90-DEG. ELBOW[9]		
90-DEG. ROUND SECTION		MITER	1.30[b]	65[b]	
		R/D=0.5	0.90		
		0.75	0.45	23	
		1.0	0.33	17	
		1.5	0.24	12	
		2.0	0.19	10	
90-DEG. RECTANGULAR SECTION		H/W R/W MITER	1.25[b]		25[c]
		0.5	1.25		25
		0.25 0.75	0.60		12
		1.0	0.37		7
		1.5	0.19		4
		MITER	1.47		49
		0.5	1.10		40
		0.5 0.75	0.30		16
		1.0	0.28		9
		1.5	0.13		4
		MITER	1.50		75
		0.5	1.00		50
		1.0 0.75	0.41		21
		1.0	0.22		11
		1.5	0.09		4.5
		MITER	1.38		110
		0.5	0.96		65
		4.0 0.75	0.37		45
		1.0	0.18		17
		1.5	0.07		8
90-DEG. SQUARE SECTION WITH SPLITTER VANES		H/W R_1/W R_2/W MITER 0.5			28[d]
		0.5 0.4	0.70[e]		19
		0.7 0.8			12
		1.0 1.0	0.13		7.2
		1.5	0.12		
		MITER 0.5 0.5			22[d]
		0.5 0.2 0.4	0.45		16
		0.75 0.4 0.7	0.12		
		1.0 0.7 1.0	0.10		
		1.5 1.3 1.8	0.15		
MITER WITH TURNING VANES		PLATE VANES	0.35[b]		
		FORMED VANES	0.10[b]		
MITER TEE WITH VANES		CONSIDER EQUAL TO A SIMILAR ELBOW. BASE LOSS ON ENTERING VELOCITY.			
RADIUS TEE					

[a] Values based on f values of approximately 0.02.
[b] Values calculated from L/D and L/W values of Reference 6 for f = 0.02.
Note: Superscript numbers refer to references at end of chapter.

2000 cfm, the ASHAE FRICTION CHART, Fig. 3, gives a loss of 0.6 in. of water per 100 ft. of 12.4-in. diameter duct. Thus the loss from A to D is $0.6 \times 58/100 = 0.348$ in. of water.

The use of elbows of radius ratio, $R/W = 1.5$, is considered good practice with respect to both installation and operation. In a given rectangular duct 6 by 24 in., for example, the elbow loss will be greater for a flat bend where the aspect ratio, $H/W = 1/4$, than if the bend of the same radius ratio had been made in the plane of the narrow dimension giving an aspect ratio, $H/W = 4$.

Data now available for losses in compound bends,[7, 8] where two or more elbows are close together, do not warrant refinement of design calculations beyond use of the sum of the losses for the individual elbows. Actually, the losses are somewhat less than for two bends when they are placed to form a U bend, and somewhat more when they form a reversed, or S bend.

Where angles of other than 90-deg bend are encountered, the loss may be considered as directly proportional to the

Table 4 .... Pressure Losses Due to Area Changes

TYPE	ILLUSTRATION	CONDITIONS	LOSS COEFFICIENT		TYPE	ILLUSTRATION	CONDITIONS	LOSS COEFFICIENT
ABRUPT EXPANSION	A_1 A_2	A_1/A_2	C_1	C_2	ABRUPT CONTRACTION SQUARE EDGE	A_1 A_2	A_2/A_1	C_2
		0.1	0.81	81			0.0	0.34[12]
		0.2	0.64	16			0.2	0.32
		0.3	0.49	5			0.4	0.25
		0.4	0.36	2.25			0.6	0.18
		0.5	0.25	1.00			0.8	0.06
		0.6	0.16	0.45				
		0.7	0.09	0.18	GRADUAL CONTRACTION	A_1 A_2 θ	θ	
		0.8	0.04	0.06			30°	0.02[9]
		0.9	0.01	0.01			45°	0.04
GRADUAL EXPANSION	A_1 A_2 θ	θ	C_v				60°	0.07
		5°	0.17[9]		EQUAL AREA TRANSFORMATION	A_1 A_2 θ	$A_1' A_2$ $\theta \leq 14°$	C
		7°	0.22					0.15[9]
		10°	0.28		FLANGED ENTRANCE		$A=\infty$	C
		20°	0.45					0.34[12]
		30°	0.59		DUCT ENTRANCE		$A=\infty$	C
		40°	0.73					0.85[9]
ABRUPT EXIT	A_1 A_2 $(A_2=\infty)$	$A_1/A_2=0.0$	1.00		FORMED ENTRANCE		$A=\infty$	C
		A_1/A_2	C_e					0.03[9]
SQUARE EDGE ORIFICE EXIT	A_1 A_2	0.0	2.50[12]		SQUARE EDGE ORIFICE ENTRANCE	A_1 A_2	A_2/A_1	C_e
		0.2	2.44				0.0	2.50[12]
		0.4	2.28				0.2	1.90
		0.6	1.99				0.4	1.39
		0.8	1.54				0.6	0.96
		1.0	1.00				0.8	0.61
BAR ACROSS DUCT	$\frac{L}{D}$	E/D	C				1.0	0.34
		0.10	0.7[9]		PIPE ACROSS DUCT	$\frac{L}{D}$	E/D	C
		0.25	1.4				0.10	0.20[9]
		0.50	4.0				0.25	0.55
							0.50	2.0
					STREAM-LINED STRUT ACROSS DUCT	$\frac{L}{D}$	E/D	C
							0.10	0.07[9]
							0.25	0.23
							0.50	0.90
					SQUARE EDGE ORIFICE IN DUCT	$A_1 = A_2$ A_1 A_2	A_2/A_1	C_e
							0.0	2.50[12]
							0.2	1.86
							0.4	1.21
							0.6	0.64
							0.8	0.20
							1.0	0.0

Note 1: Subscript on C indicates cross-section at which velocity is calculated (See, for example, Equation 10 in text.)
Note 2: Superscript numbers refer to references at end of chapter.

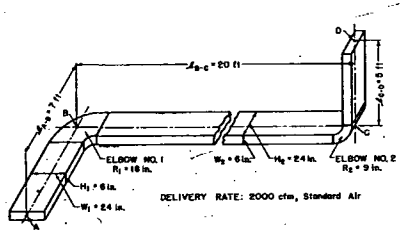


Fig. 10 .... Portion of Duct System for *Example 5*

Table 5 .... Ratio of Pressure Loss to Branch Velocity Pressure

Take-off Angle	Ratio of Velocity in Branch to Velocity in Main Duct						
	0.4	0.6	0.8	1.0	1.5	2.0	3.0
90-deg	6.5	3.1	2.0	1.5	0.95	0.74	0.62
60-deg	5.0	2.2	1.3	0.77	0.47	0.47	0.58
45-deg	3.5	1.3	0.64	0.43	0.40	0.45	0.54

angle of bend. Losses[9] for elbows discharging air directly into a large space are higher than those given for elbows within duct systems.

Turning vanes may be advantageously employed in elbows, both to reduce the pressure loss and to provide a more uniform velocity distribution downstream from the bend.[9, 10] Vanes and concentric splitters are particularly recommended where miter elbows are used, because even the simplest vane forms will produce a substantial saving in pressure loss.

PRESSURE LOSSES IN DIVIDED-FLOW FITTINGS

Analysis[11] of available data for losses at branch take-offs indicates that the loss in the straight-through section is about 35 percent of that for abrupt enlargement (1) involving the same ratios of velocities and (2) that the loss in the diverted-flow section depends on the ratio of the velocity of diverted flow to total flow and on the angle the take-off makes with the main, and is at a minimum for a natural relation that exists between these two variables. Some representative branch losses are given in Table 5.

LOSSES DUE TO AREA CHANGES

Area changes in ducts, generally unavoidable, are necessitated frequently by the building construction or changes in the volume of air carried. Experimental investigations[13, 23, 14, 12] of pressure changes, and pressure losses at changes of the area of duct cross-sections, indicate that the excess pressure loss over the normal friction loss is a dynamic loss due to a faster stream expanding into a slower stream, as determined by the actual areas occupied by the flow rather than the areas of the duct. No perceptible dynamic loss is due to the converging of the air stream itself where the flow is contracted, but the air stream continues to converge beyond the edge of the contraction and reaches a minimum at the *vena contracta*. For contraction, therefore, the dynamic loss is caused by expansion from the vena contracta to the full area following the contraction. Abrupt contraction in area may therefore be considered as a special condition of abrupt enlargement. Fig. 11 illustrates (*a*) abrupt enlargement and (*b*) abrupt contraction.

For a sudden symmetrical enlargement, a theoretical expression for the loss is

$$H_e = \left(\frac{V_1 - V_2}{4005}\right)^2 = \left(1 - \frac{A_1}{A_2}\right)^2 \left(\frac{V_1}{4005}\right)^2 = C_1 \left(\frac{V_1}{4005}\right)^2 - C_2 \left(\frac{V_2}{4005}\right)^2 \quad (8)$$

where

H_e = pressure loss due to sudden enlargement, based on standard air, inches of water.
V_1 = velocity of standard air in the inlet duct, feet per minute.



Fig. 11 .... Air Flow at Abrupt Enlargement or Contraction of Air Stream

V_2 = velocity of standard air in the outlet duct, feet per minute.
A_1 = area of the inlet duct, square feet.
A_2 = area of the outlet duct, square feet.
C_1 = loss coefficient based on area A_1.
C_2 = loss coefficient based on area A_2.

For a gradual symmetrical enlargement, Equation 8 changes to

$$H_{ge} = C_r\left(\frac{V_1 - V_2}{4005}\right)^2 = C_1C_1\left(\frac{V_1}{4005}\right)^2 = C_2C_2\left(\frac{V_2}{4005}\right)^2 \quad (9)$$

where

H_{ge} = pressure loss due to gradual enlargement, inches of water.
C_r = coefficient of loss, as ratio of loss to loss for abrupt expansion, dependent upon the total angle included between the sides of the duct.

The loss for a sudden symmetrical contraction can be expressed as

$$H_c = C_2\left(\frac{V_2}{4005}\right)^2 = C_1\left(\frac{V_1}{4005}\right)^2 = C_c\left(\frac{V_c}{4005}\right)^2 \quad (10)$$

where

H_c = pressure loss due to sudden contraction (Fig. 10b).
C_c = loss coefficient based or orifice area A_c.
V_c = velocity of air through orifice, feet per minute.

The loss for a gradual symmetrical contraction can be similarly expressed as

$$H_{gc} = C_2\left(\frac{V_2}{4005}\right)^2 \quad (11)$$

where the coefficient of loss C_2 depends on the included angle of the sides of the duct and the sharpness of the edges at the junction of taper to following duct section.

DUCT DESIGN

The following discussion refers to ducts for commercial and industrial heating, ventilating, and air-conditioning systems of the central station type. The design procedures given yield the static pressure required to overcome the resistance of the ductwork, including the supply outlets and return intakes. The fan selected for the duct system must not only produce this pressure but also the additional pressure required by the central equipment such as washers or spray chambers, heating or cooling coils, and filters. Pressure losses of these components should be obtained from the manufacturers' catalogs.

Special duct design procedures for heating ducts used in residences can be found in Chapter 18 Warm Air Heating Systems. The design of ducts in industrial exhaust systems is discussed in Chapter 52.

General rules which should be followed in the design of ducts are:

1. The air should be conveyed as directly as possible at the permissible velocities to obtain the desired results with minimum noise and greatest economy of power, material, and space.

2. Sudden changes in the direction or velocity of the air should be avoided. When sudden changes are necessary at bends, turning vanes should be used to minimize the pressure loss.

3. Diverging transition pieces should be made as gradual as practicable. As shown in the section on area changes, losses in abrupt enlargements are high and therefore such transitions should be avoided. The included angle of divergence for enlargement should not exceed 20 deg. Losses in contractions are low but the included angle of convergence should not be greater than 60 deg.

4. Where the greatest air carrying capacity per square foot of sheet metal is desired, rectangular ducts should be made as nearly square as possible. Aspect ratios (ratio of width to depth) greater than 8 to 1 should be avoided. Where possible, a ratio of 4 to 1 or less should be maintained.

5. Ducts should be constructed of smooth material, such as steel or aluminum sheet metal. For ducts made from other materials, proper allowance for the change in roughness should be made.[8]

6. Through the design procedures which follow, a reasonably precise estimate of the flow resistances offered by the system can be obtained. However, it should be recognized that in actual installations, resistances may vary considerably from the calculated values because of variation in the smoothness of materials, types of joints used, and the ability of workmen to fabricate the system in accordance with the design. Fans and motors should therefore be selected to provide at least a slight factor of safety, and dampers should be installed in each branch outlet for balancing the system.

Procedure for Duct Design

The general procedure is as follows:

1. Study the plan of the building and arrange the positions of the supply outlets to provide proper distribution of air within each space. Select outlet sizes from manufacturers' catalog data.

2. Draw a sketch of the most convenient system of ductwork, connecting the supply outlets and return intakes with the central station apparatus, taking cognizance of the building construction, avoiding all obstructions in steel work and equipment, and at the same time maintaining a simple design.

3. Calculate the sizes of all main and branch ducts by one of the procedures given in the following section.

4. Determine the total pressure requirement of the supply and return duct systems. Although the loss in total pressure of each duct run connecting the fan and each supply outlet (or return intake) should be calculated and made the same for all runs, ordinarily only the pressure loss of the duct run apparently having the greatest resistance is referred to as the pressure loss of the duct system. Dampers are relied upon for balancing the system.

Design Velocities

It is not possible to give specific rules for selecting duct velocities, but the velocities given in Table 6 have been found to give satisfactory results in designing conventional systems. Since the fan horsepower increases approximately as the square of the velocity, and noise generation increases with static pressure, velocities should be kept low for quiet and economical operation. On the other hand, as evident from Equation 3, at a given flow rate the duct size increases with decreasing velocity. For multistory buildings, it is sometimes possible to reduce the height between floors by using very small ducts, thereby effecting a considerable reduction in building investment cost. The various space-saving systems which are becoming increasingly numerous are discussed in the section High-Velocity Systems in Chapter 19 Central Systems for Air Conditioning. The design of high-velocity ducts is discussed briefly in a later section, Design of High-Velocity Ducts, in this chapter.

DESIGN METHODS

In the design of air duct systems, three methods are employed: (1) *velocity-reduction*, (2) *equal-friction*, and (3) *static-regain*. The three methods and their refinements represent different design levels of accuracy and complexity, and they should be selected, therefore, to suit the application. Simple duct systems may be designed as quickly and easily as possible, but for large installations the system static-

Table 6 .... Recommended and Maximum Duct Velocities for Conventional Systems

Designation	Recommended Velocities, Fpm: Residences	Schools, Theaters, Public Buildings	Industrial Buildings
Outdoor Air Intakes[a]	500	500	500
Filters[a]	250	300	350
Heating Coils[a]	450	500	600
Air Washers	500	500	500
Suction Connections	700	800	1000
Fan Outlets	1000–1600	1300–2000	1600–2400
Main Ducts	700–900	1000–1300	1200–1800
Branch Ducts	600	600–900	800–1000
Branch Risers	500	600–700	800
	Maximum Velocities, Fpm		
Outdoor Air Intakes[a]	800	900	1200
Filters[a]	300	350	350
Heating Coils[a]	500	600	700
Air Washers	500	500	500
Suction Connections	900	1000	1400
Fan Outlets	1700	1500–2200	1700–2800
Main Ducts	800–1200	1100–1600	1300–2200
Branch Ducts	700–1000	800–1300	1000–1800
Branch Risers	650–800	800–1200	1000–1600

[a] These velocities are for total face area, not the net free area; other velocities in table are for net free area.

pressure requirement must be determined as precisely as possible, the most accurate method being recommended.

Velocity-Reduction Method

This method consists of selecting the velocity at the fan discharge and designing for progressively lower velocities in the main at each branch duct. With the selected velocities and known air-flow rates, the various duct diameters are read directly from Figs. 2 or 3, and the equivalent rectangular sizes are obtained from Table 2. The pressure loss of the run having apparently the highest resistance is determined by adding the straight pipe, elbow, and transition losses; this total value represents the fan static pressure required for the supply-duct system. The return-air system is sized similarly, starting with the lowest velocities at the return intakes and increasing them progressively in the direction of the fan inlet. Dampers are relied upon for balancing the system.

A refinement of this method is to size the several branch ducts to dissipate the pressure available at the entrance to each. The pressure loss of the ductwork between the fan and first branch take-off is subtracted from the now known fan static pressure to obtain the available pressure at each junction. By trial, a branch velocity is found that results in the branch pressure loss being equal to, or somewhat less than, that available. The procedure is repeated for each branch.

If the fan is specified so that the static pressure available for the ductwork is known, the method consists of finding, by trial, the velocities in the main duct that will result in a pressure loss equal to the pressure available. The branch ducts are then sized as previously explained if such a refinement is deemed necessary.

The merits of the *velocity-reduction* method are that (1) duct sizes are determined very easily, and (2) velocities can be limited to those known to be safe from causing noise problems. Its weaknesses are: (1) proper choice of velocities requires experience and judgment, and (2) the designer cannot always determine by inspection which run probably has the highest resistance.

Equal-Friction Method

The principle of this method is to make the pressure loss per foot of length the same for the entire system. With this method little balancing is required for symmetrical layouts in which all runs have about the same resistance. For layouts having both short and long runs, the shortest run will require considerable dampering.

Usual practice is to select the velocity in the main duct near the fan from the standpoint of noise for the particular application. Since the flow rate (cfm) is known, this establishes a value of friction loss per 100 ft of duct in Figs. 2 or 3. This same friction loss value is maintained throughout the design. For example, the flow rate in the main after the first branch take-off is reduced by that handled by the branch. Therefore, proceed vertically downward in Figs. 2 or 3 to the new flow rate value, and read the velocity and diameter. Note that the velocity is reduced by this procedure. An advantage of this method is that it *automatically* reduces the duct velocities in the direction of flow, thereby insuring that noise will not be a problem. The equivalent rectangular size of any diameter is obtained from Table 2. By continuing the procedure, all sections, including branch ducts, are sized from Figs. 2 and 3 at the same friction loss per foot of actual length.

After sizing the system the pressure loss of the run having apparently the highest resistance is calculated. In so doing, the pressure losses of all elbows and transitions are included, and are expressed in terms of *equivalent length* of straight pipe.

The principal limitation of the *equal-friction* method is that it does not differentiate between runs having several transitions, elbows, etc., and runs having none. Only the *actual length* of duct is considered; this and the flow rate fix the duct size. Moreover, when computing the system resistance, care must be exercised that the pressure losses of transitions, elbows, etc., are included and added to the straight pipe losses.

If the pressure available for the ductwork is known, as it is for packaged equipment, this pressure can be divided by the total equivalent length of the run apparently having the highest resistance to obtain a design friction loss value per foot, for use with Fig. 2 or Fig. 3. Hence, for these applications, it is not necessary to select an initial velocity. However, the method then has the weakness that the resistances of fittings must be expressed in terms of *equivalent length*. Since transitions, elbows, etc., have predominantly dynamic losses, the equivalent length of a particular fitting varies considerably with its actual size. Note, for example, the values for elbows in Fig. 7 of Chapter 18 are related to duct size, and note also, that the elbow losses in Fig. 9 of this chapter are given in terms of the number of *diameters*. Hence, when the available pressure is known, the method requires that the duct size be estimated in advance. The calculated duct size should therefore be compared with the initial estimate, and if considerably different, a recalculation should be made using the calculated size.

Less dampering is required if the method is modified so that only the main duct is sized by the equal-friction method. The fan is selected for this total duct resistance and the pressure available at each branch take-off is found in the

manner described for the *velocity-reduction* method. The pressure available at each branch is divided by its equivalent length, in hundreds of feet, to obtain a design friction loss value for use with Fig. 2 or Fig. 3 in conjunction with the branch flow rate. The branch ducts are therefore sized as nearly as possible to dissipate all of the available pressure.

When using this modified method, care should be exercised that the velocities in short branches do not become excessive from a noise standpoint. This is easily guarded against during the design process, because the velocity can be read directly from the friction chart. If it is excessive, move horizontally to the left on the chart and select a diameter which yields a reasonable velocity. The damper for this run will have to dissipate the excess pressure. Since ductwork attenuates noise to some extent, the damper should be located as close to the main as possible. Sound treatment for this branch should also be considered. An alternative solution may be to revise the duct layout to increase the resistance of the run, for example, by relocating the branch take-off so that the total duct length is increased.

Example 6: (Equal-Friction Method). A duct layout is shown in Fig. 12. Outlets Nos. 1 and 2 deliver 750 cfm each and outlet No. 3 delivers 1000 cfm. Selecting a velocity of 1600 cfm in Section *A*, size the duct system and determine its static-pressure requirement.

Solution: The total cfm to be handled is 2500 cfm. From Fig. 3, with 2500 cfm and 1600 fpm velocity, read a diameter of 17 in. and a friction loss of 0.2 in. of water per 100 ft. By subtraction, the flow rate in Section *B* is 1750 cfm. Along the 0.2 friction line in Fig. 3, all of the ducts can be sized immediately because the flow rates are known. Results are presented in Table 7.

The rectangular equivalents were selected from Table 2 with the objective of having the same duct depth for all three branch runs.

The duct run to outlet No. 3 has the highest apparent resistance. It is decided to fabricate the elbow in Section *C* with a radius ratio of 1.2; hence, from Fig. 8 with $H/W = 1.9$, $L/W = 8$. Since $W = 1.25$ ft (15 in.), the additional equivalent length due to the elbow L is 10 ft. The total equivalent length of the run is therefore $(20 + 10 + 15 + 10 + 15) = 70$ ft. Therefore, at 0.2 per 100 ft the duct resistance is $0.2 \times 0.70 = 0.14$ in. of water. Adding to this the outlet pressure of 0.12 in., the static-pressure requirement of the duct system is 0.26 in. of water. The design is now complete, and dampers will be relied upon for adjusting the outlets to the design flow rates.

If refinement is deemed necessary, the modified method can be applied to Sections *D* and *E*. First, the static pressures available at the junctions with the main of the Section *D* and *E* branch ducts are obtained. For Section *D* it is the system pressure of 0.26 minus the friction pressure loss in Section *A*. The latter is $0.20 \times (20/100) = 0.04$; hence, the pressure at the entrance of Section *B* is 0.22 in. of water. Deducting the outlet pressure loss of 0.12, that available for the ductwork is 0.10. Assume the equivalent lengths of the branch take-off and the elbow to be 10 ft each. The total equivalent length of Section *D* is then $(10 + 10 + 10 + 5) = 35$, and the friction loss per 100 ft required to dissipate 0.10 in. of water is $0.10 \times (100/35) = 0.29$. With this unit friction loss and a flow rate of 750 cfm, Fig. 2 yields a diameter of 10.0 in. and a velocity of 1380 fpm.

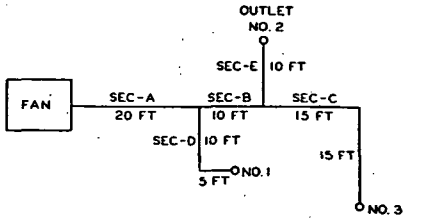


Fig. 12 .... Duct Layout for *Example 6*

Table 7 .... Tabulation of Results (*Example 6*)

Section	Flow Rate Cfm	Friction per 100 ft In. H_2O	Duct Dia. In.	Velocity Fpm	Rectangular Duct In.
A	2500	0.2	17.0	1600	20 × 12
B	1750	0.2	14.8	1480	15 × 12
C	1000	0.2	12.0	1290	15 × 8
D	750	0.2	10.7	1190	12 × 8
E	750	0.2	10.7	1190	12 × 8

Section *E* is sized in a similar manner. The pressure available is 0.26 minus the friction loss in Sections *A* and *B*; hence, 0.20. With the outlet pressure loss of 0.12 deducted, the available ductwork pressure loss is 0.08 in. Assuming that the branch take-off loss is equivalent to 10 ft of duct, the total equivalent length is 20 ft. The required friction loss is $0.08 \times (100/20) = 0.40$. With this unit friction loss and a flow rate of 750 cfm, Fig. 2 yields a diameter of 9.4 in. and a velocity of 1580 fpm. An equivalent rectangular size is 9×8 in.

Comparing these results with those in Table 7, it is evident that the modified method has reduced the size of Section *D* somewhat and that of Section *E* considerably. The reduced sizes accomplish more economically what would otherwise have to be done with dampers.

Static-Regain Method

Consider a straight run of duct with several branch take-offs attached. The flow rate of air along the run is progressively reduced by the amount diverted into each successive take-off. If, for example, the size of the run were the same throughout its length, the velocity would become progressively less in accordance with Equation 3. When velocities are reduced, a conversion of velocity pressure into static pressure occurs (as well as a loss in total pressure). The principle of the *static-regain* method is to size a duct run so that the increase in static pressure (regain) at each take-off junction just offsets the pressure loss of the succeeding section of the run.

The method provides a convenient means of designing a long run of duct having several take-offs so that essentially the same static pressure exists at the entrance to each branch. If, instead of branch ducts, supply outlets are connected directly to the run, then essentially the same static pressure will exist behind each outlet. As a consequence, outlet selection and system balancing is simplified. The method is particularly suited to large installations having several long runs of duct, with each run having many take-offs or supply outlets attached. For this type of application, little or no dampering is ordinarily required to balance the system.

The initial velocity in the main duct is selected from noise and pressure loss considerations, and the branch ducts are sized by the modified equal-friction method.

If the distance between branch take-offs is either very small or very great, it may not be feasible or economically desirable to design for the same static pressure at each junction. In such cases, the method can be used to size the main for either a progressively lower static pressure (net static-pressure loss) or a progressively higher pressure (net static-pressure gain).

If no friction or dynamic losses occurred at the junction, there would be no loss in total pressure, and the change in velocity pressure would be completely converted into a regain (rise) in static pressure, which for standard air would be:

$$P_t = \left(\frac{V_1}{4005}\right)^2 - \left(\frac{V_2}{4005}\right)^2 \qquad (12)$$

where

P_t = theoretical static-pressure regain, inches of water.
V_1 = velocity in main upstream of branch, feet per minute.
V_2 = velocity in main downstream of branch, feet per minute.

Under the best practice, 0.7 to 0.8 of the change in velocity pressure is actually recovered, but for practical design an average recovery of 0.5 is assumed. Hence, the actual regain P_r is:

$$P_r = 0.5\left[\left(\frac{V_1}{4005}\right)^2 - \left(\frac{V_2}{4005}\right)^2\right] \qquad (13)$$

Design charts based on Equation 13 and rectangular ducts having aspect ratios of 3 to 1 or less are presented in Figs. 14 and 15.

The *duct length* of any section should include the equivalent length of any elbows or transitions within the section. The charts apply to constructions where regain takes place unaccompanied by radical change in direction; namely, to *straight-through sections* of divided-flow fittings.

Example 7: (Static-Regain Method). The duct shown in Fig. 13 handles 8000 cfm. Determine the duct sizes in Section *A*, *B*, *C*, *D*, *E*, *F*, and *G*, maintaining an operating pressure of 0.12 in. water in the duct behind each outlet. Find the total pressure loss of the system.

Solution: The following nine steps indicate the solution:

1. Assume the velocity in Section *A* to be 1500 fpm. This results in an initial duct size of 48 x 16 in. The 16-in. duct depth will be maintained throughout the system.

2. The circular equivalent of a 48 x 16-in. duct is 29.2 in. (from Table 2), and with 8000 cfm flowing in this duct, the friction loss from Fig. 3 is 0.13 in. per 100 ft.

3. To size the branch ducts, determine the shortest equivalent length of duct up to the first outlet. Section $B = 25$ ft; Section $F = 10$ + equivalent length of elbow. Assume the width of the duct in Section *F* to be 15 in.; therefore $H/W = 15/16 = 0.94$. Also the radius of the elbow should not be less than the width ($R/W = 1.00$), resulting in an equivalent length L, from Fig. 8, of $10 \times (15/12) = 12.5$ ft. Equivalent length of Section $F = 10 + 12.5 = 22.5$ ft. Therefore, Section *F* being shorter than Section *B*, size Section *F* first, using the same friction rate as in Section *A*, 0.13 in. per 100 ft. In Fig. 2, with 2000 cfm and a friction rate of 0.13 in., find an equivalent diameter of 17 in. for Section *F*. The rectangular equivalent of a 17-in. duct is 15 x 16 in. (from Table 2), resulting in a velocity of 1200 fpm.

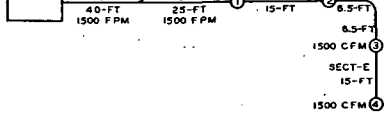


Fig. 13 .... Duct Layout for *Example 7*

Table 8 .... Tabulation of Results (*Example 7*)

Section	Air Volume Cfm	Equivalent Length Ft	Velocity Fpm	Rectangular Duct In.	Dia In.	Friction Per 100 ft In. H_2O	Net Pressure Loss In. H_2O
A	8000	40	1500	48 × 16	29.2	0.13	0.05
B	6000	25	1500	36 × 16	—	—	0.03
C	4500	15	1300	31 × 16	—	—	0
D	3000	26[a]	1040	26 × 16	—	—	0
E	1500	15	860	16 × 16	—	—	0
F	2000	22.5	1200	15 × 16	17	0.13	.03
G	1000	15	900	10 × 16	—	—	0

[a] Includes 13 equivalent ft for elbow (see Step 7 in *Example 7*).

Note: The shorter equivalent length line is sized first to prevent velocities in other branches from exceeding recommended values.

4. Loss in Section $F = 0.13 \times 0.225 = 0.03$ in. water.

5. Since the operating pressure of Outlet 1 is to be the same as that of Outlet 6, the loss in Section *B* must equal the loss in Section *F*. Using the Static-Regain Chart, Fig. 15, size Section *B* for a net loss of 0.03 in. water.

The procedure for using these charts is indicated by arrowheads and dashed lines on Fig. 15. Proceed as follows:

a. Locate the velocity of the preceding (upstream) duct section along the velocity scale on the left margin (1500 fpm).

b. Proceed horizontally to the air volume abscissa (6000 cfm).

c. Proceed parallel to the curved lines to intersect the diagonal base line.

d. Go vertically to the net loss desired (0.03 net loss).

e. Go horizontally to the air velocity base line.

f. Proceed parallel to the curved lines to intersect ordinate for 25-ft duct length.

g. Move horizontally to the air velocity scale and read the velocity in the downstream section of duct (1500 fpm).

6. This procedure is repeated for Section *C* except that the *no gain or loss* line is used in previous Step 5 d. Outlet No. 2 will then have the same static pressure behind it as Outlet No. 1, thus fulfilling the problem condition of the same operating pressure for all outlets.

7. At this point, the equivalent length of the elbow in Section *D* should be estimated. Its width will be somewhat less than that of Section *C*. Assume $W = 26$ in.; hence $H/W = 16/26 = 0.6$. With a typical radius ratio of 1.25, from Fig. 8, $L/W = 6$; therefore $L = 6 \times (26/12) = 13$ ft. Adding this to the actual length of Section *D* results in a total equivalent length of 26 ft.

8. Using the *no gain or loss* line in Fig. 14, size Sections *D*, *E*, and *G*. Results are listed in Table 8.

9. The total pressure loss of the system is the sum of the losses in Sections *A* and *B* (or *F*) and the outlet operating pressure:

Loss in Section *A*	= 0.13 × 0.40 = 0.05
Loss in Section *B* (or *F*)	= 0.03
Outlet Operating Pressure	= 0.12
Total Pressure Loss	= 0.20 in.

DESIGN OF HIGH-VELOCITY DUCTS

The transmission of air at high velocities, though common in industrial exhaust systems for many years, has gained wide acceptance in comfort air-conditioning and ventilation systems in the past few years only. This acceptance is due partly to the use of improved fans and of special sound-attenuation and control equipment, and partly to improved design and installation methods based upon a better understanding of the general problems connected with the design

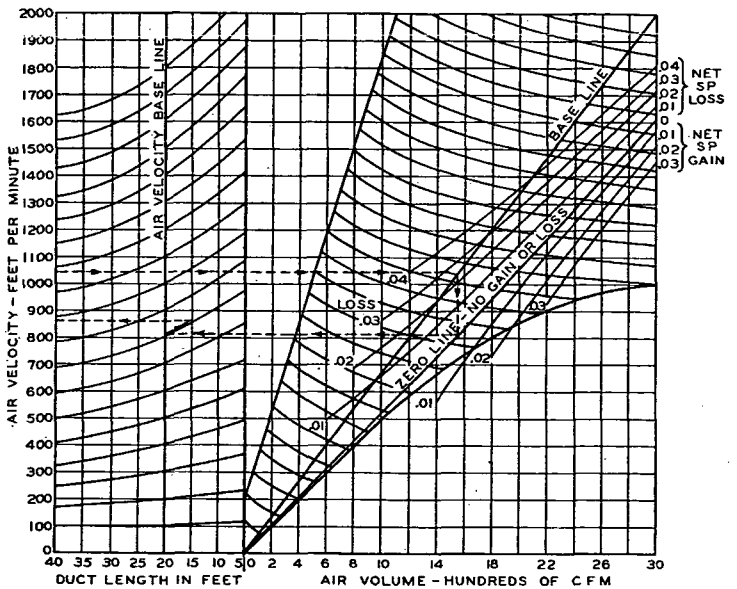


Fig. 14 .... Static-Regain Chart*-For Determining Velocity in Ducts Carrying 0 to 3000 Cfm

* Based on Equation 13.

and installation of high-velocity air-conditioning systems.[16-27] For general information on these systems, the reader is referred to Chapter 19—Central Systems for Air Conditioning.

The design of high-velocity duct systems involves a compromise between reduction of duct size and the consequent necessity for higher fan horsepower. While the size of the ducts and hence the air velocities are governed in large part by the available building space, the maximum velocities given later in this section should not be exceeded without careful examination of all factors involved.

The following *general rules* will prove helpful in designing high-velocity duct systems:

1. The same general rules which apply to conventional duct design also apply to high-velocity duct design. (See Section—Design Methods.)

2. When designing high-velocity systems, static regain must be accounted for. Latest data indicate that approximately 85 percent of the original velocity pressure can be regained. This may easily account for 1 in. water static pressure and, therefore, a method which does not take into consideration static regain will result in the selection of oversized fans and motors.

3. Proper sound control must be provided. To take care of low-frequency fan noise, a sound attenuator must usually be installed after the fan. When acoustic terminal devices are not used, attenuator chambers or sound-lined ducts must be installed after mixing valves, in order to attenuate duct and valve noise to acceptable levels.

4. Air velocities in main ducts may vary from 2500 fpm in some installations to over 6000 fpm in others. Corresponding maximum branch-duct velocities will then vary from 2000 fpm to 4500 fpm.

One general procedure for duct sizing used by many designers is to maintain velocities approximately constant in main riser and trunks until the friction loss reaches 1 in. per 100 feet. When further reductions in duct size take place, velocities are selected to maintain a constant friction loss of 1 in. per 100 feet. Final branches are generally designed with constant area so that decreasing capacity and velocity will allow full static regain. With this procedure the final velocity of an end branch duct may be 1000 fpm, or even lower.

The above procedure is not intended to be exclusive, since ducts may also be designed by other methods. (See foregoing section.)

5. Round ducts are used to a great extent in high-velocity systems, because they are easier to seal and do not require bracing or stiffening. However, it is commercially practicable to utilize rectangular ductwork in high-velocity systems. The duct aspect ratio should be kept low and should not exceed 4:1. The duct must be rigidly braced to withstand the static pressure of the system. All longitudinal seams and cross-seams must be made tight. Mastic sealing compounds and special tapes are available as less expensive substitutes for positive seals such as soldering or welding. (See Duct Construction Details.)

6. Low-loss take-offs and fittings should be used. Rigid turning vanes should be installed in all rectangular duct right-angle elbows. In round ducts, turning vanes should be avoided and long-radius elbows used instead. Both 90-deg and 45-deg duct take-offs are used. However, the use of conical take-offs or angular take-offs is recommended, in order to keep duct friction as low as possible.[28]

7. Duct dampers are not required in the inlet connections to an acoustic terminal device. Capacity dampering at each outlet can be adjusted by means of the acoustic terminal damper. A branch duct supplying air to a number of acoustic terminals may be equipped with a branch-line balancing damper as an aid in field balancing. In a single-duct system, the branch-line damper may be manually adjusted. In a dual-duct system, the branch-line balancing damper may be a static-pressure regulating damper controlling the pressure in the branch-line duct. In this manner the branch-duct capacity may be raised or lowered by adjusting the setting on the branch-duct damper. Face dampers on the two decks of a one-fan dual-duct system also may be used as static-pressure regulating dampers. An inlet-vane damper on the fan is usually effective for system capacity control.

Control

In primary-air high-velocity systems, it is not necessary to control static pressures in order to prevent static-pressure unbalance, because this system operates with constant air volume. In all-air high-velocity systems operating with variable air flow, some control of static pressures may be required in order to prevent *static-pressure unbalance*, which is a large deviation from design static pressure at the inlet of a terminal, caused by large deviations in air flow.

In single-duct high-velocity systems, operating with changing air volume, the variations in static pressures can be limited by:

a. Static-pressure controllers operating dampers in the air-distributing-system.

b. Static-pressure controllers operating inlet vane dampers on the fan.

c. Zoning and changing air-supply temperature in response to static-pressure changes.

In dual-duct systems, the daily and seasonal variations in heating and cooling loads produce constantly changing demands for cold and warm air, causing a wide variance in flow and consequently in duct static pressures. It is, therefore, necessary to control the total fan delivery and in some cases the duct static pressure to limit pressure at the terminals.

At present, there are three methods used by the industry to control static pressures in dual-duct systems:

a. By dampers operated by static-pressure regulators located at critical points in the air-distributing system. The number, arrangement, and location of such dampers and static taps are dictated entirely by the complexity of the air-distributing systems, the initial fan pressure, and the particular fan characteristics.

b. By static-pressure controllers, regulating cold and warm air temperatures in order to limit the variations in the air flow in individual ducts. This method is applicable to centrally zoned systems and may involve an increase in total fan capacity.

c. By volume regulators in each individual air-mixing valve or acoustic terminal device. When this method is used, fixed volumetric delivery is maintained at each outlet, and the system characteristics remain constant through the entire range of operation.

Performance and pressure-volume characteristics of dual-duct systems are discussed in the paper by N. S. Shataloff.[28] In the other papers, listed in the reference section of this chapter, may be found further discussions of the types of air-conditioning systems which use high-velocity ducts. (References 16–23 and 25–27.)

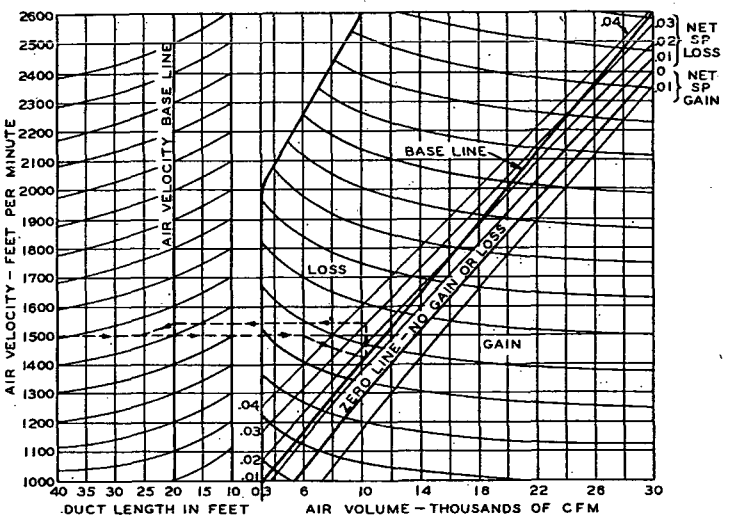


Fig. 15 .... Static-Regain Chart*-For Determining Velocity in Ducts Carrying 3000 to 30,000 Cfm

* Based on Equation 13.

Table 9 .... Recommended Construction for Rectangular Sheet-Metal Ducts

High Pressure	Medium Pressure	Low Pressure			Duct Dimension in Inches	Recommended Construction
Steel U.S. Std. Gage	Steel U.S. Std. Gage	Steel U.S. Std. Gage	Aluminum B & S Gage	Copper Cold Rolled		Transverse Joints and Bracing
22	24	26	24 (0.020)	16 oz	Up thru 12	S slip, drive slip, 1 in. pocket lock on 8 ft centers.
20	22	24	22 (0.025)	24 oz	13 thru 18	S slip, drive slip, 1 in. pocket lock on 8 ft centers.
					19 thru 30	S slip, 1 in. pocket lock on 4 ft centers. S slip, 1 in. pocket lock on 8 ft centers with 1 × 1 × 1/8 angles 4 ft from joint. S slip, 1 in. pocket lock on 8 ft centers with cross break 1 in. standing seam on 5 ft centers.
18	20	22	20 (0.032)	32 oz	31 thru 42	1 in standing S cleat, bar slip, pocket lock on 4 ft centers. 1 in. standing S cleat, bar slip, pocket lock on 8 ft centers with 1 × 1 × 1/8 in. angle 4 ft from joint. 1 in. standing seam on 4 ft centers. Longitudinal standing seam with 1 × 1 × 1/8 in. angles on 4 ft centers.
					43 thru 54	1 1/4 in. standing S cleat, bar slip, pocket lock on 4 ft centers. 1 1/4 in. standing S cleat, bar slip, pocket lock on 8 ft centers with 1 1/4 × 1 1/4 × 1/8 in. angles 4 ft from joint. 1 1/4 in. standing seam on 3 ft centers.
16	18	20	18 (0.040)	36 oz	55 thru 60	Longitudinal standing seam inside with 1 1/4 × 1 1/4 × 1/8 in. angles on 4 ft centers.
					61 thru 84	1 1/4 in. standing S cleat, bar slip, pocket lock on 4 ft centers with 1 1/4 × 1 1/4 × 1/8 in. angles 4 ft from joint. 1 1/4 in. standing S cleat, bar slip, pocket lock on 8 ft centers with 1 1/4 × 1 1/4 × 1/8 in. angles on 2 ft. centers. 1 1/4 in. standing seam on 3 ft centers. Longitudinal standing seams inside with 1 1/4 × 1 1/4 × 1/8 in. angles on 2 ft centers.
14	16	18	16 (0.051)	48 oz	85 thru 96	1 1/4 in. standing S cleat, bar slip, pocket lock reinforced with 1 1/4 × 1 1/4 × 1/8 in., or companion angles on 4 ft centers. 1 1/4 in. standing S cleat, bar slip, pocket lock reinforced with 1 1/4 × 1 1/4 × 3/16 in. or companion angles on 8 ft centers with 1 1/4 × 1 1/4 × 3/16 in. angles on 2 ft centers. Longitudinal standing seams inside with 1 1/4 × 1 1/4 × 1/8 in. angles on 2 ft centers.
					Over 96	1 1/4 in. standing S cleat, bar slip, pocket lock reinforced with 2 × 2 × 1/4 in. angles, or companion angles on 4 ft centers. 1 1/4 in. standing S cleat, bar slip, pocket lock reinforced with 2 × 2 × 1/8 in. angles, or companion angles on 8 ft centers with 2 × 2 × 1/4 in. angles on 2 ft centers. 1 1/4 in. standing seams with 2 × 2 × 1/4 in. angles on 2 ft centers. Longitudinal standing seams inside with 2 × 2 × 1/4 in. angles on 2 ft centers.

Note: Where special rigidity or stiffness is required, ducts should be constructed of metal two gage numbers heavier e.g., use 22 gage instead of 24 gage. Ducts larger than 96 in. require special field study for hanging and supporting methods. Other joint construction of equivalent mechanical strength and airtightness may be used. Recommended methods of fastening bracing to ductwork include riveting, bolting, and tack welding. Bracing, stiffening members, or angle connections for copper ductwork, where exposed to the weather, should be of brass or aluminum, of suitable thickness for strength.

DUCT CONSTRUCTION DETAILS

The recommended gages for steel (or iron), aluminum, and copper sheet-metal rectangular ducts are given in Table 9. No general agreement has yet been reached as to the exact limits of the classifications *high pressure, medium pressure,* and *low pressure* shown in Table 9. Suggested ranges of static pressures *in the duct system* for these classifications are:

Low Pressure = up to 1½ in. water.
Medium Pressure = 1½ in. to 4 in. water.
High Pressure = 4 in. to 8 in. water.

It should be noted that these values are the pressures existing in the duct, not the total pressure at the fan.

Table 10 .... Recommended Duct Construction for Round and Flat-Oval Ducts[a, b]

Steel U.S. Std. Gage		Size Dia or Max Width	Bracing on Flat Surface
Round	Flat-oval		
24	22	To 10 in.	None
22	20	11 to 20 in.	1 1/4 × 1 1/4 × 1/8 in. angle or 1 1/2 × 1/2 × 1/8 in. channel on 4 ft centers.
20	18	21 to 40 in.	1 1/2 × 1 1/2 × 1/8 in. angle or 2 × 1/2 × 1/8 in. channel on 4 ft centers.
18	16	41 to 60 in.	2 × 2 × 1/8 in. angle or 2 1/2 × 5/8 × 3/16 in. channel on 2 ft centers.
16	14	61 in. and over	2 × 2 × 1/8 in. angle or 2 1/2 × 5/8 × 3/16 in. channel on 2 ft centers.

[a] All fittings should be of same gage as pipe or heavier.
[b] Satisfactory for pressure in duct up to 10 in. water.

The recommended gages for steel (or iron) sheet-metal round or flat-oval ducts are given in Table 10.

Steel or iron sheets are specified according to the manufacturers or U. S. Standard Gage System. Aluminum sheets are specified according to the American or Brown & Sharpe Gage System. Copper sheets are usually specified by the weight in ounces per square foot. Weights of black and galvanized-steel and -iron sheets per square foot of surface for various gages are given in Table 11. Similar data for 2S aluminum sheets will be found in Table 12. Weights of standard copper sheets are given in Table 13. In calculating the total weight of a given length of ductwork from these tables, it is customary to add 15 percent to the weight to cover hangers, slips, and scrap. Duct bracing or stiffening members should be calculated separately.

Rectangular ducts are generally constructed by breaking the corners and grooving the longitudinal seam, although some fabricators still use the standing seam. Elbows and transformation sections are generally formed with Pittsburgh corner seams because this seam is easier to lock in place than the double seam, but double seams are equally acceptable. The constructions of these various seams, as well as the types of girth connections, are shown in Fig. 16. The application of the various slips, connections, and bracing is outlined in Table 9. The end slip may be used wherever *S* slips are recommended.

Designs *M* to *P* of Fig. 16 are for flush-type seams on ductwork where joints are to be concealed. For smooth external appearance the seams may be filled with suitable filler. Screws or rivets used should be of oven-head type.

Round and flat-oval ducts should be fabricated in accordance with the recommendations given in Table 10. Sections may be assembled using beaded couplings, swaged ends, or beaded and crimped small ends. Sections are held in place, using suitable sealers, with sheet-metal screws or rivets, or are welded together.

Spiral conduit is acceptable in place of round or flat-oval duct, with recommended gages shown in the following table:

Diameter, Inches	Recommended U. S. Std. Gage
3 to 8	Not lighter than 26 gage
9 to 24	Not lighter than 24 gage

Table 11 .... Weights of Black and Galvanized Sheets

	U. S. Std. Gage	Approximate Thickness, In.		Weight Per Square Foot	
		Steel	Iron	Ounces	Pounds
Black Sheets	30	0.0123	0.0125	8	0.500
	28	0.0153	0.0156	10	0.625
	26	0.0184	0.0188	12	0.750
	24	0.0245	0.0250	16	1.000
	22	0.0306	0.0313	20	1.250
	20	0.0368	0.0375	24	1.500
	18	0.0490	0.0500	32	2.000
	16	0.0613	0.0625	40	2.500
	14	0.0766	0.0781	50	3.125
	12	0.1072	0.1094	70	4.375
	11	0.1225	0.1250	80	5.000
	10	0.1379	0.1406	90	5.625
Galvanized Sheets[a]	30	0.0163	0.0165	10.5	0.656
	28	0.0193	0.0196	12.5	0.781
	26	0.0224	0.0228	14.5	0.906
	24	0.0285	0.0290	18.5	1.156
	22	0.0346	0.0353	22.5	1.406
	20	0.0408	0.0415	26.5	1.656
	18	0.0530	0.0540	34.5	2.156
	16	0.0653	0.0665	42.5	2.656
	14	0.0806	0.0821	52.5	3.281
	12	0.1112	0.1134	72.5	4.531
	11	0.1265	0.1290	82.5	5.156
	10	0.1419	0.1446	92.5	5.781

[a] Galvanized sheets are gaged before galvanizing and are therefore approximately 0.004 in. thicker.

Table 12 .... Weights and Thicknesses of 2S Aluminum (Density 0.098 lb/cu in.)

B. & S. Gage	Thickness, Inches		Weight per Square Foot	
	Decimal	Nearest Fraction	Ounces	Pounds
28	0.012	1/64	2.7	0.169
26	0.016	1/64	3.6	0.226
24	0.020	1/64	4.5	0.282
22	0.025	1/32	5.4	0.353
20	0.032	1/32	7.2	0.452
18	0.040	3/64	9.0	0.563
16	0.051	3/64	11.5	0.720
14	0.064	1/16	14.4	0.903

Aluminum sheets of the 2S- and 3S-type alloy and ¾ hard temper are readily workable, and can be used for practically all ductwork. The 2S type (commercially pure aluminum) is suitable for all, except very large ducts. For large ducts, where more strength is desired, the 3S alloy with ¾ or ½ hard temper is frequently used. The higher tempers, particularly full hard, do not have the formability of the lower tempers. Joints can be of any of the standard designs, and can be fabricated in the same manner as iron. Repeated sharp bending and rebending should be avoided, as aluminum has a tendency to crack under such treatment. Aluminum of 16 B. & S. gage or heavier can readily be welded by the metallic arc or acetylene process. Riveting is done in the same manner as in iron or steel sheet. Self-tapping screws tend to loosen because of the softness of aluminum.

The construction of elbows and changes of shape cannot be definitely outlined, because of the varied conditions en-

Table 13 .... Weights and Thicknesses of Standard Copper Sheets[a]

Rolled to Weight

Weight per Sq Ft		Thickness, Inches		Nearest Gage No.		
Oz	Lb	Decimal Equivalent	Nearest Fraction	B. & S.	Stubs	U. S. Std.
10	0.625	0.0135	1/64	27	29	29
12	0.750	0.0162	1/64	26	27	28
14	0.875	0.0189	1/64	25	26	26
16	1.000	0.0216	1/32	23	24	25
18	1.125	0.0243	1/32	22	23	24
20	1.250	0.0270	1/32	21	22	23
24	1.500	0.0324	1/32	20	21	22
28	1.750	0.0378	1/32	19	20	20
32	2.000	0.0432	3/64	17	19	19
36	2.250	0.0486	3/64	16	18	18
40	2.500	0.0540	3/64	15	17	17
44	2.750	0.0594	1/16	15	17	17
48	3.000	0.0648	1/16	14	16	16
56	3.500	0.0756	5/64	13	15	14
64	4.000	0.0864	5/64	11	14	13

[a] Variations from these weights must be expected in practice.

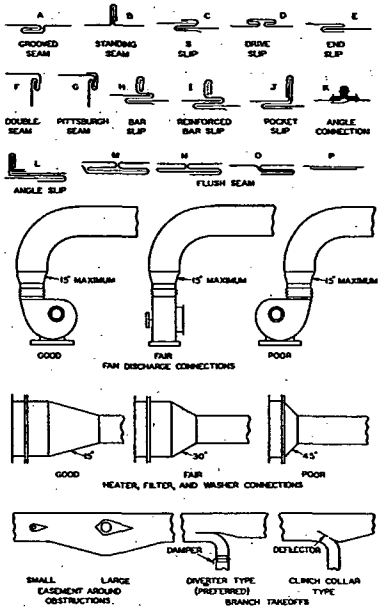


Fig. 16 .... Sheet-Metal Duct and Arrangement Details

countered in the field, but in general, long-radius elbows and gradual changes in shape tend to maintain uniform velocities accompanied by decreased turbulence, lower resistance, and a minimum of noise.

Heavy canvas connections are recommended on both the inlet and outlet to all fans. Self-vulcanising adhesive tapes are available for this purpose and for sealing joints in duct-work. Where a fire hazard exists, the material used must satisfy the requirements of any codes or authorities having jurisdiction. The fan discharge connections shown in Fig. 16 are marked *good*, *fair*, and *poor* in the order of the amount of turbulence produced. An inspection of the heater connections shown in Fig. 16 will readily show that uniform velocity through the heater cannot be expected in the diagram noted *poor*. When obstructions cannot be avoided, the duct area should never be decreased more than 10 percent, and then a *streamlined* collar should be used. Larger obstructions require an increase in the duct size in order to maintain as nearly uniform velocity as possible. Branch take-offs should always be arranged to cut or slice into the air stream in order to reduce as far as possible the losses in velocity head.

Wherever ducts pass through fire walls or connect two fire areas of a building, automatic fire dampers should be provided. For design of such dampers and other fire-protective details, see Pamphlet No. 90A of the National Board of Fire Underwriters.[29]

HEAT LOSSES FROM DUCTS

In designing duct systems, the heat gains or losses of the ducts can be quite considerable, not only if the duct passes through unconditioned space, but also on long duct runs within conditioned space. Proper insulation will remedy this situation considerably, but sometimes a redistribution of the supply air is necessary in order to compensate for the heat exchanges that occur.

The heat loss from a given length of duct can be expressed by:

$$Q_w = UPl\left[\left(\frac{t_1 + t_2}{2}\right) - t_3\right] \qquad (14)$$

where

Q_w = heat loss through duct walls, Btu per hour.
U = overall coefficient of heat transfer for the duct wall, Btu per (hour) (square foot) (Fahrenheit degree).
P = perimeter of duct, feet.
l = length of duct, feet.
t_1 = temperature of air entering duct, Fahrenheit.
t_2 = temperature of air leaving duct, Fahrenheit.
t_3 = temperature of air surrounding duct, Fahrenheit.

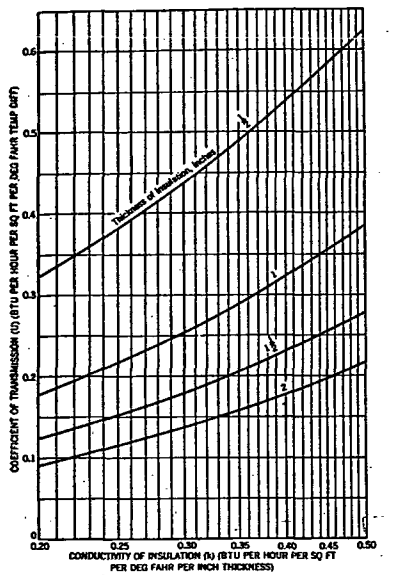
To obtain the temperature drop in warm air for a given distance of transmission, or the temperature rise if the duct carries air cooler than the room through which it passes, the following formulas can be used:

$$t_1 = \frac{t_3(y + 1) - 2t_3}{(y - 1)} \qquad (15)$$

$$t_2 = \frac{t_1(y - 1) + 2t_3}{(y + 1)} \qquad (16)$$

where

$$y = \frac{28.8AV\rho}{UPl} \text{ for rectangular ducts or,}$$



[a] *For round ducts less than 30 in. diameter, increase heat transmission values by the percentages shown below.*

Thickness of Insulation (Inches)	1/2	1	1 1/2	2
12- to 21-in. Duct Diameter	3%	5%	7%	9%
21- to 30-in. Duct Diameter	1%	2%	3%	4%

Fig. 17 .... Heat Loss Coefficients for Insulated Ducts[a]

$$y = \frac{7.2DV\rho}{Ul} \text{ for round ducts,}$$

A = cross-sectional area of duct, square feet.
ρ = density of air, pounds per cubic foot.
V = mean velocity of fluid, feet per minute.
D = diameter of round duct, feet.

In using Equations 15 and 16, one of the duct air temperatures will be unknown and will be obtained by substitution of the other known or assumed values.

Heat loss coefficients for insulated ducts with various conductivities are given in Fig. 17. The conductivities of various materials, which are based on mean temperatures, about 70 F, will be found in Table 4 of Chapter 9. For cases where the mean temperature is other than that at which the test was conducted, a correction should be made. However, in most cases the effect of this factor will be small and may be neglected.[a]

Example 8: Determine the entering air temperature and heat loss for a duct 24 x 36 in. cross-section and 70 ft in length, insulated with 1½-in. of a material having a conductivity of 0.35 Btu at 86 F mean temperature, carrying air at a velocity of 1200 fpm, measured at 70 F, to deliver air at 120 F with air surrounding the duct at 40 F.

Solution: Referring to Fig. 17, the overall heat transmission coefficient is found to be 0.49 Btu. From Table 2, Chapter 3, the density of air at 70 F and 29.921 in. Hg is found to be 1/13.348 = 0.0749 lb per cu ft. Substituting these and the other given values in Equation 16, y and t_1 will be as follows:

$$y = \frac{28.8 \times 6 \times 1200 \times 0.0749}{0.49 \times 10 \times 70} = 45.3$$

$$t_1 = \frac{120(45.3 + 1) - 80}{45.3 - 1} = 123.7 \text{ F}$$

Substituting in Equation 14:

$$Q_w = 0.49 \times 10 \times 70\left[\left(\frac{123.7 + 120}{2}\right) - 40\right]$$

$$= 28{,}100 \text{ Btu per hr.}$$

For special considerations which apply to insulation of ducts in marine installations see Chapter 48.

REFERENCES

[1] D. K. Wright, Jr.: ASHVE Research Report No. 1280—A new friction chart for round ducts (ASHVE Transactions, Vol. 51, 1945, p. 303).

[2] L. F. Moody: Friction factors for pipe flow (*ASME Transactions*, Vol. 66, 1944, p. 671).

[3] R. D. Madison and W. R. Elliot: Friction charts for gases including correction for temperature, viscosity and pipe roughness (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, October 1946, p. 107).

[4] F. W. Hutchinson: ASHVE Research Report No. 1469—Friction losses in round aluminum ducts (ASHVE Transactions, Vol. 59, 1953, p. 127).

[5] R. G. Huebscher: Friction equivalents for round, square and rectangular ducts (ASHVE Transactions, Vol. 54, 1948, p. 101).

[6] D. W. Locklin: ASHVE Research Report No. 1405—Energy losses in 90-degree duct elbows: A survey and analysis of available information (ASHVE Transactions, Vol. 56, 1950, p. 479).

[7] J. R. Weske: *Pressure Loss in Ducts with Compound Elbows* (National Advisory Committee for Aeronautics, *Advance Restricted Report* W-39, February 1943).

[8] R. D. Madison and J. R. Parker: Pressure losses in rectangular elbows (*Heating, Piping and Air Conditioning*, July, p. 365; August, p. 427; September, p. 483; 1936).

[9] W. H. Carrier, R. E. Cherne, and W. A. Grant: *Modern Air Conditioning, Heating and Ventilating* (Pitman Publishing Corp., New York, 1950, 2nd ed., p. 248).

[10] M. C. Stuart, C. F. Warner, and W. C. Roberts: ASHVE Research Report No. 1216—Effect of vanes in reducing loss in elbows in seven-inch square ventilating duct (ASHVE Transactions, Vol. 48, 1942, p. 409).

[11] S. F. Gilman: Pressure losses of divided-flow fittings (ASHAE Transactions, Vol. 61, 1955, p. 281).

[12] G. E. McElroy: *Pressure Losses Due to Bends and Area Changes in Mine Airways* (*U. S. Bureau of Mines Information Circular* I.C. 6663, p. 4).

[13] A. P. Krats and J. R. Fellows: *Pressure Losses Resulting from Changes in Cross-Sectional Area in Air Ducts* (University of Illinois, *Engineering Experiment Station Bulletin* No. 300).

[14] R. D. Madison (ed.): *Fan Engineering* (Buffalo Forge Company, Buffalo, New York, 1948, 5th ed., p. 124).

[15] J. R. Henry: *Design of Power Plant Installations: Pressure-Loss Characteristics of Duct Components* (National Advisory Committee for Aeronautics, *Advance Restricted Report* 1AF26, June 1944, L-203).

[16] L. O. Paul: Installing ducts for higher pressures, and Why higher duct velocities? (*Heating, Piping and Air Conditioning*, April 1953, p. 98 and February 1954, p. 109).

[17] R. D. Tutt: Modern trends in air distribution (*Refrigerating Engineering*, May 1953, p. 509).

[18] W. W. Kennedy: Design factors in high velocity air distribution (*Heating and Ventilating*, January 1954, p. 83).

[19] J. W. Kreuttner: Can air conditioning be simplified in large buildings? (*Heating, Piping and Air Conditioning*, August 1954, p. 94).

[20] Gardner Savage: Air conditioning an operating hotel (*Heating and Ventilating*, September 1954, p. 100).

[21] C. M. Wilson: Handbook on high velocity air distribution (*Heating, Piping and Air Conditioning*, November 1954, p. 94). Discussions (*Heating, Piping and Air Conditioning*, December 1954, p. 73; February 1955, p. 69; March 1955, p. 99; June 1955, p. 82).

[22] R. W. Waterfill: Air conditioning of multi-room buildings (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 233).

[23] P. B. Gordon: Air conditioning multi-story buildings (*Heating, Piping and Air Conditioning*, April 1955, p. 112). Discussion (*Heating, Piping and Air Conditioning*, May 1955, p. 103).

[24] N. S. Shataloff: Elements of dual duct design and performance (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 257).

[25] N. J. Janisse: How to control high velocity double duct air systems (*Heating, Piping and Air Conditioning*, November 1955, p. 122 and December 1955, p. 100).

[26] E. F. Snyder, Jr.: Self-actuated room control from high speed air (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 295).

[27] *High-velocity Air Distribution* (collected papers read at Symposium at ASHAE 62nd Annual Meeting, January 25, 1956).

[28] C. M. Ashley, S. F. Gilman, and R. A. Church: Branch-fitting performance at high velocity (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 279).

[29] *National Board of Fire Underwriters Standards* (*NBFU Pamphlet* No. 90, p. 21).

BIBLIOGRAPHY

Fluid Flow (See Chapter 4)

Hunter Rouse: *Elementary Mechanics of Fluids* (John Wiley & Sons, Inc., New York, 1946).

J. C. Hunsaker and B. G. Rightmire: *Engineering Applications of Fluid Mechanics* (McGraw-Hill Book Co., New York, 1947).

Clifford McClain: *Fluid Flow in Pipes* (The Industrial Press, New York, 1952).

H. C. Berry: *Flow and Fan—Principles of Moving Air Through Ducts* (The Industrial Press, New York, 1954).

R. J. S. Pigott: The flow of fluids in closed conduits (*Mechanical Engineering*, Vol. 55, 1933, p. 497).

E. Kemler: A study of the data on the flow of fluids in pipes (*ASME Transactions*, Vol. 55, 1933, p. 7).

Flow of Air in Ducts

F. C. Houghten, J. B. Schmieler, J. A. Zalovcik, and N. Ivanovic: ASHVE RESEARCH REPORT No. 1105—Frictional resistance to the flow of air in straight ducts (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 35).

J. B. Schmieler, F. C. Houghten, and H. T. Olson: ASHVE RESEARCH REPORT No. 1154—Analysis of factors affecting duct friction (ASHVE TRANSACTIONS, Vol. 46, 1940, p. 193).

G. R. Whitnah and J. V. Borry: Pressure loss characteristics of small diameter round duct systems (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November 1952, p. 111).

H. G. Conn, W. G. Colborne, and W. G. Brown: ASHVE RESEARCH REPORT No. 1470—Pressure losses in 4-inch diameter galvanized metal duct and fittings (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 139).

Pressure Loss in Elbows

F. L. Busey: Loss of pressure due to elbows in the transmission of air through pipes or ducts (ASHVE TRANSACTIONS, Vol. 19, 1913, p. 366).

Loring Wirt: New data for the design of elbows in duct systems (*General Electric Review*, Vol. 30, June 1927, p. 286).

O. E. Parker: An investigation of pressure losses in air duct elbows (Northeastern University thesis, May 28, 1934).

C. H. McLellan and W. A. Bartlett, Jr.: *Investigation of Air Flow in Right Angle Elbows in a Rectangular Duct* (National Advisory Committee for Aeronautics, *Advanced Restricted Report* L-328, October 1941).

M. C. Stuart, C. F. Warner, and W. C. Roberts: ASHVE RESEARCH REPORT No. 1211—Pressure loss caused by elbows in eight-inch round ventilating duct (ASHVE TRANSACTIONS, Vol. 48, 1942, p. 335).

J. R. Weske: *Experimental Investigation of Velocity Distributions Downstream of Single Duct Bends* (*National Advisory Committee for Aeronautics Technical Note* 1471, January 1948).

J. R. Weske: Investigations of the flow in curved ducts at large Reynolds numbers (*Applied Mechanics Journal*, December 1948, p. 344).

R. K. Guthrie: How much pressure loss in round elbows? (*Heating, Piping and Air Conditioning*, March 1955, p. 130).

R. D. Madison and R. M. Conner: Discussion (*Heating, Piping and Air Conditioning*, April 1955, p. 89).

R. K. Guthrie: Discussion (*Heating, Piping and Air Conditioning*, June 1955, p. 81).

Pressure Loss in Divided-flow Fittings

H. H. Korst, H. A. Buckley, S. Konzo, and R. W. Roose: ASHVE RESEARCH REPORT No. 1392—Fitting losses for extended-plenum forced air systems (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 259).

J. W. Holl, S. F. Gilman, R. J. Martin, and S. Konzo: ASHVE RESEARCH REPORT No. 1430—Pressure losses of take-offs for extended-plenum duct systems (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 419).

L. G. Miller, C. H. Pesterfield, and R. J. Waalkes: Resistance of rectangular divided-flow fittings (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 145).

Duct Design

L. G. Miller: ASHVE RESEARCH REPORT No. 1050—A rational method of duct design (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 71).

Kirby Walker: Modern thinking applied to duct design (*Heating, Piping and Air Conditioning*, March 1949, p. 85; April 1949, p. 104; May 1949, p. 91; July 1949, p. 95; September 1949, p. 96; November 1949, p. 97; March 1950, p. 97; May 1950, p. 94).

Pointers on air conditioning duct design and installation (*Reference Section, Heating and Ventilating*, October 1951).

Peter Franck: Economical duct layouts (*Heating, Piping and Air Conditioning*, December 1952, p. 100).

F. W. Hutchinson: *The Design of Aluminum Duct Systems* (Kaiser Aluminum & Chemical Sales, Inc., Oakland, California, 1954).

R. W. Ruppert: Nomograph simplifies duct design (*HPAC Data Sheet, Heating, Piping and Air Conditioning*, July 1955, p. 127).

V. J. Turecamo: To size ducts adequately select the right pressure drop (*Heating, Piping and Air Conditioning*, April 1956, p. 110).

Miscellaneous

R. H. Heilman and R. A. McArthur: Performance tests of asbestos insulating air ducts (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 197).

CHAPTER 22

FANS

Types, Fan Performance, Fan Laws, Fan Performance Curves, System Characteristics, Fan Arrangements, Fan Control, Motive Power, Fan Selection, Fan Installation, Fan Applications

IN HEATING, ventilating and air-conditioning practice, the devices used to produce air flow are variously known as fans, blowers, exhausters or propellers. The ASME Test Code[1] limits fans to those in which the fluid density change does not exceed 7 percent (one psi at atmospheric pressure) and labels as compressors those devices operating beyond that pressure range. Since air conditioning rarely requires pressures of over ½ psi, all such devices will be known as *fans* and the air will be considered non-compressible.

TYPES

Fans are divided into two general classifications: (1) *centrifugal or radial flow* in which the air flows radially through the impeller within a scroll type housing, and (2) *axial flow* in which the air flows axially through the impeller within a cylinder or ring.

Centrifugal fans are further subdivided into types denoted by the curvature or slope of the impeller blades, the angle of which largely determines the operating characteristics. For a given output, a forward inclination of blade indicates a relatively low speed of operation, and a backward inclination, a relatively high speed of operation. Many intermediate forms are also found.

Axial-flow fans are subdivided into types differentiated mainly by their enclosures and refinements of impellers and appurtenances. All types vary in shape, number and angles of blades; ratios of hub diameter to impeller diameter; materials and methods of fabrication, depending upon design and preference of manufacturer. *Tubeaxial* and *vaneaxial* fans, usually used against appreciable resistance, commonly have relatively large hubs and helical blades (the angle varies radially along the blade). The blades may be of uniform thickness, either flat or cambered, and either cast or made of plates; or they may be of air-foil sections, either cast or of double thickness sheet. Streamlining of both impeller and enclosure is common practice. *Vaneaxial* fans incorporate guide vanes to modify performance and increase efficiency. *Propeller* fans customarily used for free delivery, or against low resistance, also are found with a variety of blade conformations, but are simple in construction. They are merely mounted within a plate or ring.

The fan nomenclature in Fig. 1 has been standardized by the Air Moving and Conditioning Association.[2]

FAN PERFORMANCE

Fan performance is a statement of volume, total pressures, static pressures, speed, power input, mechanical efficiency, and static efficiency, at a stated density. These terms are defined by the Air Moving and Conditioning Association[3] as follows:

1. *Volume* handled by a fan is the number of cubic feet of air per minute expressed at fan outlet conditions.

2. *Total pressure* of a fan is the rise of pressure from fan inlet to fan outlet.

3. *Velocity pressure* of a fan is the pressure corresponding to the average velocity determination from the volume of air flow at the fan outlet area.

4. *Static pressure* of a fan is the total pressure diminished by the fan velocity pressure.

5. *Power output* of a fan is expressed in horsepower and is based on fan volume and the fan total pressure.

6. *Power input* to a fan is expressed in horsepower and is measured horsepower delivered to the fan shaft.

7. *Mechanical efficiency* of a fan is the ratio of power output to power input.

8. *Static efficiency* of a fan is the mechanical efficiency multiplied by the ratio of static pressure to the total pressure.

9. *Fan outlet area* is the inside area of the fan outlet.

10. *Fan inlet area* is the inside area of the inlet collar.

While the total pressure truly represents the actual pressure developed by the fan, the static pressure may best represent the useful pressure for overcoming resistance. In many installations, since the outlet velocity of the fan is greater than the duct velocity, some of the velocity pressure may be utilized by conversion to static pressure within the system. However, due to the uncertainty of the flow at the points of velocity change, the amount of conversion is seldom known and therefore, most fan tables list only the static pressure as available to overcome the system resistance.

According to the Standard Test Code[4] the efficiencies may be determined by the formulas:

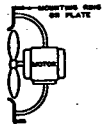
$$\text{Mechanical (total) Efficiency} = \frac{0.0001573 \times (\text{cfm}) \times \text{total pressure (inches water)}}{\text{horsepower input}}$$

$$\text{Static Efficiency} = \frac{0.0001573 \times (\text{cfm}) \times \text{static pressure (inches water)}}{\text{horsepower input}}$$

As the static pressure is often more useful than total pressure, static efficiency is likewise many times more useful than

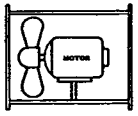
mechanical efficiency. However, where a high outlet velocity can be effectively utilized, the static efficiency fails to be a satisfactory measure of performance. Also when a fan operates against no resistance, the static efficiency becomes zero and is meaningless. Under such circumstances, many engineers prefer to use mechanical efficiency.

Sound developed by a fan is a characteristic which is becoming increasingly important. Unfortunately, no method has yet been devised for accurately measuring the sound actually discharged into a duct system. The ASHAE Research Laboratory, in cooperation with the U. S. Navy, has a program underway seeking to find a method. Many manufacturers list the average sound (for various fan operating



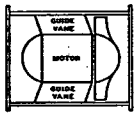
Propeller Fan

A propeller fan consists of a propeller or disc wheel within a mounting ring or plate.



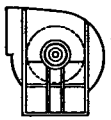
Tubeaxial Fan

A tubeaxial fan consists of an axial-flow wheel within a cylinder.



Vaneaxial Fan

A vaneaxial fan consists of an axial-flow wheel within a cylinder, combined with a set of air guide vanes located either before or after the wheel.



Centrifugal Fan

A centrifugal fan consists of a fan rotor or wheel within a scroll type of housing.

Fig. 1. . . . Names and Definitions of Types of Fans

conditions) measured at seven stations near the fan. These stations, as specified in the AMCA Test Code,[4] are located in a horizontal plane passing through the fan shaft, and are at a distance of one wheel diameter (but not less than 5 ft) from the fan. Such values are useful in comparing the relative sound generated by various types and sizes of fans under comparable operating conditions.

FAN LAWS[5].

The performances of fans of all types follow certain laws which are useful in predicting the effect upon performance of changes in the conditions of operation, the duty required of the installation, or the size of the equipment due to the space, power, or speed limitations. In the following laws, groups 1 to 6, Q = air volume and P = static, velocity, or total pressure. The laws pertaining to fan size apply only to fans geometrically similar, i.e., those in which all dimensions are proportional to some linear dimension denoted as *size*. If the size number is also linearly proportional, it may be used; otherwise, wheel diameter is commonly used as a size criterion.

1. Variation in Fan Speed:
   Constant Air Density—Constant System
   (*a*) Q: Varies as fan speed.
   (*b*) P: Varies as square of fan speed.
   (*c*) Power: Varies as cube of fan speed.

2. Variation in Fan Size:
   Constant Tip Speed—Constant Air Density
   Constant Fan Proportions—Fixed Point of Rating
   (*a*) Q: Varies as square of wheel diameter.
   (*b*) P: Remains constant.
   (*c*) RPM: Varies inversely as wheel diameter.
   (*d*) Power: Varies as square of wheel diameter.

3. Variation in Fan Size:
   At Constant RPM—Constant Air Density
   Constant Fan Proportions—Fixed Point of Rating
   (*a*) Q: Varies as cube of wheel diameter.
   (*b*) P: Varies as square of wheel diameter.
   (*c*) Tip Speed: Varies as wheel diameter.
   (*d*) Power: Varies as fifth power of diameter.

4. Variation in Air Density:
   Constant Volume—Constant System
   Fixed Fan Size—Constant Fan Speed
   (*a*) Q: Constant.
   (*b*) P: Varies as density.
   (*c*) Power: Varies as density.

5. Variation in Air Density:
   Constant Pressure—Constant System
   Fixed Fan Size—Variable Fan Speed
   (*a*) Q: Varies inversely as square root of density.
   (*b*) P: Constant.
   (*c*) RPM: Varies inversely as square root of density.
   (*d*) Power: Varies inversely as square root of density.

6. Variation in Air Density:
   Constant Weight of Air—Constant System
   Fixed Fan Size—Variable Fan Speed
   (*a*) Q: Varies inversely as density.
   (*b*) P: Varies inversely as density.
   (*c*) RPM: Varies inversely as density.
   (*d*) Power: Varies inversely as square root of density.

Examples 1 to 4 illustrate the application of the preceding fan laws.

Example 1: A certain fan delivers 12,000 cfm at a static pressure of 1 in. of water when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation 15,000 cfm are desired, what will be the speed, static pressure, and power?

$$\text{Speed} = 400 \times \frac{15{,}000}{12{,}000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp}$$

Example 2: A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0.075 lb per cubic foot) at a static pressure of 1 in. of water when operating at 400 rpm, and requires 4 hp. If the air temperature is increased to 200 F (density 0.0602 lb) and the speed of the fan remains the same, what will be the static pressure and power?

$$\text{Static pressure} = 1 \times \frac{0.0602}{0.075} = 0.80 \text{ in.}$$

$$\text{Power} = 4 \times \frac{0.0602}{0.075} = 3.20 \text{ hp}$$

Example 3: If the speed of the fan of *Example 2* is increased so as to produce a static pressure of 1 in. of water at at 200 F as at 70 F, what will be the speed, capacity, static power?

$$\text{Speed} = 400 \times \sqrt{\frac{0.075}{0.0602}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \sqrt{\frac{0.075}{0.0602}}$$

$$= 13{,}392 \text{ cfm (measured at 200 F)}$$

$$\text{Power} = 4 \times \sqrt{\frac{0.075}{0.0602}} = 4.46 \text{ hp}$$

Example 4: If the speed of the fan of the previous examples is increased so as to deliver the same weight of air at 200 F as at 70 F, what will be the speed, capacity, static pressure, and power?

$$\text{Speed} = 400 \times \frac{0.075}{0.0602} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.075}{0.0602} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.075}{0.0602} = 1.25 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{0.075}{0.0602}\right)^3 = 6.20 \text{ hp}$$

The fan laws stated may be combined to give other overall values. One useful combination is the product of Laws 1 and 3 which gives the following relations:

Capacity varies as the ratio of size cubed, times the ratio of the rpm.

Pressure varies as the ratio of size squared, times the ratio of the rpm squared.

Horsepower varies as the ratio of the size to fifth power, times the ratio of the rpm cubed.

Example 5: Assuming that a fan with a 36 in. diameter blast wheel will deliver 12,000 cfm at 70 F at 1 in. static pressure, requiring 4.0 brake hp when operating at 400 rpm, what is the capacity, pressure, and horsepower of a homologous fan having a 45 in. wheel at the same speed?

$$\text{Capacity} = \left(\frac{45}{36}\right)^3 \times \left(\frac{400}{400}\right) \times 12{,}000 = 23{,}400 \text{ cfm}$$

$$\text{Static pressure} = \left(\frac{45}{36}\right)^2 \times \left(\frac{400}{400}\right)^2 \times 1 = 1.56 \text{ in.}$$

$$\text{Horsepower} = \left(\frac{45}{36}\right)^5 \times \left(\frac{400}{400}\right)^3 \times 4 = 12.2 \text{ hp}$$

FAN PERFORMANCE CURVES

Fan performance curves are the graphical presentation (for constant speed and air density) of the relation of total pressure, static pressure, power input, and mechanical efficiency, and static efficiency, to actual volume, for the desired range of volumes. Figs. 2, 3 and 4 illustrate performance (sometimes called characteristic) curves of various types of fans.

Centrifugal fans[6] may be roughly divided into three classes: (1) those with the tip of the blades curved forward in the direction of rotation; (2) those with straight radial blades; and (3) those with the tip of the blades inclined backward away from direction of rotation. They are also characterized as slow speed, moderate speed, and high speed types, respectively, although the actual speed range of each may be wide and overlapping. The highest speed type may operate as high as 200 percent of the speed of the lowest speed type, to deliver the same volume of air at the same pressure. The differentiating curvature is always the tip of the blade, since the inlet edge, if inclined, is always curved forward to minimize the shock loss at entrance. Straight radial blades are most frequently found in pressure fans and material-handling fans.

Centrifugal fans produce pressure from two independent sources: (1) from the centrifugal force created by rotating the enclosed air column, and (2) from the kinetic energy imparted to the air by virtue of its velocity leaving the impeller. This velocity in turn is a combination of rotative velocity of the impeller and air speed relative to the impeller. When the blades tip forward, these two velocities are cumulative, and when backward, oppositional. Thus a fan with forward-curved blades depends less on centrifugal force for its pressure, and more on velocity pressure conversion in the scroll, with the result that it may run at relatively low speed. Conversely, a fan having backward-curved blades builds up more of its pressure by centrifugal force (a more efficient form of energy transfer) and less by velocity conversion and, therefore, must run at a higher speed. Likewise, a fan having forward-curved blades will produce the greatest capacity of any type of the same size when operating against no resistance.

Since the energy imparted to the air depends on the velocities,[7] and since the velocities are cumulative with a fan having forward-curved blades, the theoretical energy per pound of air rises rapidly with an increase of air delivery. With the velocities oppositional in the fan having backward-curved blades, the energy per pound of air may decrease, and in a fan having straight blades it is roughly constant. Thus the shape of the horsepower curve definitely identifies the blade angle.

Performance curves of a typical *forward-curved* blade centrifugal fan are shown in Fig. 2. The pressure rises from free delivery toward no delivery, with a characteristic drop at low capacities, because a large share of the pressure is being generated by conversion of velocity, which is small at low capacity. The maximum efficiency occurs at approximately maximum pressure. The horsepower curve reflects the increase in energy by rising rapidly from no delivery to free delivery. The sound is a minimum at maximum efficiency, and rises toward free delivery as the velocities increase.

Performance curves of a typical *backward-curved* blade centrifugal fan are shown in Fig. 3. The pressure is constantly rising from free delivery nearly to point of no de-

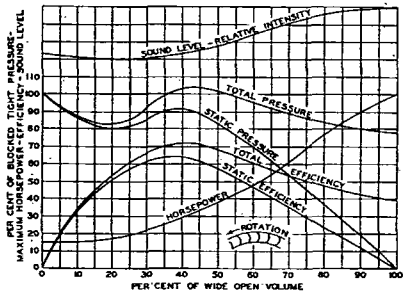


Fig. 2.... Percentage Performance Curves of a Forward-Curved Blade Centrifugal Fan

livery. The horsepower reflects the energy-velocity relationship by rising to a maximum value as the capacity increases, and then decreasing with further increase in capacity to give a self-limiting horsepower characteristic. The maximum horsepower coincides approximately with the maximum efficiency. The sound is again a minimum near maximum efficiency, but is little or no higher at free delivery than at low capacities.

Between the extremes of forward and full-backward-curved blades, there exists a number of intermediate designs which show varying degrees of similarity to the curves in Figs. 2 and 3. A common variation is a fan having modified backward, single, or double-curved blades and equipped with fixed inlet vanes. Such vanes applied to a partially backward-curved impeller give the steep, constantly rising pressure characteristic, and the self-limiting horsepower feature of the full-backward-curve impeller. They also stabilize the flow entering the impeller when adverse flow conditions exist in the approach to the inlet.

Axial-flow fans develop none of their static pressure by centrifugal force, but all from the change in velocity in passing through the impeller, and its conversion into static pressure. They are thus inherently high-velocity fans, and are very dependent on blade conformation for good characteristics. For that reason, an air-foil section, such as developed in wind tunnels for aircraft work[2] is frequently used. Since any shape of blade can only be correct for a narrow range of capacity at constant speed, the performance curves for any blade show definite characteristics. To absorb energy, the air must be given a tangential motion in passing the impeller,[1] and when operating against higher pressures, must have guide vanes (see vaneaxial fans) to obtain best efficiencies.

While *axial-flow fans* are inherently a higher capacity type than centrifugal fans, they, too, may be designed with widely varying characteristics. As with a centrifugal fan, the pressure rises generally from free delivery to no delivery, but tubeaxial and vaneaxial fans may have a drop in pressure when the capacity decreases below a certain volume, a condition also found in the case of the centrifugal fan having forward-curved blades. The pressure drop is caused by the same condition for both fans, i.e., the static pressure is largely dependent on conversion of velocity pressure, and velocity pressure is small at low capacity. Tubeaxial and vaneaxial fans may also have performance curves resembling somewhat those of a centrifugal fan with backward-curved blades. Fig. 4 shows the performance curves for a typical design.

The horsepower curve may be flat with a self-limiting characteristic as in a backward-curved blade centrifugal fan, or it may have a generally downward trend from no delivery to free delivery with the maximum at no delivery, contrary to that of a centrifugal fan. The type of guide vanes in a vaneaxial fan has a distinct bearing on the shape of the horsepower curve. The maximum efficiency tends to occur at a percentage of free delivery capacity higher than for a centrifugal fan.

The sound curve, which may have a minimum value comparable to centrifugal fans, is again lowest near maximum efficiency, but has a characteristic rise when the fan is operating at low capacities and the *stall* point of the blade section is reached.

Since *propeller fans* are designed for operation near free delivery, less attention is given the regaining of velocity to static pressure, and the pressure curve rises constantly from free delivery to no delivery. The horsepower is highest at no delivery, and decreases toward free delivery, in contrast to a centrifugal fan. Maximum total efficiency is obtained at a higher percentage of free delivery than for other types.

SYSTEM CHARACTERISTICS

Any ventilating system consisting of ductwork, heaters, air washers, filters, etc., has a system characteristic which is individual to that system, and is independent of any fan which may be applied to the system. This characteristic may be expressed in curve form in exactly the same manner that fan characteristics may be shown. Typical system characteristic curves are shown as *A*, *B*, and *C* in Fig. 5. These curves are drawn to follow the simple parabolic law in which the static pressure or resistance to flow of air varies as the square of the volume flowing through the system. Heating and ventilating systems follow this law very closely and no serious error is introduced by its use.

When the characteristic curve of a constant-speed fan of a given size is superimposed upon a system characteristic curve, the relation between the two is at once apparent. The only point common to the two curves is the point at the intersection of the system characteristic curve and the fan characteristic curve, and it is at this point that the combination will operate. In Fig. 5, system characteristic curves *A*, *B*, and *C* cross the fan characteristic curve at points *X*, *Y*, and *Z*. The fan whose curve is shown, when applied to systems having characteristic curves *A*, *B*, and *C*, will deliver 10,000, 13,000 or 16,400 cfm, respectively.

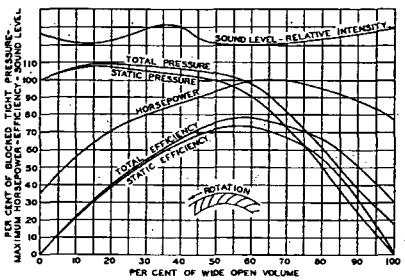


Fig. 3.... Percentage Performance Curves of a Backward-Curved Blade Centrifugal Fan

The curves in Fig. 5 also illustrate the effect of errors that may be made in calculating the resistance of a ventilating system. For instance, if a given system requires 13,000 cfm, and the resistance to flow of the system has been computed as 1.25 in. static pressure, such a system would be represented by system characteristic curve *B* in Fig. 5. If a 100 percent error had been made and the resistance were 2.5 in. instead of 1.25 in., then the system characteristic would be as shown in curve *A*, and would cross the fan curve at 10,000 cfm. Such an error would cause the flow of air to be decreased from a design volume of 13,000 cfm to 10,000 cfm. If the resistance to flow had been overestimated and the resistance actually were 0.625 in., the system characteristic curve would be as shown in curve *C*, and the fan would deliver 16,400 cfm to the system instead of the design volume of 13,000 cfm.

In this example, extreme errors have been selected to emphasize the effect the square function of the system characteristic has in maintaining the fan performance within comparatively narrow limits. In the first example, a system estimated at half what it should have been, resulted in a drop of 23 percent in volume; and in the second example, a system estimated at twice what it should have been resulted in an increase of 26 percent in volume.

In some instances fans may be applied to variable-flow systems. In such cases, the limiting systems may be plotted and the effect on fan performance examined. For instance, a system might have a characteristic curve between *A*, shown in Fig. 5, as one limit, and *B* as the other limit. The fan performance will then fall between points *X* and *Y* on the fan curve at a point determined by the system characteristics at that particular time. If *A* and *B* are the limiting characteristic curves of the systems, the fan performance will never be outside the points *X* or *Y*.

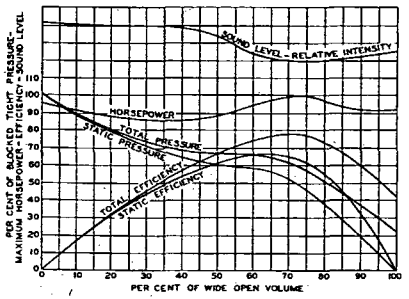


Fig. 4.... Percentage Performance Curves of an Axial-Flow Fan

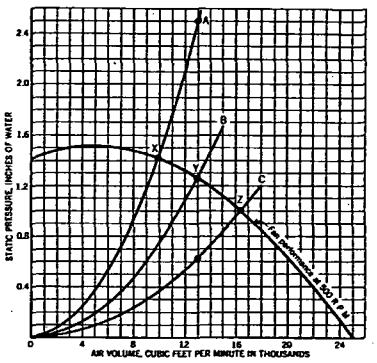


Fig. 5.... Parabolic System Characteristic Curves

FAN ARRANGEMENTS

Centrifugal fan arrangements have been standardized by the Air Moving and Conditioning Association. Figs. 6, 7, and 8 show the accepted designation as to arrangement of drive, rotation, discharge, and motor position, for belt drive. Axial-flow fans are either belt driven or direct connected, in accordance with individual manufacturer's arrangements. Usually a choice of antifriction or sleeve bearings is available.

FAN CONTROL

In some heating and ventilating systems it is desirable to vary the volume of air handled by the fan. This may be accomplished by a number of methods. Where the change is made infrequently, the pulley or sheave on the driving motor or fan may be changed to vary the speed of the fan and alter the air volume. Dampers may be placed in the duct system to vary the volume. Variable-speed pulleys or transmissions, such as fan belt change boxes, or electric or hydraulic couplings, may be used to vary the fan speed. Variable-speed motors and variable inlet vanes on fans may also be used to adjust the fan volume. All of these methods will give control. From a power consumption consideration, a reduction of fan speed is most efficient. Inlet vanes save some power, while dampers save the least. From consideration of first cost, dampers usually are the lowest in cost. In some installations, adjustments of volume are desirable at various times during the day, or continuously. In others, an increased supply of air in summer, over that needed in winter, is demanded. The demands in each case will dictate which type of control is most desirable. Where noise is a factor, a lowering of fan speed, if possible, is preferred as a control means, because of the resulting reduction in sound level.

In addition to the above types of control, *tubeaxial* and *vaneaxial* fans are sometimes made with adjustable blades to permit balancing of the fan against the system, or making seasonal adjustment.

MOTIVE POWER

Heating, ventilating, and air-conditioning fans are usually driven by electric motors, although other prime movers may be used. The small sizes of fans, and especially those operating in the higher speed range, are equipped with direct-connected motors. For larger size fans, and those operating at lower speed, V-belt drives are generally used.

In selecting the size of motor for operating a fan, it is advisable to select at least the standard size next larger than the fan requirements. Direct-connected motors do not require so great a safety factor as belted units. Justification for liberal power provision exists only in systems where it is possible that larger volumes of air may be required at intervals, and made available by use of bypass dampers, thus greatly reducing the system resistance. If such a system includes a fan with forward-curved blades, it would be necessary that the motor be sized for the maximum volume and duty. If such a system includes fans with backward-curved blades, the volume peak would not make it necessary to provide additional motor power. In selecting fans for such a system, sound ratings should be given careful consideration.

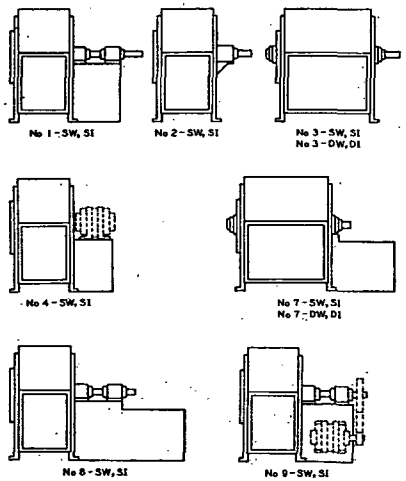


Fig. 6 .... Arrangement of Fan Drives

Arr. 1, SW, SI. For belt drive or direct connection. Wheel overhung. Two bearings on base.

Arr. 2, SW, SI. For belt drive or direct connection. Wheel overhung. Bearings in bracket supported by fan housing.

Arr. 3, SW, SI. For belt drive or direct connection. One bearing on each side and supported by fan housing. Not recommended in sizes 27 in. diameter wheel and smaller.

Arr. 3, DW, DI. For belt drive or direct connection. One bearing on each side and supported by fan housing.

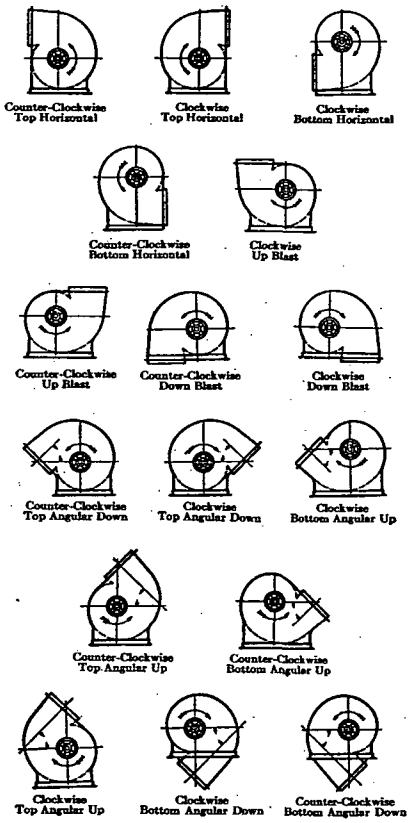
Arr. 4, SW, SI. For direct drive. Wheel overhung on prime mover shaft. No bearings on fan. Base or equivalent for prime mover.

Arr. 7, SW, SI. For belt drive or direct connection. Arrangement No. 3 plus base for prime mover. Not recommended in sizes 27 in. diameter and smaller.

Arr. 7, DW, DI. For belt drive or direct connection. Arrangement No. 3 plus base for prime mover.

Arr. 8, SW, SI. For belt drive or direct connection. Arrangement No. 1 plus base for prime mover.

Arr. 9, SW, SI. For belt drive. Arrangement No. 1 designed for mounting prime mover on side of base.



Note: Direction of Rotation is determined from the drive side for either single or double width or single or double inlet fans. (The driving side of a single inlet fan is considered to be the side opposite the inlet regardless of the actual location of the drive.) For fan inverted for ceiling suspension, direction of rotation and discharge is determined when fan is resting on floor.

Fig. 7 .... Designation of Direction of Rotation and Discharge

Where a system is constant, and has no provision for volume change that would materially reduce the resistance, and when the resistance calculations are reasonably precise, there is no necessity for too liberal a motor allowance (even where fans with forward-curved blades are used) if the fan has been properly selected. Fig. 5 shows that the system resistance varies as the square of the volume, and that the fan static pressure varies approximately inversely as the volume, thus greatly offsetting the trend toward both increase in air delivery and motor load. Reference to Fig. 5 indicates that there is no justification for allowing large spare motor capacity. It is generally more economical to operate motors well loaded.

Since the power consumption of fans varies as the cube of the speed, very little starting torque is required of the motor. Refer to Chapter 45 for characteristics of various types of motors.

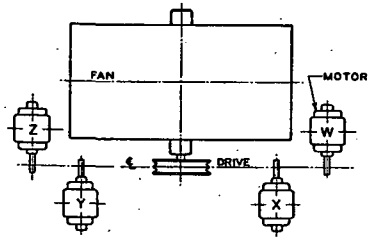
FAN SELECTION

The following information is required to select the proper *type and size of fan*:

1. Capacity in cubic feet per minute.
2. Static pressure or system resistance.
3. Air density if other than standard.
4. Type of application or service.
5. Arrangement of system.
6. Prevailing sound level or use of space served.
7. Nature of load.
8. Type of motive power available.

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressures: (1) volume of air in cubic feet per minute (68 F, 50 percent relative humidity, 0.075 lb per cubic foot); (2) outlet velocity; (3) revolutions per minute; (4) brake horsepower; (5) tip or peripheral speed; and (6) static pressure. The most efficient operating point is usually shown by either boldface or italicized figures in the capacity tables.

Often the service determines the type of fan. When operation occurs with little or no resistance, and particularly without a duct system, the propeller fan is indicated for convenience and low cost. When resistance is low the power required is low, and efficiency becomes of secondary importance. When a duct system is involved the choice is usually made between a centrifugal fan and a tubeaxial or vaneaxial. At times the capacity-pressure-speed relationship (specific speed)[9] dictates a choice. Usually, space, efficiency, sound, cost, and serviceability must all be considered.[10] In general, centrifugal and axial fans are comparable in efficiency and sound, but the latter are lighter and require considerably less space, especially if arranged for straight-through operation. The comparison cannot be made on the cost of fans only, but the difference in cost of ductwork, mounting, and servicing must be included. A vaneaxial is more efficient and quieter than a tubeaxial, but is more expensive, and frequently requires more space. While requiring less space than the centrifugal, the axial flow fan is inherently less accessible for service. When high-temperature air or air containing corrosive elements is being conveyed, motors and bearings should be located outside of the air stream. This requirement may determine the type of fan to be used. Where the system resistance is indefinite or variable, the pressure, horsepower, and noise characteristics of centrifugal fans usually indicate their selection. Under such conditions, a steep and constantly rising pressure curve permits less variation in air delivery when the resistance varies. Likewise, a flat sound curve minimizes the change of moving into a region of increased noise. A fan having a high effi-



Location of motor is determined by facing the drive side of fan or blower and designating the motor position by letters W, X, Y, or Z as the case may be.

Fig. 8 .... Motor Position, Belt or Chain Drive

Table 1 .... Good Operating Velocities and Tip Speeds for Ventilating Fans

Static Pressure Inches of Water	Forward Curved Blade Fans		Backward Tipped and Double Curved Blade Fans		Tubeaxial and Vaneaxial Fans
	Outlet Velocity Fpm	Tip Speed Fpm	Outlet Velocity Fpm	Tip Speed Fpm	Wheel Velocity* Fpm
¼	800–1100	1300–1500	800–1100	2500–3100	1100–1500
⅜	800–1250	1600–1850	800–1250	3000–3750	1250–1700
½	900–1450	1850–2150	900–1450	3400–4250	1400–1900
⅝	1000–1600	2050–2350	1000–1600	3800–4750	1500–2100
¾	1100–1750	2250–2600	1100–1750	4150–5200	1650–2350
⅞	1200–1900	2450–2850	1200–1900	4500–5600	1800–2500
1	1250–2000	2600–3000	1250–2000	4800–6000	1900–2700
1¼	1400–2300	2900–3350	1400–2300	5350–6700	2150–3000
1½	1550–2500	3200–3700	1550–2500	5900–7400	2350–3300
1¾	1700–2700	3400–3950	1700–2700	6350–7950	2500–3550
2	1800–2900	3700–4300	1800–2900	6800–8500	2700–3800
2¼	1900–3050	3900–4500	1900–3050	7200–9000	2900–4050
2½	2000–3200	4100–4750	2000–3200	7600–9500	3100–4300
3	2200–3500	4500–5200	2200–3500	8300–10500	3300–4600
3½	2400–3800	4900–5650	2400–3800	9000–11500	
4	2550–4050	5200–6000	2550–4050	9600–12000	
4½	2700–4300	5500–6350	2700–4300	10000–12500	
5	2850–4550	5800–6700	2850–4550	10500–13000	
5½	2950–4750	6100–7050	2950–4750	11000–13750	
6	3100–5000	6400–7400	3100–5000	12000–15000	
7	3350–5400	6900–7950	3350–5400	12700–16000	
8	3600–5750	7350–8500	3600–5750	13500–17000	
9	3800–6100	7800–9000	3800–6100	14500–18000	
10	4000–6400	8200–9450	4000–6400	15000–18500	

ciency over a wide range is more desirable than one which reaches an even higher maximum efficiency, but decreases more rapidly on either side of a narrow range. A self-limiting horsepower curve may permit more accurate selection of motor size.

The selection of size of fan usually involves balancing cost and space against sound and efficiency. Unless the pressure involved is so high that a smaller fan running at greater speed requires a higher class[2] of construction, the smallest fan is the cheapest in first cost of the fan only. However, as the cost of the driving equipment is also involved in the total installation cost, fan efficiency must be considered for that reason as well as its bearing on the cost of operation. In some cases where large fans operate long hours per year, selection at absolute maximum efficiency is indicated. Generally, however, the power saving by selecting for optimum efficiency does not justify the extra cost, and a slightly smaller fan gives the best balance of cost and efficiency. Reference to Figs. 2 to 4 shows that all types tend to have a minimum sound near maximum efficiency. When noise is a consideration, therefore, selection approaching maximum efficiency is indicated. Too large a fan may not only cause an unnecessary investment and an increased power consumption and sound, but may also give faulty performance if it is of a type having an unstable pressure characteristic at low capacity.

Table 1 shows the outlet velocities and impeller tip speeds recognized as good practice for various static pressures for centrifugal, tubeaxial, and vaneaxial fans applied to average heating, ventilating, and air-conditioning applications. Fans for churches, schools, residences, and other buildings having a low prevailing noise level should be selected for lower than average outlet velocities.

FAN INSTALLATION

In designing heating, ventilating, and air-conditioning systems, the characteristics of the fans available for use therewith should not be ignored. If double inlet fans or multiple fans in parallel are used, care must be taken that both inlets have the same free area and general approach conditions.

The dimensions of the ductwork and the size of the various devices whose individual resistances determine the static pressure, dictate the fan selection. Often a minor modification of the system may permit use of a smaller motor, and even a lower class[2] of fan, with considerable saving of cost. Invariably, the sound generated is affected, as fans operating at high pressure produce more noise than at lower pressure (see Chapter 25). Minimizing the resistance may be the best insurance against noise. On the other hand, sometimes the lowest overall cost results from selecting the minimum size of system equipment, and then installing adequate acoustical and vibration treatment.

All ducts should be connected to fan outlets and inlets by means of unpainted canvas or other flexible material. Access should be provided in the connections for periodic removal of any accumulations tending to unbalance the rotor. When operating against high resistance, or when ambient noise levels are low, it is preferable to locate the fan in a room removed from occupied areas or acoustically treated to prevent sound transmission. The lighter building constructions which are common today, make it desirable to mount fans and driving motors on resilient bases designed to prevent transmission of vibrations through floors to the building structure. Conduits, pipes, and other rigid members should not be attached to fans. Noises due to high velocities, abrupt turns, grilles, and other items not connected with the fan, may be present. Treatment of such problems, as well as the design of sound and vibration absorbents, are covered in Chapter 25.

FAN APPLICATIONS

Many fan applications and the corresponding types of fan commonly used are listed in the following paragraphs. Reference is also made to the chapters where the applications are discussed.

Central-system supply fans (Chapter 19) are usually of the centrifugal type, since this application requires a wide range of satisfactory and quiet operation against high pressures. They can readily be connected to apparatus of large cross-section on the inlet side, and to relatively small ducts on the outlet side.

Comparative sizes have been standardized among manufacturers,[3] and most rating tables cover a range of 700 to 500,000 cfm, and static pressures from ¼ to 15 in. of water.

Central-system exhaust fans are predominately centrifugal, but the space conservation of the axial is being increasingly utilized. Tubeaxial and vaneaxial fan sizes are not yet standardized, but several manufacturers list capacities from 2000 to 125,000 cfm, and static pressures up to 3 in. of water.

Exhaust fans are found in all types. *Wall fans* are predominantly of the propeller type, since they operate against little or no resistance. They are listed in capacities from 1000 to 75,000 cfm. They are sometimes incorporated in factory-built penthouses or roof caps, or are provided with matching automatic louvers. *Hood-exhaust fans* (Chapter 52) involving ductwork, are predominantly centrifugal, especially if handling hot, corrosive, or erosive fumes, where it is best to keep the bearings and drive remote from the air stream. Otherwise, axial fans are applicable, and where little or no ductwork is involved, propeller fans are suitable. *Spraybooth-exhaust fans* (Chapter 52) are frequently centrifugal, especially if built into self-contained booths. Tubeaxial fans lend themselves particularly well to this application, where ease of cleaning and of suspension in a section of ductwork are advantageous. For such application built-in cleanout doors are desirable. *Material handling fans* (Chapter 52) are always straight radial (or modified) blade centrifugal type. They are of heavier construction, and have fewer blades and greater clearances than ventilating fans. Many characteristics are compromised to provide wear resistance and ease of maintenance. They are commonly listed in capacities from 600 to 125,000 cfm, and static pressures up to 18 in. water.

Mine-fan applications[11] vary greatly and require fans ranging from small portable units for local ventilation, to immense slow speed centrifugal fans, often steam-engine-driven, for general or emergency ventilation. Vaneaxial fans are well suited to mine ventilation. For underground location, their compactness saves on cost of excavations, and above ground, their ready reversibility is valuable in emergencies, even if reversal causes a reduction in capacity.

Marine fans (Chapter 48) are both centrifugal and vaneaxial types. The latter are particularly well adapted to both combatant and noncombatant ships, where compactness and light weight are invaluable.

Unitary systems, i.e., unit heaters, unit ventilators, unit humidifiers, unit air conditioners, unit air coolers, and unit evaporative condensers are equipped with centrifugal or propeller fans, the latter usually being limited to the relatively small suspended type where no ductwork is involved. Fans for units having considerable internal or possible external resistance, are mostly of the forward-curved blade, or so-called mixed-flow centrifugal type. The latter is really a centrifugal type with axial inlets, having a pressure curve resembling a backward-curved-blade centrifugal fan. Both of these types have the high capacities (in relation to displacement) requisite for a compact unit. Ratings are frequently given for these units as separate fans, as well as in conjunction with the various internal resistances. In multiple units on a common shaft they are listed up to 40,000 cfm capacities.

Cooling tower fans (Chapter 40) are predominantly of the propeller type, but axial types are also used for packed towers, and occasionally a centrifugal fan is used to supply forced draft.

Circulating fans are invariably of propeller or disk type, and are made in a vast variety of blade shapes and arrangements. They are designed for pleasing appearance, as well as utility.

General purpose fans are centrifugal fans of conventional design, built for service in the lower capacity ranges. They are built with the fan wheel mounted on the motor shaft, or connected to a self-contained belt-driven arrangement. They are listed in capacities from 100 to 20,000 cfm, and static pressures up to 1½ in. water.

Kitchen fans for domestic use are small propeller fans arranged for window or wall mounting, and with various useful fixtures. Their capacities range from 300 to 1200 cfm.

Attic fans are used during the warm seasons to draw large volumes of outdoor air through a house or other building whenever the indoor temperature exceeds the outdoor, and thereby utilize the cooling effect of the relatively cool evening or night air. Research by the ASHAE[12] indicates that a two to three-minute air change is desirable in the North while in the South, a one-minute change is recommended to provide the additional cooling effect of air motion.

Fans may be centrally located in an attic or other unused space, or in a hallway, and arranged to draw proportionately from several rooms; or local window units may be installed in a single room. Central units draw from the living quarters and discharge into the attic, whence the air escapes through windows or grilles; or the air may be drawn through grilles into the attic with the fan discharging directly outdoors. Discharge openings on the lee side are preferred.

Attic fans are usually of propeller type, and should be selected to operate at low velocities to minimize noise. Noise is more of a problem on local units, but care should be taken to prevent transmission of noise or vibration on all installations, because they operate during sleeping hours. Central units are available in sizes from 3,000 to 30,000 cfm, and window units up to 8,000 cfm.

Fans For Special Applications

Fans are used to handle many gases other than air at normal temperature. These range from heated air to many types of fumes, vapors, and industrial gases which may be corrosive, explosive, toxic, radio active, or merely noxious.

For handling corrosive gases fans should be constructed of a material suitable for the particular gas being handled, although at times it is economical to plan on replacing a unit of standard material at more frequent intervals. Explosive combinations usually dictate a non-sparking fan in which either the fan, or only those parts which might strike together if not adjusted, are built of non-sparking materials. Noxious, toxic, radio-active, or pure or valuable gases call for special construction to prevent leakage. This usually consists of a welded gas-tight housing, using flanged inlet and outlet, and some form of shaft seal.

In most of these special applications, since it is desirable to keep all bearings, drives and motors outside the gas stream, centrifugal fans are usually used and in arrangements 1, 2, 4, or 8 (Fig. 6). The exception is in the case of fans handling explosive mixtures, where propeller or axial fans are permissible when built with explosion-proof motors and non-ferrous wheels.

Fans handling hot air or gas are generally of the centrifugal types and follow arrangements 1 or 8 so that bearings, drives and motors are remote from the heat. If of double-inlet type, they have inlet boxes and long shafts to keep the bearings outside the stream. Propeller or axial types are rare when the temperature exceeds 150 F. With temperatures above 600 F special heat-resistant materials may be required for the rotors. In addition to a remote location for the bearings, external cooling is frequently required when the temperature exceeds 200 to 250 F. This is often obtained by use of air-cooled or water-cooled jackets. With anti-friction bearings a head-radiating disc or a small circulating impeller between the fan and the nearest bearing is sometimes sufficient. Oil lubrication rather than grease lubrication is commonly recommended.

REFERENCES

[1] ASME *Test Code for Fans* (American Society of Mechanical Engineers, PTC 11-1946).

[2] Standards, Definitions and terms in use by the fan and blower industry (*National Association of Fan Manufacturers Bulletin* No. 110, 1952, p. 2).

[3] Standard test code for centrifugal and axial fans (ASHVE and the National Association of Fan Manufacturers, *NAFM Bulletin* No. 110, 1952, p. 18).

[4] Sound measurement test code for centrifugal and axial fans (*NAFM Bulletin* No. 110, 1952, p. 33). W. H. Hoppmann II and Fred Lager: Noise ratings of ventilating fans (ASHVE TRANSACTIONS, Vol. 51, 1945, p. 271).

[5] R. D. Moyer: Fan laws simplify performance calculations (*Power*, August 1946).

[6] F. L. Busey: The centrifugal fan (ASHVE TRANSACTIONS, Vol. 21, 1915, p. 43).

[7] S. A. Moss, C. W. Smith, and W. R. Foote: Energy transfer between a fluid and a rotor for pump and turbine machinery (*ASME Transactions*, 1941).

[8] E. N. Jacobs, K. E. Ward, and R. M. Pinkerton: *The Characteristics of 78 Related Airfoil Sections from Tests in the Variable Density Wind Tunnel* (*National Advisory Committee for Aviation Report* No. 460). R. M. Pinkerton and Harry Greenberg: *Aerodynamic Characteristics of a Large Number of Airfoils Tested in the Variable Density Wind Tunnel* (*National Advisory Committee for Aeronautics Report* No. 678).

[9] M. C. Stuart and J. B. Lusk: The specific characteristics of fans (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 57).

[10] W. R. Heath and A. E. Criqui: The axial flow fan and its place in ventilation (ASHVE TRANSACTIONS, Vol. 50, 1944).

[11] J. J. Walsh: Mine ventilation (ASHVE TRANSACTIONS, Vol. 23, 1917, p. 659). D. Harrington: Mine ventilation and its relation to health and safety (ASHVE TRANSACTIONS, Vol. 51, 1945, p. 243).

[12] G. B. Helmrich and G. H. Tuttle: Comfort cooling with

attic ventilating fans (ASHVE TRANSACTIONS, Vol. 40, 1934, p. 155). A. P. Krats and S. Konzo: ASHVE RESEARCH REPORT No. 979—Study of summer cooling in the research residence for the summer of 1933 (ASHVE TRANSACTIONS, Vol. 40, 1934, p. 167). W. A. Hinton and A. F. Poor: ASHVE RESEARCH REPORT No. 1198—The effect of attic fan operation on the cooling of a structure (ASHVE TRANSACTIONS, Vol. 48, 1942, p. 145). W. H. Badgett: *The Installation and Use of Attic Fans (Agricultural and Mechanical College of Texas Bulletin* No. 52, 1940). W. A. Hinton and W. G. Wanamaker: Some effects of attic fan operation on comfort (ASHVE TRANSACTIONS, Vol. 50, 1944, p. 371).

BIBLIOGRAPHY

A. H. Church: *The Centrifugal Pumps and Blowers* (John Wiley & Sons, New York).

Fan Engineering (Buffalo Forge Co.).

Theodore Baumeister, Jr.: *Fans* (McGraw-Hill Co., New York).

Curt Keller: *The Theory and Performance of Axial Flow Fans*, adapted by L. S. Marks and J. R. Weske (McGraw-Hill Co., New York).

CHAPTER 23

AIR HEATING AND COOLING COILS

Uses for Coils; Coil Construction and Arrangement: Steam Coils, Water Coils, Direct-Expansion Coils, Flow Arrangement, Applications; Coil Selection: Heating, Cooling, and Dehumidifying Coils; Heat Transfer and Air-Flow Resistance; Performance of Heating and Dry Cooling Coils; Overall Coefficient of Heat Transfer; Performance of Dehumidifying Coils; Determining Refrigeration Load

COILS described in this chapter are used for heating or cooling an air stream under forced convection. Surface-coil equipment may be made up of a number of banks assembled in the field, or the entire assembly may be factory constructed. The applications of each type of coil are limited to the field within which it is rated. Other limitations are imposed by code regulations, by proper choice of materials for the fluids used and the condition of the air handled, or by an economic analysis of the possible alternates for each installation.

USES FOR COILS

For *heating service*, coils are used as tempering coils, preheaters, reheaters, or booster heaters. The function of the coils is air heating only, but the apparatus assembly may include means for humidification and air cleaning. Steam or hot water are the usual heating media, although others are used in special cases, such as reheating by means of discharge gas from a refrigerating system.

Coils are used for *air cooling* with or without accompanying dehumidification. Examples of cooling applications without dehumidification are precooling coils using well water or other relatively high temperature water to reduce the load on the refrigerating machinery, or water-cooled coils removing sensible heat in connection with chemical moisture-absorption apparatus. A major portion of coil equipment is designed to handle both sensible cooling and dehumidification. The assembly usually includes air cleaning means to protect the coil from accumulation of dirt, and to keep dust and foreign matter out of the conditioned space. Although cooling and dehumidification are the usual functions, cooling coils are additionally and purposely wetted to aid in air cleaning and odor absorption.

The usual cooling media used in *surface coils* are cold water or Group I (ASA Classification) refrigerants, but others are used in special cases. Brines are seldom required for the range of applications covered by this chapter, although there are cases where low entering-air temperatures with large latent heat loads require a refrigerant temperature so low that use of water becomes impracticable. Sometimes, also, brine from an industrial system already installed is the only convenient source of refrigeration.

For combined cooling and dehumidifying, surface coils present an alternate to spray dehumidifiers. For many applications it is possible, by proper selection of apparatus, choice of air velocities, refrigerant temperatures, etc., to perform the same duty with either. In a few cases both sprays and coils are used. The coils may then be installed within the spray chamber, either in series with the sprays or below them. In making the selection between spray and surface dehumidifiers, certain advantages of each should be considered. The fact that a spray dehumidifier is usually designed to deliver nearly saturated air, tends to simplify the control problem. In this case the dry-bulb temperature is also the dew point, and hence, a dew-point control can be arranged by using a simple duct thermostat. Spray dehumidifiers have an advantage over unwetted coils of obtaining more air cleaning and odor absorption. On the other hand, coils make possible a closed and balanced cooling-water circuit, obviating the unbalanced pumping head, the complication of water-level control, and danger from possible floods incidental to multiple-spray dehumidifiers, especially if located on different levels. The use of coils often makes it possible for the same surface to serve for summer cooling and winter heating by circulating cold water in one season and hot water in the other, with consequent saving in apparatus and piping. Another advantage is that where the surface-coil system can be used with direct expansion of refrigerant, it is comparatively low in initial and operating costs. For comfort-conditioning applications, the regulations of the applicable national and local safety codes should be consulted by the design engineer.

COIL CONSTRUCTION AND ARRANGEMENT

Coils are basically of two types, those consisting of plain tubes or pipe, and those having *extended surfaces*. The former are little used for the applications covered by this chapter, but are often employed where conditions cause frost accumulation, and for cooling within spray dehumidifiers.

The heat transmission from air passing over a tube to a fluid flowing within it is impeded by three resistances. The first is that from the air to the surface of the tube and is usually called the outside surface resistance or air-film resistance. The second is the resistance to the conduction of heat through the metal itself. Finally there is another surface or film resistance to the flow of heat between the inside surface of the metal and the fluid in the tube. For the applications under consideration both the resistance of the metal wall to heat conduction, and the inside surface or film resistance are usually low as compared with the air-side surface resistance. Economy in space, weight, and cost makes it advantageous to decrease the external surface resistance, where it is proportionately large, to approach that of the tube wall, and that from the tube to refrigerant. This may be accomplished by increasing the external surface by means of fins. Sometimes water spray is applied to the same type surface as would have been used without it. The overall heat transfer is not necessarily

increased much, but the water spray may serve other purposes than to increase the flow of heat, such as air and coil cleaning.

In fin or *extended-surface* coils the external surface of the tubes is known as *primary*, and the fin surface is called *secondary*. The primary surface consists generally of round tubes or pipes which may be staggered, or in some cases placed in line with respect to the air flow. The staggered arrangement is usually preferred because it obtains a somewhat higher heat transfer value. Numerous types of fin arrangement are used, the most common of which are spiral, flat, and flat-crinkled or corrugated, all as shown in Fig. 1. While

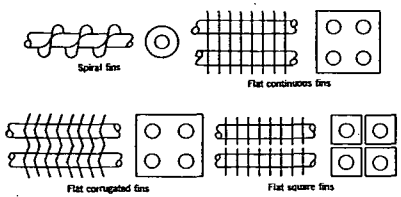


Fig. 1 . . . . Types of Fin-Coil Arrangement

the spiral fin surrounds each tube individually in all cases, the flat types may be continuous (including several rows of tubes), or they may be round or square, with individual fins for each tube. All of these, as well as other less common types, are in use, the selection for a particular installation being based on economic considerations, space requirements, and resistances of individual designs of coils. A most important factor in the performance of extended-surface coils is the bond between the fin and the tube. An intimate contact between the tube and the fin must be maintained permanently in order to assure a continuation of rated performance after the heating units have been in service for a period of time. In some coils, fins are wound on the tubes under pressure, in order to upset the metal slightly at the fin root, and are then given a coating of solder while the fin and tube are still revolving, for the purpose of assuring a uniform coating of solder. In other types, the spiral fin may be knurled into a shallow groove on the exterior of the tube. The tube may be expanded after the fins are assembled, or the tube-hole flanges of a flat or corrugated fin may be made to override those in the preceding fin and so compress them upon the tube. There are also types of construction where the fin is formed out of the material of the tube itself.

For *heating coils*, materials most generally used are copper and aluminum. Steel is occasionally used where sodium or calcium chloride brine is circulated in the tubes. Aluminum fins on copper tubes are common construction. Generally speaking, brass does not serve as a satisfactory fin material because of corrosion difficulties. *Cooling coils* for water or for volatile refrigerants most frequently have aluminum fins and copper tubes, although copper fins on copper tubes are also used. There are many makes of heating and cooling coils of the light-weight extended-surface type for both heating and cooling with tubes commonly ⅜, ½, ⅝, ¾, and 1 in. outside diameter, and with fins spaced three per inch up to fourteen per inch. The tube spacing generally varies from about 1 to 2½ in. on centers, depending upon the width of individual fins and on other considerations of performance. Fin spacing should be chosen for the duty to be performed, with special attention being paid to possibility of lint accumulation and, especially in lower temperature dehumidifying, the consideration of frost accumulation.

Steam Coils

For proper performance of steam heating coils, condensate and air must be continuously eliminated and the steam must be evenly distributed to the individual tubes. This distribution is usually accomplished by individual orifices in the tubes, by distributing plates and orifices in the steam header, or by perforated internal steam-distributing pipes extending into the individual tubes. The latter arrangement has the advantage of distributing the steam throughout the length of each tube, and is conducive to uniform temperature of delivered air. The tendency of condensate to freeze at the bottom of the coil with cold entering air and light heating loads, is also minimized. This is especially valuable for outdoor air preheaters.

Water Coils

The performance of water coils, for heating or cooling, depends on the elimination of air from the system and proper distribution of water. Air elimination is taken care of in the system piping as described in Chapter 28. To assure a pressure drop sufficient for adequate distribution, but at the same time to provide against excessive pumping head where large water quantities are handled, water coils are provided with various water-circuit arrangements. For instance, a typical coil 18 tubes high and 6 tubes deep in the direction of air flow can be arranged for 6, 9, 18, 24, or 36 parallel water circuits, as conditions may require. Orifices in individual tubes are occasionally employed, but are usually unnecessary as the resistance of individual water circuits is generally sufficient to effect a satisfactory distribution. In precooling coils using well water, where there

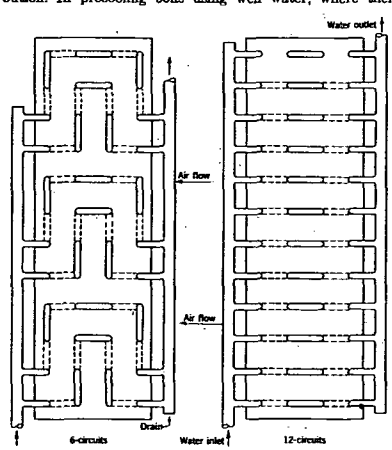


Fig. 2 . . . . Various Water Circuit Arrangements

may be considerable sand and other foreign matter in the water, provision for cleaning of individual tubes is of advantage. It is important to arrange water coils for complete drainage (see Fig. 2). The drains are usually provided in the water piping at the coil header.

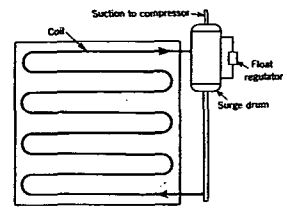


Fig. 3 . . . . Direct-Expansion Coil with Flooded System

Direct-Expansion Coils

Coils for volatile refrigerants present more complex problems of fluid distribution than water, brine, or steam coils. It is desirable that the coil be effectively and uniformly cooled throughout, and necessary that the compressor be protected from entrained, unevaporated refrigerant. There are two types, namely, flooded systems, and thermal-expansion-valve systems, as shown in Figs. 3 and 4. In a flooded coil, the circulation is similar to that in a water tube boiler. The liquid is maintained at the proper level by the action of a float regulator as shown in Fig. 3. The thermal-expansion-valve system depends upon the thermal valve to feed automatically just as much liquid to the coils as is required to maintain the superheat at the coil-suction outlet within predetermined limits, which vary from about 6 to 15 deg. The thermal-valve arrangement is in common use for the type of coils covered by this chapter, while the flooded system is rarely used.

With the flooded system the refrigerant distribution through the tubes depends on properly selecting the length of the feeds, and the head of liquid imposed upon the liquid inlets. The flow of refrigerant from the liquid inlets must be horizontal and upward. The possibility of gas trapping must be avoided. No auxiliary distributing devices are required.

In the thermal-valve system, the path of each refrigerant feed from the distributor to the suction header is called a circuit. The length of each circuit should be chosen to provide good heat transfer and return of oil, as well as a reasonable pressure drop across the circuit. The external loading conditions, the tube size, and other factors, influence the practical length of a circuit. Because of the various factors involved, the practical length and the arrangement of the circuits is developed from laboratory tests and the design of coils for units produced in volume are usually verified by additional tests. It is general practice to obtain superheat by suitable design of the coil rather than by use of external heat exchangers or other auxiliary devices.

To insure reasonably uniform refrigerant distribution in multi-circuit coils, it is common practice to provide a distributing means, between valve and coil liquid inlets, to divide the refrigerant equally among the feeds. Such a distributor must be effective for distributing both liquid and vapor, because the entering refrigerant is a mixture of the two. Fig. 5 shows three typical types of distributors. In distributor A the liquid and gas mixture from the thermal valve is led tangentially into a chamber. The coil feed connections extend outward radially at the top of this chamber. In distributor B the refrigerant is discharged at a high velocity through a central jet against the end plate, forming a uniform mixture of gas and liquid within the distributor, from which individual connections are led as shown. In type C the refrigerant enters at high velocity from the thermal valve and is discharged against the end plug in which the individual liquid feeds are closely arranged. These distributors can be used in either vertical or horizontal position. There are also other types of headers such as the centrifugal and weir type. The individual liquid connections from the distributor to the coil inlet are commonly made of small diameter tubing, and are all of the same length and diameter in order to impose the same friction between the distributor and the coil. Since the thermal valves act in response to the superheat at the coil outlet, this superheat should be produced with the least possible sacrifice of active evaporating surface. Sometimes a single thermal valve is used per coil. In other cases, multiple valves are used, with the coil divided across the air flow or parallel to the air flow as shown in Fig. 6. The arrangement of Fig. 7 should be avoided, since it offers the disadvantage of unequal load on the two parallel circuits.

Control of Coils

Chapter 43 discusses methods of controlling air-heating and cooling coils to meet system or space requirements, and factors that should be considered in sizing automatic valves for steam and water coils.

Flow Arrangement

In all heat transfer processes, the relative directions of flow of the fluids influence the performance of the heat-transfer surface. In air-heating and cooling coils, the air usually flows at right angles to the tubes. In a coil having only one row in the direction of air flow, the relative directions of flow would all be at right angles. In coils having more than one row in the direction of air flow, the media in the tubes may be variously circuited as illustrated for

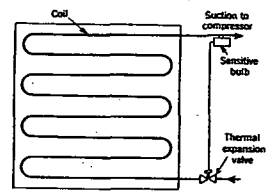


Fig. 4 . . . . Direct-Expansion Coil with Thermal Valve System

water or steam coils in Fig. 8. It is the usual practice to designate a coil that is circuited and employed as in Fig. 8A, as a *parallel-flow* coil. Figs. 8B and 8C show the general arrangements that are termed *counter-flow* and *cross-flow* respectively. Cross flow is common in steam

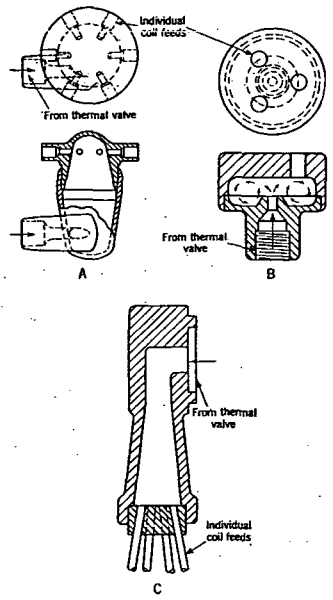


Fig. 5 .... Refrigerant Feed Distributing Heads

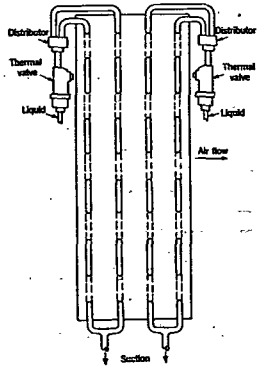


Fig. 7 .... Arrangement for Depth Control

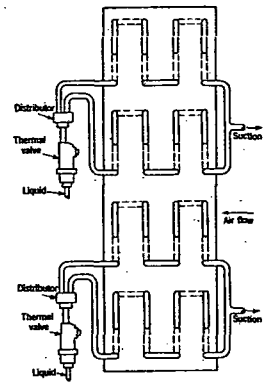


Fig. 6 .... Arrangement for Face Control

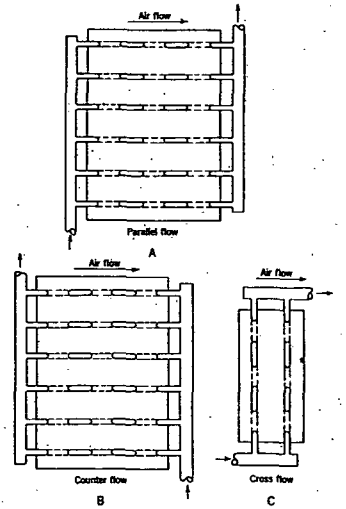


Fig. 8 .... Flow of Media in Tubes in Relation to Air Flow

heating coils, the temperature within the tubes being substantially uniform, and the mean temperature difference the same whatever the direction of flow, relative to the air. The parallel and counter-flow arrangements as illustrated in Figs. 8A and 8B are common in brine or water coils, with the counter-flow arrangement being the preferred arrangement to secure the advantage of highest possible mean temperature difference. Most direct-expansion coils also follow the general scheme of counter flow but the circuiting requirements for proper loading sometimes result in a combination of the other flow arrangements with counter flow in the same coil. Cross flow is generally avoided in the volatile refrigerant coils because of the unequal loading of the parallel circuits.

Applications

Heating coils in field-assembled banks are used for a number of purposes as described in Chapter 19. They may be arranged with the air flow vertical or horizontal, although the latter is more common. For steam heating, the coils may be set with the tubes vertical or horizontal. In the latter case the coil should be sloped to provide for condensate drainage. Because of the multi-circuit feed arrangement and the necessity for avoiding air and water pockets, water heating coils are generally arranged with the tubes horizontal. Certain precautions must be taken against freezing. Where steam coils are used with entering air below freezing temperature, throttling the steam supply may cause freezing of the condensate in the bottom of the coil, if the tubes are of the variety not provided with internal distributing pipes or an equivalent arrangement.

Some coils have inner distributing tubes, and also have supply and return headers cast in one piece. In this type of coil the condensate that forms in the outer tube has resulted from steam fed from the inner tube orifices. This condensate flowing back along the warm inner tube is prevented from freezing. A wide range of modulation at very low temperatures without danger of freezing, is therefore obtained. As an added precaution, with both steam and water coils, the outdoor air inlet dampers are often closed automatically when the fan is stopped to avoid trouble caused by very cold outdoor air drifting in during *off* periods.

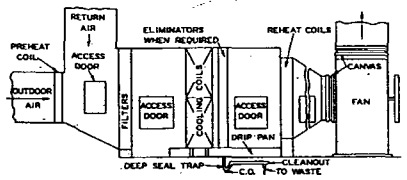


Fig. 9 .... Typical Arrangement of Cooling Coils in a Central System

A typical arrangement of coils is shown in Fig. 9. All of the air should be filtered to prevent dirt, insects, and foreign matter from accumulating on the coils. The cooling coil and humidifier must be provided with a drip pan to catch the condensate formed during the cooling cycle and the excess water from the humidifier. The drain connection should be on the downstream side of the coils, made of ample size (not less than 1¼-in. pipe), provided with accessible clean-outs, a deep-seal trap, and discharge to an indirect waste or storm sewer, so that there will be no possibility that sewer gas may enter the system. Precautions against freezing must be taken with the humidifier piping and drain connections as well as with steam or water coils.

Face and bypass dampers have proven to be satisfactory means of capacity reduction for cooling coils using non-volatile refrigerants. The use of such face and bypass damper

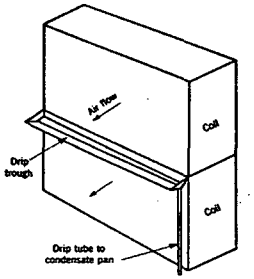


Fig. 10 .... Coil Arranged with Drip Trough

arrangements on volatile refrigerant coils poses difficult refrigerant control problems which are most severe when a change of setting at the thermostat imposes a rapid change in the dampers at the coils. On many installations, the coil bank is divided into 2, 3, or 4 sections and each section is connected to an individual compressor, or connected to a modulating compressor or compressors which will adjust to the load as each succeeding coil section is cut out by the thermostat.

The design features of the coil (fin spacing and type of fins, whether circular fins or plate) determine at what air velocity condensed moisture will be blown off the coil. Water blows off some coils below 400 fpm *face* velocity, while this does not occur on others at *face* velocities in excess of 500 fpm. Where there is a likelihood of water being blown off the coil face, eliminator plates should be placed on the downstream side of the coil. Plate coils are most frequently installed without eliminator plates. When a number of coil sections are stacked one above another, there is a tendency for the condensate to be carried out into the air stream as it drips from one coil to the next one. In such cases, drip troughs as shown in Fig. 10 are used to collect the water and conduct it to the drain pan.

The enclosure around a coil or filter bank should be corrosion resistant and should have adequate access doors to provide for changing of air filters, cleaning of coils, adjusting expansion valves, oiling motors, etc.

Sometimes cooling and dehumidifying coils are sprayed with water to increase the rate of heat transfer and to provide outlet air approaching saturation. This arrangement requires a collecting tank and recirculating pump to maintain water circulation whenever the apparatus is in operation. Fig. 11 illustrates this arrangement with a bypass around the coil. This bypass is often used to facilitate

thermostatic control. It is advantageous to direct only return air through the bypass rather than a mixture of return and outdoor air in order to hold the humidity ratio under thermostatic control.

Although both heating and cooling coils are made of sufficient strength to take up expansion and contraction arising within themselves, care should be taken to avoid imposing strains from the piping on the coil connections. (See Chapter 26.)

COIL SELECTION

In the selection of a coil it is necessary to consider various factors:

1. The duty required: heating, cooling, dehumidifying, and the capacity required to maintain balance with other system components such as compressor equipment in the case of direct expansion coils.
2. Temperature of entering air—dry-bulb only if there is no dehumidification; dry- and wet-bulb if moisture is to be removed.
3. Available heating and cooling media.
4. Space and dimensional limitations.
5. Air quantity limitations.

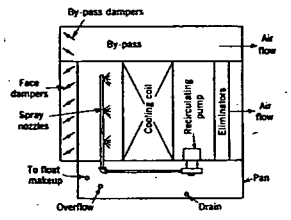


Fig. 11 .... Recirculating Spray System for Cleaning Coils

6. Allowable resistances in air circuit and through tubes.
7. Allowable resistance between the inlet and the outlet of the tube circuits of the coil.
8. Characteristics of individual coil designs.
9. Individual installation requirements, such as, for example, the type of automatic control to be used.

The duties required may be determined from information in Chapter 9, 11, 12, 13, and 19. There may, or may not, be a choice of cooling and heating media, as well as operating temperatures, depending upon whether the installation is new or is in combination with present sources of heating or cooling. Space limitations are dictated by the requirements of individual cases. The air quantity is influenced by a number of considerations. The air quantity through heating coils is often made the same as that necessary to handle the summer cooling load. The air handled may be limited by the use of installed ventilating ducts for air distribution or may be determined by requirements for satisfactory air distribution or ventilation. The resistance through the air circuit influences the fan horsepower and speed. This resistance may be limited to allow the use of a given size of fan motor, or to keep the operating expense low, or it may be limited because of sound-level requirements. The allowable friction through the water or brine circuit may be dictated by the head available from a given size of pump and pump motor. The performance of a surface heating or cooling coil depends upon correct choice of the original equipment, and upon proper application and maintenance.

Coil ratings are based on a uniform *face* velocity. Interference with uniform air flow through the coil will affect performance. Such air flow interference may be caused by air entrance at odd angles or by inadvertent blocking of a portion of the coil face. To obtain rated performance it is necessary that the air quantity be adjusted on the job to that used when selecting the coil, and that it be kept at this value. The most common cause of a reduction of air quantity are the fouling of the filters and collection of dirt on the coils. These difficulties can be avoided by proper design and regular servicing. There are a number of ways in which coils may be cleaned. A common method is to wash them with water. They can sometimes be brushed and cleaned with a vacuum cleaner. In bad cases of neglect, especially on restaurant jobs where grease and dirt have accumulated, it is sometimes necessary to remove the coils and wash off the accumulation with steam, compressed air and water, or hot water. The best practice, however, is to keep the filters serviced, and to inspect and wash the coils at regular intervals.

The proper selection of coils requires an understanding of the requirements of each case, and should be based on an economic analysis of the plant design as a whole. While no general rule can be established for the selection of heating or cooling coils, it is possible, nevertheless, to point out the limits of usual practice and to indicate the influence of the variables involved in the coil selection.

Heating Coils

Steam and *hot water heating coils* are usually *rated* within these limits:

Air Face Velocity—200 to 1200 fpm, sometimes up to 1500 fpm.
Steam Pressure—2 to 200 psig, sometimes up to 350 psig.
Hot Water Temperature—150 to 225 F.
Water Velocity—2 to 6 fps.

Individual cases may deviate widely, but the tabulation given herewith will serve as a guide to usual *heating installation practice*:

Air Face Velocity—500 to 800 fpm face, 500 being a common figure.
Delivered Air Temperature—varies from about 72 F for ventilation only, to about 150 F for complete heating.
Steam Pressure—2 to 10 psig, 5 psig being common.
Hot Water Temperature—150 to 225 F.
Water Velocity—2 to 6 fps.
Water Quantity—Based on about 20 deg temperature drop through a hot-water coil.
Air Resistance—The total resistance through heating coils is usually limited to from ⅜ to ⅝ in. of water for public buildings, to about 1 in. for factories.

The selection of heating coils is relatively simple as it involves dry-bulb temperatures and sensible heat only, without the complication of simultaneous latent heat loads, as in cooling coils. For a given duty, entering air temperature, and steam pressure, it is possible to select several arrangements of the same design of coil depending upon the relative importance of space, cross-sectional area, and air resistance.

Cooling Coils

Cooling and *dehumidifying coils* are usually *rated* within these limits:

Entering Air Dry-Bulb—60 to 100 F.
Entering Air Wet-Bulb—50 to 80 F.
Air Face Velocities—300 to 800 fpm (sometimes as low as 200 and as high as 1200).
Volatile Refrigerant Temperatures—25 to 55 F, at coil suction outlet.
Water Temperatures—40 to 65 F.
Water Quantities—2 to 6 gpm per ton, or equivalent to a water temperature rise of from 4 to 12 deg.
Water Velocity—2 to 6 fps.

The ratio of total to sensible heat removed varies in practice from 1.00 to about 1.65, i.e., sensible heat is from 60 to 100 percent of total, depending on the application. (See Chapter 19.) Since required ratios may demand wide variations in air velocities, refrigerant temperatures, and coil depth, general rules as to their values may be misleading. On most comfort installations air face velocities between 400 and 600 fpm are frequent, 500 being a common value. Refrigerant temperatures ordinarily vary between 35 and 50 F where cooling is accompanied by dehumidification. Water velocities range from 2 to about 6 fps.

When no dehumidification is desired, for which condition the dew point of the entering air is equal to or lower than the cooling coil surface temperature, the coil selection is made on the basis of dry-bulb temperatures and sensible-heat transfer only, the same as with heating coils. It is possible also to choose various arrangements of face area, depth, air velocity, etc., for the same duty.

Dehumidifying Coils

The performance of coils accomplishing both cooling and dehumidification is determined by a series of tests carried on under laboratory conditions. Coil ratings can be prepared from such laboratory test data to facilitate the selection of coils for cooling and dehumidifying duty. The coil capacity should be in balance with the capacity of related equipment, such as compressors, and the temperature of the circulating medium.

The selection of cooling coils for factory-assembled self-contained air conditioners is generally accomplished in conjunction with laboratory testing. The current industry standards call for ratings at 33.4 cfm per thousand Btu of cooling capacity and this is approximately 400 cfm per ton of refrigeration. The use of an entering air condition of 80 F dry-bulb and 67 F wet-bulb is representative of the entering air conditions actually encountered in many comfort operations because, while the indoor conditions are usually lower than 67 wet-bulb, the introduction of outdoor air will usually bring the mixture of air to the cooling coil up to an approximation of the 67 wet-bulb entering air condition at design conditions.

The selection of cooling coils for field assembled projects is usually accomplished by the use of coil rating tables. The practice of selecting coils from the load division indicated by the load calculation has worked out satisfactorily for the usual human comfort applications. Additional design precautions and refinements are being used for more exacting industrial applications and for improved results on all types of air conditioning in the more humid areas. One of these refinements is the use of a separate cooling coil to cool and dehumidify the ventilation air before admixture with recirculated air. This procedure takes care of one of the main sources of moisture in the usual application. Provision for reheat is required for some industrial applications, and is used for better results on very special commercial applications.

In checking the operating results obtained from cooling coils in various air-conditioning installations, it is necessary to keep in mind the influence of the climatic conditions of the various areas encountered. The majority of problems are encountered at *light-load* conditions when the cooling requirement is considerably less than at design conditions. In the hot, dry climates, where the outdoor dew points are so constantly low that dehumidifying is not generally a problem, the *light-load* condition does not pose any special problems. In the hot, humid climates, where the outdoor dew points are generally high and close to the dry-bulb temperatures, the *light-load* condition has a higher proportion of moisture and a correspondingly lower proportion of sensible heat. These climatic conditions result in higher dew points in the conditioned spaces during the *light-load* conditions unless some of the special means for controlling the inside dew points are used. In the geographic locations where warm weather occurs with both high and low dew points, such as around the Great Lakes, the *light-load* operating conditions with the higher percentage of moisture loading will be encountered less frequently and consistently than in the south coastal areas.

Care should be taken to avoid freezing at light loads. In general, freezing occurs when the coil surface temperature falls to 32 F. With usual coils for comfort installations, this does not occur unless the evaporating temperature at the coil outlet is about 20 to 25 F. The exact value depends on the design of the coil and the amount of loading. Although it is not customary to choose coil and condensing units to balance at low temperatures at peak loads, there is danger of this occurring when the load decreases. This is further aggravated if a bypass is used so that less air is passed through the coil at light loads.

HEAT TRANSFER AND AIR FLOW RESISTANCE

The transfer of heat between the heating or cooling medium and the air stream is influenced by several variables:

1. The temperature difference.
2. The design and surface arrangement of the coil.
3. The velocity and character of the air stream.
4. The velocity and character of the medium in the tubes.

The driving force is usually taken as the logarithmic mean temperature difference for heating or cooling without dehumidification. The rating of cooling coils for combined cooling and dehumidification is discussed later in this chapter. With volatile refrigerants there is often an appreciable pressure drop and corresponding change in evaporating temperature through the refrigerant circuit. The problem is further complicated by the fact that the refrigerant is evaporating in part of the circuit, and superheating in the remainder. In the case of volatile refrigerants, a cooling coil is tested and rated in conjunction with a specific distributing and liquid-metering device, and the capacities are stated for a given superheat condition of leaving vapor.

The design and surface arrangement of the coil include

such items as materials, type, thickness, height, and spacing of the fins, and the ratio of this surface to that of the tube, the use of the staggered or in-line tube arrangement, and provisions to increase the air turbulence such as the use of corrugated instead of flat fins. Staggered tubes increase the total heat transfer, as against the in-line arrangement, and corrugated fins may be more effective than flat. This design and surface arrangement has a large effect on the air-film heat-transfer resistance.

The velocity of the air usually considered is the coil *face* velocity. This bears a *varied relation to the actual velocity* over the surface, depending upon the individual coil design. As long as a fixed design of coil is under consideration face velocities may be used, but they may be unsatisfactory in comparing different designs, as it is the *actual surface velocity that is significant*. The air volume is usually based on standard air at 70 F and a barometric pressure of 29.92 in. mercury.

At the same mass air velocity, varying performance can be obtained depending upon the turbulence of the air flow into the coil, and upon the uniformity of distribution of air over the coil face. The latter is very important in obtaining reliable test ratings, and in realizing rated performance in actual installations. The air resistance through the coils will assist in distributing the air properly, but where the inlet duct connections are brought in at sharp angles to the coil face, the effect is frequently bad and there may even be reverse air currents through a portion of the coils. This reduces the capacity, but can be avoided by proper layout or by the use of vanes or baffles.

Heat-transfer information on plain pipe coils has been developed and verified through many tests. In the case of finned coils, the heat transfer from the cooling or heating medium to the air stream is dependent on so many factors that reliable rating and performance information for any design of coil must be based upon actual tests of the specific coil. Mathematical comparisons of different designs of coils on a square foot of surface and face area basis may be misleading. The selection of finned coils should be made from curves or tables of coil performance prepared from a series of adequate and reliable tests. There are cases in which the engineer must extend available data or design for a single unique installation. For such purposes the following coil calculations will be useful.

PERFORMANCE OF HEATING AND DRY COOLING COILS

The performance of heating and dry cooling coils depends in general upon:

1. The overall coefficient of heat transfer from the fluid within the coil to the air it heats or cools.
2. The mean temperature difference between the fluid within the coil and the air flowing over the coil.
3. The physical dimensions of the coil.

Thus, for any one definite operating condition, the heating or cooling capacity of a given coil is expressed by the following basic formula:

$$q_t = U \times (\Delta t_m) \times A \times N \qquad (1)$$

where

q_t = total heat transfer of the coil, Btu per (hour) (square foot of coil face area).

U = overall coefficient of heat transfer, Btu per (hour) (square foot of external coil surface) (Fahrenheit degree temperature difference between the fluid within the coil and the air flowing over the coil).

Δt_m = mean temperature difference, Fahrenheit degrees, between the fluid within the coil and the air passing over it. (This is commonly taken as the logarithmic mean temperature difference.)

A = external surface area of the given coil, square feet per (square foot of coil face area) (row of coil depth).

N = number of rows of coil depth.

Overall Coefficient of Heat Transfer

While the overall coefficients of heat transfer for plain pipe coils have been defined by numerous tests within close limits, the verification of a plain coil design by a series of tests is an accepted commercial practice. The overall coefficients of heat transfer for finned coils should always be obtained from tests. The data from a series of tests may be used for purposes of extending coil data beyond test range but such data should later be verified by test.

Considering any coil, whether of bare pipe or of finned type, the overall heat-transfer coefficient for a given size and design of coil can always be considered as a combined effect of three individual heat-transfer coefficients, namely:

1. The film coefficient of heat transfer between air and the external surface of the coil, usually given in Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference).
2. The conductivity of the coil material—tube wall, fins, ribs, etc., usually given in Btu per (hour) (square foot of surface) (Fahrenheit degree per inch).
3. The film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, usually given in Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature difference).

These three individual coefficients acting in series result in an overall coefficient of heat transfer in accordance with the basic laws given in Chapters 5 and 9. For a *bare-pipe* coil the overall coefficient of heat transfer, whether for heating or for cooling (without dehumidification), can be expressed by a simplified basic formula as follows:

$$U = \frac{1}{\frac{R}{f_i} + \frac{L}{k} + \frac{1}{f_o}}. \qquad (2)$$

where

U = overall coefficient of heat transfer, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and fluid within the coil).

f_i = film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature difference between that surface and the average fluid temperature).

f_o = film coefficient of heat transfer between air and the external surface of the coil, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between the mass of air and the external surface).

k = conductivity of material from which the bare pipe is constructed, Btu per (hour) (square foot) (Fahrenheit degree per inch thickness).

L = thickness of tube wall inches.

R = ratio between external and internal surface of the bare tube, usually varying from 1.03 to 1.15 for the tube used in typical heating or cooling coils. This ratio R is inserted in the formula in order to place internal fluid coefficient of heat transfer on the basis of external surface.

Frequently, when pipe or tube walls are thin and of material having high conductivity (as is the case in construction of typical heating and cooling coils) the term L/k in Equation 2 becomes negligible and is generally disregarded. (The effect of the term L/k in typical bare-pipe heating or cooling coils seldom exceeds 1 to 2 percent of the overall coefficient.) Thus, in its simplest form, for bare pipe:

$$U = \frac{1}{\frac{R}{f_i} + \frac{1}{f_o}}. \qquad (3)$$

For finned coils the formula* for the overall coefficient of heat transfer can be conveniently written:

$$U = \frac{1}{\frac{R}{f_i} + \frac{1}{\eta f_o}} \qquad (4)$$

in which the term η, called the *fin efficiency*, is introduced to allow for the resistance to heat flow encountered in the fins.

The term R, in this case, is the ratio of *total* external surface to internal surface. For typical designs of finned coils for heating or cooling, this ratio varies from 10 to 30. Term R is again introduced to place the internal surface coefficient of heat transfer on a basis of external surface.

The performances of all heating and dry cooling coils are influenced by these same factors. But, when cooling coils operate wet or act as dehumidifying coils, the performance cannot be predicted on the basis of overall coefficients.

PERFORMANCE OF DEHUMIDIFYING COILS

When the dew point of the air leaving a cooling coil is lower than the dew point of the air entering the coil, some moisture removal has been accomplished. A coil that normally accomplishes (or is designed to accomplish) moisture removal in addition to sensible-heat cooling is termed a dehumidifying coil.

In most air-conditioning processes, the air may be considered as a mixture of water vapor and the dry components. Both dry components and the water vapor enter an air-conditioning coil at the same dry-bulb temperature; both the dry components and the water vapor lose sensible heat during the contact with the first portion of the cooling coil in the same manner as in a dry cooling coil. As the dry-bulb temperature of the mixture approaches the dew point of the water-vapor components, moisture removal starts.

The psychrometric path of air through a cooling coil is generally assumed to follow a path similar to that shown in Fig. 12. When the dry-bulb temperature of the air mixture in the coil falls below the entering dew-point temperature, moisture removal proceeds. The indications are that the lower the leaving dry-bulb temperature below the entering dew-point temperature, the less will be the difference between the leaving dry-bulb temperature and the leaving dew-point temperature.

The first portion of a cooling coil (in the direction of air flow) may function in the same manner as a dry cooling coil. Where the moisture removal starts, the cooling surfaces also

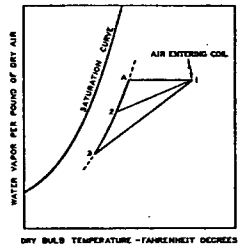


Fig. 12 .... Performance of Dehumidifying Coil

continue the removal of sensible heat, thereby carrying the load due to both. As saturation is approached in the cooling coil, each degree of sensible cooling is approximately matched by a corresponding degree of dew-point decrease. However, while the sensible heat removal from the dry air remains approximately constant per degree change, the amount of latent heat removal per degree of dew-point change varies considerably because moisture content varies widely at different temperatures.

For example, the following tabulation compares the amount of moisture removal involved in a reduction of one degree of dew point from 60 to 59 F with the removal from 50 to 49 F:

Dew Point	$W_s \times 10^3$ lb/(lb)	Dew Point	$W_s \times 10^3$ lb/(lb)
60	11.080	50	7.658
59	10.690	49	7.374
Difference	0.390	Difference	0.284

The above values are given in Table 2, Chapter 3.

When cooling coils act as dehumidifying coils, the performance can be predicted accurately only from tests at a sufficient number of points to establish the definite performance characteristics of the coil under varying conditions of loading and of entering air. Dehumidifying coils employing volatile refrigerants are generally rated in conjunction with specific refrigerant distributing and flow control equipment. The combination of the coil with its refrigerant control equipment (such as distributor and expansion valve, and capillary tube or float valve) must be tested at both the higher and lower capacities of its rated range. The lower capacities impose a test on the distributor to provide equal distribution and on the control to modulate without *hunting* at the lower capacities. The higher capacities result in a greater pressure drop through the coil system and a test of the maximum feeding capacity of the flow control device at various *head pressures*.

Most coil manufacturers have their own methods of producing performance rating tables from a suitable number of coil-performance tests. A method of testing and rating coils,

* Rational development and rating of extended air cooling surface, by H. B. Pownall (*Refrigerating Engineering*, October 1935, p. 211).

titled *Joint ASRE-ASHAE Standard Method of Testing and Rating Forced-Circulation Air-Cooling and Air-Heating Coils*, has been adopted (1958).

DETERMINING REFRIGERATION LOAD

The following determination of the refrigeration load shows a division of the true *sensible* and *latent heat* loss of the air, which is accurate within the limitations of the data. These divisions will not correspond to load determination obtained from approximate factors or constants.

The total refrigeration load q_t of a cooling and dehumidifying coil (or air washer) is indicated on Fig. 13 and consists of the following components:

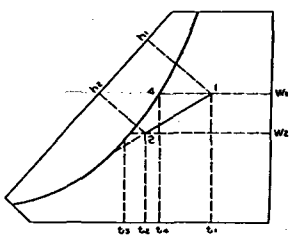


Fig. 13 .... Psychrometric Performance of Cooling Coil in Central System

1. The *sensible heat* q_s removed from the dry air and moisture in cooling from entering temperature t_1 to leaving temperature t_2.

2. The *latent heat* q_s removed to condense the moisture at the dew-point temperature t_4 of the entering air.

3. The heat of *subcooling* q_w removed from the condensate in cooling it from the condensing temperature t_4 to the leaving condensate temperature t_3.

Items 1, 2, and 3 may be related by

$$q_t = q_s + q_s + q_w \tag{5}$$

If only the *total heat* value is desired, it may be computed by

$$q_t = (h_1 - h_2) - (w_1 - w_2)h_{w3} \tag{6}$$

where

h_1 and h_2 = enthalpy at points 1 and 2 respectively.
w_1 and w_2 = humidity ratio at points 1 and 2 respectively.
h_{w3} = enthalpy of saturated liquid at the final temperature, t_3.

If a breakdown into *latent* and *sensible heat* components is desired, the following relations may be used:

The *latent heat* may be found from

$$q_s = (w_1 - w_2)h_{fs4} \tag{7}$$

where

h_{fs4} = enthalpy at the condensing temperature, t_4.

The *sensible heat* may be shown to be

$$q_s + q_w = (h_1 - h_2) - (w_1 - w_2)h_{s4} + (w_1 - w_2)(h_{w4} - h_{w3}) \tag{8}$$

where

h_{s4} = enthalpy at the condensing temperature, t_4.
h_{w4} = enthalpy of saturated liquid at condensing temperature, t_4.

The last term in Equation 8 is the heat of *subcooling* the condensate from the condensing temperature t_4 to its final temperature t_3. Then,

$$q_w = (w_1 - w_2)(h_{w4} - h_{w3}) \tag{9}$$

All values for solving the foregoing equations may be found on the ASHAE Psychrometric Chart and Tables 2 and 3 of Chapter 3.

Example 1: Air enters a coil at 90 F dry-bulb, 75 F wet-bulb; it leaves at 61 F dry-bulb, 58 F wet-bulb; leaving water is assumed to leave at a temperature between the leaving air dew point and coil surface temperature of 54 F. Find the total, latent, and sensible cooling loads on the coil.

Solution: From the ASHAE Psychrometric Chart, find the following:

h_1 = 38.42 Btu per lb of dry air.
h_2 = 25.10 Btu per lb of dry air.
t_4 = 69 F wet-bulb of entering air.
w_1 = 0.01525 lb per lb of dry air.
w_2 = 0.00960 lb per lb of dry air.

From Table 3, find:

h_{w4} = 37.11 Btu per lb.
h_{w3} = 22.12 Btu per lb.
h_{fs4} = 1054.27 Btu per lb.
h_{s4} = 1091.34 Btu per lb.

The total heat from Equation 6 is

$$q_t = (38.42 - 25.10) - (0.01525 - 0.00960) \times 22.12$$
$$= 13.32 - (0.00565 \times 22.12)$$
$$= 13.32 - 0.12 = 13.20 \text{ Btu per lb dry air.}$$

The *latent heat* from Equation 7 is

$$q_s = 0.00565 \times 1054.27 = 5.96 \text{ Btu per lb of dry air.}$$

The *sensible heat* by difference, is

$$q_s + q_w = q_t - q_s = 13.20 - 5.96 = 7.24 \text{ Btu per lb of dry air.}$$

Or the *sensible heat* may be computed from Equation 8 as

$$q_s + q_w = (38.42 - 25.10) - (0.00565 \times 1091.34) + 0.00565\,(37.11 - 22.12)$$
$$= 13.32 - 6.16 + 0.00565 \times 14.99$$
$$= 13.32 - 6.16 + 0.08 = 7.24 \text{ Btu per lb of dry air.}$$

The *subcooling* of the condensate as a part of the sensible heat is indicated by the last term of the equation, 0.08 Btu per lb of dry air.

CHAPTER 24

AIR CLEANING

Atmospheric Air Cleaners: Airborne Particulate Matter, Viscous-Impingement Filters, Dry Air Filters, Electronic Air Cleaners, Air Filter Performance, Selection, Maintenance, Installation, Adsorption of Vapors; Industrial Air and Gas Cleaners: Degree of Cleaning, Selection, Types, Application, Inertial Separators, Scrubbers, Wet Collectors, Filters, Electrostatic Precipitators, Adsorbers, Absorbers, Combustion Devices

AIR cleaning devices remove contaminants from an air or gas stream. They are available in a wide range of designs to meet various air cleaning requirements. Degree of removal required, quantity and characteristics of the contaminant to be removed, and conditions of the air or gas stream will have a bearing on the device selected for a given application. Definitions and a discussion of contaminant characteristics are given in Chapter 7, together with some consideration of their origin.

Air cleaning devices are divided in this chapter into two basic groups: Atmospheric Air Cleaners, described in Part I, and Industrial Air and Gas Cleaners, described in Part II.

Atmospheric Air Cleaners are ordinarily used to remove particulates such as are found in outdoor air, and are employed in ventilation, air conditioning, and heating systems where dust content seldom exceeds 4 grains per 1000 cu ft of air.

Industrial Air and Gas Cleaners are ordinarily used for the heavier concentrations encountered in local exhaust ventilation where the particulate content (loading) ranges from 100 to 20,000 grains per 1000 cu ft of air. Because of these heavier loadings, atmospheric air cleaners can seldom be used for the control of process aerosols in industrial applications.

PART I—ATMOSPHERIC AIR CLEANERS

Conventional air filters and electronic air cleaners are installed in air handling systems to remove dusts. These dusts constitute a mixture of particle sizes within the classification of temporary and permanent impurities listed in Fig. 1, Chapter 7, including bacteria, pollens, house dusts, and similar allergens which motivate attacks on persons of allergic sensitivity.[1,2]

Since the purpose of the filter or cleaner is to free the air of as much existing contamination as practicable, the degree of air cleanliness required should influence the choice of apparatus. Atmospheric dusts are mixtures of particles in all sizes. The removal of these particles and fractions becomes progressively difficult as the particle size decreases. Smoke particles are of major importance in many applications. Air cleaners will justify their cost through a reduction in housekeeping expense in the ventilated space, by the protection of the equipment in the ventilation system itself, and by providing relatively dust-free air for critical manufacturing processes.

Cleaning devices for atmospheric air are classified by the principle employed to collect dirt particles such as impingement on viscous coated media, filtration through porous media, or electronic air cleaning. In some cases a filter may display a combination of these principles. Each type of air cleaner has certain advantages. There are applications where it is desirable to pass air through a series of two or more different types to obtain optimum results.

AIRBORNE PARTICULATE MATTER

Suspensions of particulate matter in the air are called aerosols and consist of smokes, dusts, mists, and fumes. The characteristics of the aerosols which affect the performance of an air cleaner include particle size, concentration, shape, density, velocity, and surface characteristics. One of the most important of these is size.

The rate of settling of particles varies approximately as the square of the diameter. Particles larger than 10 microns in diameter settle so rapidly that the concentration of such particles decreases rapidly as the distance from the source increases. Particles less than 1 micron in size settle so slowly that the ordinary convection currents prevent sustained downward motion, and consequently such particles remain in the air for long periods.

If particles could be examined through a super microscope having a magnification of 250,000 diameters, a tobacco smoke particle of 0.1 micron would appear to be 1 in. in diameter, or approximately the size of a golf ball; a soft coal smoke particle 0.3 micron in diameter would appear like a baseball; a ragweed pollen grain of 20 microns in diameter would appear 16.5 ft in diameter, while the 50 micron particle (just visible to the naked eye and able to pass through a 270 mesh screen) would appear to be 50 ft in diameter. Consideration of this range in particle size from a golf ball to a sphere 50 ft in diameter will emphasize the difficulty of devising any single test to measure adequately the performance of air cleaning devices under all conditions of service.

The particles in the atmosphere can range in size from less than 0.01 micron up to things which are caught by an ordinary fly screen, such as lint, feathers, and insects. Almost all conceivable shapes and sizes are represented. The material is very commonly soot, ash, soil, lint, smoke, and fumes, but may include particles of almost any inorganic or organic material and even such living organisms as virus, bacteria,[3] and fungus spores.

This wide variety makes it impossible to design one type of cleaner which will be best for all applications. Mechanical filters of the low-pressure type can remove large particles effectively. Other mechanical filters can remove extremely fine particles, but may be handicapped somewhat by high-pressure drop and cost. Electronic air cleaners have the ability to give high effectiveness on normal atmospheric contamination, with a low operating-pressure drop.

As a general rule, the removal of the coarser dust particles and lint from the ventilating air produces tangible results in

so far as cleanliness in a house or building is concerned, because much of the finer dust remains suspended in the air and is removed from the building by the circulating air. However, since some of the fine dusts, especially smoke and fume particles, are undoubtedly deposited by means other than settling, such as electrical or thermal precipitation[4, 5] and by contact, the ability to remove small particles is desirable in an air cleaner if it can be obtained at a reasonable cost.

VISCOUS-IMPINGEMENT TYPE FILTERS

The viscous-impingement type of filters consist of relatively coarse media constructed of a suitable fiber, screen, wire, mesh, metal stamping or plates, or a combination of media. The filter may be of the unit type manually-cleaned, the replaceable type, or the automatic moving-curtain type.

The medium in a viscous-impingement type filter is usually a fiber pack for non-automatic types, or a series of metal plates for automatic self-cleaning types. In either case, the medium is treated with a viscous substance, often an oil or grease, called the *adhesive* or the *saturant*, intended to retain dust particles which come in contact with it. Also, in either case, the arrangement is such that the air stream is broken up into many small air streams, and these are caused to change direction abruptly a number of times in order to throw the dust particles, by momentum, against the adhesive. Several desirable characteristics of an adhesive for air cleaners of this type are: (1) its surface tension should be such as to produce a homogeneous film or coating on the filter medium; (2) the viscosity should vary only slightly with normal changes of temperature; (3) it should prevent the development of mold spores and bacteria on the filter medium; (4) the liquid should have high capillarity, or ability to wet and retain the dust at all operating temperatures; (5) evaporation should be slight; (6) it should be fire resistant; (7) it should be odorless.

Various fibrous materials have been used as filtering media in unit filters of the viscous-impingement type. These include glass fiber, steel wool, similar *wool* of non-ferrous metals, wire screen, animal hair, hemp fibers, and other materials. In such filters, the medium is often packed more densely on the discharge than on the approach side, in order to increase the dust-holding capacity. This results in a selective arrestance of dust with the larger particles nearer the approach face. The arrangement also permits some penetration of lint into (but not through) the filter, so that the amount of lint which can be tolerated on the filter is also increased. Due to plane surface area the viscous-impingement type filter, however, may be inferior to some dry types if the air carries a high percentage of lint.

The resistance of air filters obviously increases with the rate of air flow through them. Face velocities of about 300 fpm, and resistances in the range from 0.1 to 0.2 in. water, when the device is new and clean, are usual for ventilation system filters.

Filters designated as high-velocity units are also available. In most instances these filters employ a uniform media pack and give best performance at velocities about 500 fpm. Impingement type air filters should hold a substantial amount of adhesive so that the dirt collected by the filter will be retained. The design and construction should also provide a goodly proportion of free space through the media so that high velocities can be used without excessive resistances. The design should assure that the media throughout the thickness of the filter will be effectively utilized and therefore, the filter should be designed to minimize the formation of a clogging dirt mat on the filter face. High-velocity filters are apt to be adversely affected if the amount of lint in the air to be cleaned is excessive.

The resistance of filters increases with dust or lint loading, and it is the resistance due to this cause that ordinarily necessitates servicing. The rate of loading obviously depends upon the amount as well as the kind of dust in the air, and for this reason, periods between servicing cannot be predicted with certainty. Manometers are often installed to measure the pressure drop across filter banks and thereby indicate when the filter requires cleaning. The pressure drop tolerated differs between operators and system designs. The resistance of a filter bank can be kept desirably low by periodically servicing some, but not all, of the units in the bank at one time. It is to be noted that a decline in efficiency may be the limiting factor in filter life rather than increased resistance due to load.

The method of cleaning viscous-impingement unit filters differs for different types of filters and kinds of dust. Much dry dust or lint can often be removed by rapping the filter.

Throw-away filters are constructed of inexpensive materials, and are designed to be discarded after one period of use. The frame is frequently a combination of cardboard and wire.

Cleanable filters usually have metal frames. Various cleaning methods have been recommended including: air jet, water jet, steam jet, washing in kerosene, and dipping in an oil. The last may serve both to clean the filter and add the necessary adhesive.

It is not mandatory that unit filters be removed from their metal frames for cleaning outside of the system. Cleaning of unit filters in place is feasible by means of hot water sprayed from a hose or by means of fixed nozzles to accomplish washing and adhesive application. Where filters are cleaned in place, provision should be made to collect and drain the water so as to prevent leakage from the filter housing.

Moving-Curtain Viscous Filters

Moving-curtain air filters are available in two main types. In one type the filtering medium making up the curtain is considered "permanent" and means are provided to remove the dust from the curtain mechanically. In the other type the curtain consists of a roll of viscous-coated fibrous filter medium which is not permanent but which is replaced after accumulating its dust load. Filters with moving cloth media, not viscous coated, are available essentially for lint removal.

The medium in the permanent, mechanically cleaned, moving-curtain type consists of a series of specially formed metal plates mounted on a pair of chains. The chains are mounted on sprockets located at the top and bottom of the filter housing, so that the filter medium can be moved as a continuous curtain up one side and down the other side of the sprockets. The arrangement is such that, at the bottom, the medium passes through a bath of special oil which both serves to remove the dirt from the plates, and acts as an adhesive when the cleaned plates next pass through the air stream. The plates forming the filtering medium or curtain usually overlap each other, and due to their special shape, form many small air passages between them. These air passages turn abruptly one or more times in order to give the impingement effect.

The medium in the replaceable moving-curtain type filter is supplied in roll form, one roll being sufficient for one to two years of operation in typical applications. The medium forming the moving curtain is unrolled from the supply roll

at the top, is moved down across the air stream where its dust load is accumulated, and then is rewound on a roll at the bottom. Replacement is accomplished by inserting a clean roll of medium at the top and disposing of the loaded dirty roll. In typical replacement medium, moving-curtain filters a signal is provided to indicate the end of the medium roll has reached the airstream. Usually provision is also made to stop the automatic curtain-moving mechanism before the end of the curtain is reached.

An electrically driven rotating device is usually supplied with a moving-curtain filter. The device may be set to move the curtain periodically, or a special switch, actuated by pressure drop, may be used to govern its motion. In operation, the resistance of an automatic filter will remain approximately constant as long as proper operation is obtained. A resistance of ½ in. water at a face velocity of 500 fpm is typical of this class.

DRY AIR FILTERS

The media in dry filters are usually fabrics or fabric-like materials. Media of wool felt, cotton batting (both glazed and unglazed), cellulose fiber, and other materials have been used commercially. The medium in a filter of this class is usually supported by a wire frame in the form of pockets or V-shaped pleats in order to increase the area exposed to the passage of air. A 2-ft-square unit may contain from 15 to 30 sq ft of medium.

Dry air filters, by virtue of the large area of medium used, have a comparatively high lint-holding capacity. The efficiency of dry type filters is usually higher than that of viscous impingement filters, while the life of the former based on dust-holding capacity, may be lower. Dust tends to clog the fine pores or openings of dry filters more quickly, thereby causing a higher rate of resistance rise than for viscous impingement filters. Efficiency with very fine particles equivalent to that of electronic air cleaners is not uncommon with certain types of dry filters. A comparatively deep bed, ½ to 1 in. thick, of filtering media having individual fibers in the order of 1 micron in diameter, is required for such high efficiency.

Dry filters may consist of a cleanable medium held in permanent frames, or a throw-away or replaceable filtering medium held in permanent frames, or the entire filter may be of the throw-away type. Usually the filtering medium alone is replaced after having collected its full dust load.

ELECTRONIC AIR CLEANERS

Two principal types of electronic air cleaners are available for air filter application: (1) ionizing type collectors where particles are given a definite charge by passing through an ionization zone, and (2) charged-media type collectors where the energized collecting medium induces a charge on the particles.

Ionizing Type Electronic Air Cleaners

Electronic air cleaners use electrostatic precipitation principles to collect particulate matter (see Fig. 1) but operate on much lower voltages than the type commonly used on industrial stacks. The *designation* Electronic Air Cleaner has been standardized to distinguish the class of electrostatic precipitator which alone is suitable for cleaning ventilating air.

In a typical case, a potential of 12,000 volts may be used to create the ionizing field, and some 6000 volts between the plates upon which the precipitation of dust occurs. These voltages, which are capable of momentary shock to personnel similar to that of a spark plug, necessitate some safety measures. A typical arrangement provides means for making the equipment inoperative when any door affording access to high-voltage parts is opened. Operation can be resumed only after all doors are closed. The voltages necessary for operation of the equipment are usually obtained by means of electronic high-voltage direct-current power packs operating from a 110–120 volt, 60-cycle, single-phase building service. Electric power consumption is about 10 watts per 1000 cfm, plus the approximately 40 watts required to energize the rectifier tube heaters.

Air cleaners of this type offer negligible resistance to air flow and, therefore, care must be exercised in arranging the duct approaches on the entering and leaving sides of the cleaners in order that the air flow may be distributed uniformly over the cross-sectional area. The efficiency of the electronic air cleaner is sensitive to air velocity, and the device itself has much less tendency to rectify the air stream than filters which have higher resistances. In most systems, resistance is deliberately added in the form of a perforated plate, pre-filters, or after-filters for the purpose of obtaining a uniform distribution of air. The resistance, including the baffles or after-filters, is constant and generally ranges from 0.15 to 0.25 in. of water at the usual velocities of 300 to 400 fpm. Such plates or filters, however, cannot compensate for defects in the duct layout ahead of the cleaner. Suitable screens of no coarser than 16 mesh should be installed across all outdoor air entrances to prevent insects, leaves, bits of paper, and similar material from entering the cleaner. Where lint is present in appreciable quantities, devices for lint removal should be installed ahead of the cleaner.

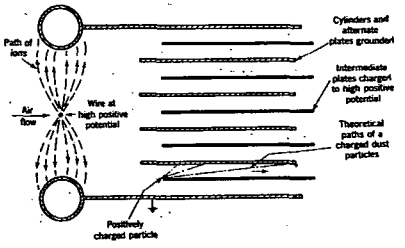


Fig. 1 . . . . Diagrammatic Cross-Section of Ionizing Type Electronic Air Cleaner

Electronic air cleaners of the ionizing type are efficient, low-pressure-drop devices for removing fine dust and smoke particles. They are available in fixed collector-plate and moving collector-plate types. The fixed collector plates are often coated with a special oil as an adhesive. Cleaning is generally accomplished by washing the cells in place with hot water from a water hose or by means of fixed or moving nozzle systems. The bottom of the equipment is made water tight and provided with a drain. In one moving-plate type, the grounded elements on which the dirt collects are mounted

on chains, and are alternately dipped in oil and exposed to the air stream.

Charged-Media Electronic Air Cleaners

The charged-media type air cleaner consists of a dielectric filtering medium, usually arranged in pleats as in typical dry filters. No ionization means are employed. The dielectric filtering medium may consist of glass fiber mat, cellulose mat,

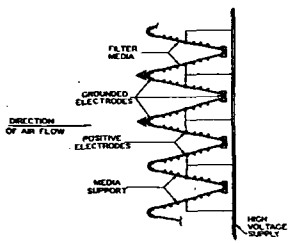


Fig. 2 .... Cross-Section Diagram of Charged Media Type Electronic Air Cleaner

or other similar material, and is supported on or in contact with a gridwork consisting of alternately grounded and charged members, the latter usually being held at a potential of 12,000 volts d-c. An intense and non-uniform electrostatic field is thus created through the dielectric medium. Airborne particles approaching the field are polarized and drawn toward filaments or fibers of the media. The general arrangement illustrating this type of filter is shown in Fig. 2.

The precipitator of this type offers resistance to air flow, when clean, on the order of 0.10 in. of water at 250 fpm velocity and, unlike the ionizing type, the resistance of the charged-media type electronic cleaner rises as dust is accumulated on the media. Because of these characteristics the filter tends to equalize the air distribution over the face of the filter. Like typical replaceable-media mechanical filters, the charged-media precipitator is serviced by replacing the filtering medium. The dielectric properties of the media are impaired when the relative humidity exceeds 70 percent.

AIR CLEANER PERFORMANCE AND TESTING

Atmospheric air cleaners are generally rated in terms of the three more critical factors in performance; namely, (1) resistance vs air flow, (2) efficiency or arrestance vs dust load at design air flow, and, (3) resistance vs dust load at design air flow.

Air flow is expressed in cubic feet per minute and is therefore a function of face velocity and filter size. Face velocity is defined as the average velocity of the air entering the effective face area of the cleaner and may be expressed by the equation:

$$V = Q/A \qquad (1)$$

where

V = face velocity, feet per minute.
Q = air flow, cubic feet per minute.
A = effective face area, square feet.

Effective face area is further defined as that area of the filter face inside the enclosing frame, flanges, or seals, but not corrected for any area occupied by any bracing, and retaining or strengthening grid members, located within the enclosing frame.

The rated capacity of general ventilation filters is often based on face velocities in the range of 250 to 520 fpm. Resistance to air flow, measured in inches of water column, at rated capacity, is generally available from the manufacturer (see Catalog Data Section). A suitable allowance must be made in the system design for increase in resistance as the filter becomes loaded with dust. Fig. 3 indicates the range of resistance to air flow found on tests of four unit filters. Type A is a dense pack used in bacterium control; Type B is a medium pack used for general ventilation work; Type C is a low-resistance unit, for use where low resistance is the important factor and maximum cleaning efficiencies are not essential; and Type D is a high-velocity viscous-impingement filter.

The effectiveness of air cleaners in removing a given dust from the air is called efficiency or arrestance. If the efficiency is denoted by E, then algebraically E is given by:

$$E = \frac{D_1 - D_2}{D_1} \qquad (2)$$

where

D_1 = amount of dust per unit volume in uncleaned air.
D_2 = amount of dust per unit volume in cleaned air.

A number of procedures for evaluating air cleaner efficiency have been proposed and are in use. These procedures differ principally in the type of test dust that is used and in the method of measuring the dust content of the entering and leaving air. The numerical value of the arrestance or efficiency of an air cleaner may be different for different methods of test even when the same test dust is used, and is generally different for different test dusts using the same test method. Caution is to be exercised in interpreting published arrestance data, since the test efficiency may be different from that obtained in an actual installation, or in the space being supplied with air.

The two most common methods for evaluating efficiency are: (1) the weight method, in which the weight concentration of dust in the air entering and leaving the filter is measured by some suitable means, and (2) the dust spot test in which the dust content of the entering and leaving air is measured in terms of its ability to stain filter paper. The particle count method is not used for efficiency evaluation, except in investigation of filter performance on specific par-

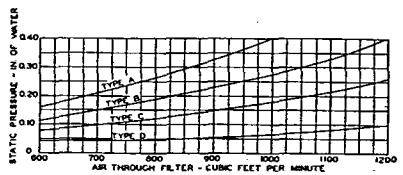


Fig. 3 .... Resistance to Air Flow of Four Unit Air Filters

ticles such as pollen, or on certain industrial dusts harmful to health. Dust particles can be captured on microscope slides by means of one of the various kinds of impingement devices. The process is useful if inspection and analysis of dust are desired, but particle counting is not sufficiently precise for evaluating the efficiency of a cleaner operating on a heterogeneous dust.

Among the weight methods that have been used are the Air Filter Institute Code[6] and modifications of the former ASHVE Code.[7] By the latter test method, a known weight of a prepared dust is injected into air supplied to the filter, and the quantity of dust in the cleaned air is determined by extracting and weighing the dust from a known volume of cleaned air. Dust extraction from the air is accomplished by drawing the air through a porous crucible or thimble by means of a high vacuum. The test dust specified in the ASHVE Code is a mixture of coal ash and carbon black.

The Air Filter Institute Code, Section I, specifies a test dust as follows:

"The test dust mixture consists of 72 percent standardized air cleaner test dust, fine (Arizona road dust, fine); 25 percent K-1 carbon black; and 3 percent No. 7 cotton linters (percentages by weight)."

As in the ASHVE Code, the entering concentration (D_1) is determined by feeding a known weight of the prepared test dust. The concentration of dust in the air leaving the filter (D_2) is determined by passing the entire air flow through a high-efficiency filtering blanket known as an absolute filter, the gain in weight of which is measured. Test data and results are presented on standardized forms made available by the Air Filter Institute.

The dust-spot or discoloration method of test for air-cleaner efficiency was developed at the National Bureau of Standards.[8] In effect, this method is based on a comparison of the volumes of air before and after cleaning by the filter that must be drawn through unit areas of suitable filter paper to yield dust spots of equal opacity to transmitted light as determined by means of a photometer. For example, if equal opacities are obtained when the amount of air sampled, per unit area of dust spot, is four times as great for the downstream paper as for the upstream paper, the efficiency of the filter is considered to be 75 percent, since the dustiness of the cleaned air, as measured by change in opacity, is one-fourth that of the uncleaned air. To obtain equal opacities, samples may be drawn at equal rates for equal times through unequal spot areas, or at equal rates through equal areas for unequal times, as best suits the circumstances. Tests may be made with atmospheric air, or with an artificial dust dispersed in the air stream.

Dust-holding capacity is defined as the amount of dust which a filter can retain and have a resistance less than some arbitrary value. The term applies only to fixed-type air cleaners. Under the AFI[6] and the former ASHVE[7] procedures dust-holding capacity is defined as the product of the weight of dust reaching the filter and the average efficiency at a specified resistance and air flow. Dust loading and arrestance determinations are made simultaneously, arrestance and resistance being determined for each increment of feed with the results plotted against weight of dust fed or against accumulated load in the filter.

Dirt-loading measurements are made at the National Bureau of Standards[9] by feeding a mixture of 4 percent cotton linters and 96 percent Cottrell precipitate (by weight) to the test filter. The two ingredients are separately dispersed into the air stream, using two aspirating-type injectors. The measurements show the relationship between the pressure drop and the weight of dirt mixture received at the filter. The curves in Fig. 4 illustrate the difference in the dirt-loading characteristics of two filters, one a viscous-impingement type and the other a dry filter with an extended-surface cellulose-fiber medium, with dust and with lint.

SELECTION AND MAINTENANCE

To evaluate filters and air cleaners properly for a particular application, two factors should be carefully weighed: (1) the degree of air cleanliness required and (2) disposal of the dirt after it is removed from the air. These factors affect initial costs, operating costs, and the extent of maintenance that will be required. Savings that accrue through reduction in housekeeping expenses, protection of valuable property and equipment, ability to carry on dust-free manufacturing processes, improved working conditions, and even health benefits should be credited against the cost of installing and operating an adequate system. The capacity and physical size of the unit required may emphasize the need for self-cleaning features. Operating costs, predicted life, and efficiency are more important than first cost, because air cleaning is a continuing process.

While electronic air cleaners have a higher first cost, they exhibit very high efficiencies in cleaning atmospheric air, due largely to their ability to remove fine dust which shows no gravitational effect. They are equally effective on coolant oil mists from high-speed cutting and grinding machines. System resistance remains unchanged as dirt is being collected, and the resulting residue is disposed of directly to prepare the equipment for further duty. The charged-media type cleaners are less effective than the ionizing type, but are appreciably better than mechanical filters in the matter of small particle collection.

The advantage of the moving-curtain impingement type filter consists in the small amount of attention which it requires. Such devices are therefore to be recommended where labor is scarce, or where reliable and frequent attention to filters cannot be assumed. The constant pressure drop of this type of filter is an advantage. The first cost is substantially greater than that of unit filters, and the dust arrestance may not be any higher.

Unit filters constitute the majority of air cleaners now in use, and some choice is possible between the types avail-

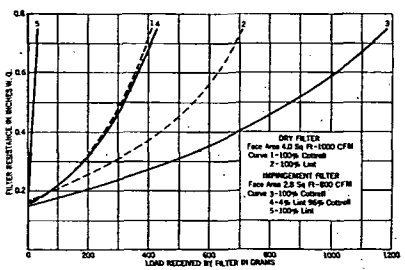


Fig. 4 .... Dust and Lint Holding Capacity of Two Filters

able. Where lint in an eminently dry state predominates, a dry filter obviously may be preferable to other types because of its lint-holding capacity. If the lint is greasy, or if oil vapor exists in the air, the dry filter, if it is of the cleanable type, may be troublesome, since grease tends to make it difficult to clean. Most dry types, however, employ a throw-away type of medium which is held in permanent metal frames so that the difficulty of cleaning the medium is avoided. Some dry filters are capable of high efficiencies, compared to other unit filters on fine particles, but their dust-holding capacity for such dust may be inferior to that of the viscous-impingement type.

Viscous-impingement unit filters are practically standard in size, and their overall dimensions are small when compared with their ratings.

Throw-away units are often installed in series so that the one in front, which usually becomes plugged with lint, can be discarded, after which the downstream unit is moved to the front and replaced by a new unit.

Viscous-impingement unit filters do not have efficiencies as high as can be expected with some other types of unit filters, but their first cost and upkeep are generally lower, whether of the cleanable or the throw-away type. They require more careful attention than the moving-curtain type if the resistance is to be maintained within reasonable limits.

FILTER INSTALLATION

Many air cleaners are available in units of convenient size for handling when installing, cleaning, or replacing. Such units are usually designated as filters or unit filters. A typical unit filter may be 20 in. square and from one to several inches thick, depending on the manufacture and proposed use. In large systems, a number of such units are installed adjacent to each other and collectively called a bank of filters.

Air cleaners are commonly installed in the outdoor-air intake ducts of buildings, and generally in the recirculating and bypass air ducts as well. Cleaners are logically placed ahead of heating or cooling coils and other air-conditioning equipment in the system to protect them from dust. The character of the dust arrested by the filters in an air-intake duct is likely to be mostly particulate matter of a greasy nature, while lint may predominate in dust from within the building.

The published performance data for all air filters are based on *straight-through* unrestricted air flow. Filters should be installed so that the face area is at right angles to the air flow whenever possible. Eddy currents and dead air spaces should be avoided, and air should be distributed uniformly over the entire filter surface, using baffles or diffusers if necessary.

Failure of air-filter installations to give satisfactory results can, in most cases, be traced to faulty installation or improper maintenance or both.

The most important requirements of a satisfactory and efficiently operating air filter installation are:

1. The filter must be of ample size for the amount of air it is expected to handle. An overload of 10 to 15 percent is regarded as the maximum allowable. When air volume is subject to increase, a larger filter should be installed.
2. The filter must be suited to the operating conditions, such as degree of air cleanliness required, amount of dust in the entering air, type of duty, allowable pressure drop, operating temperatures, and maintenance facilities.
3. The filter type should be the most economical for the specific application. The first cost of the installation should be balanced against efficiency and depreciation as well as expense and convenience of maintenance.

The following recommendations apply to filters installed with central fan systems:

1. Duct connections to and from the filter should change size or shape gradually to insure even air distribution over the entire filter area.
2. Sufficient space should be provided in front as well as behind the filter to make it accessible for inspection and service. A distance of two feet may be regarded as the minimum.
3. Access doors of convenient size should be provided in the sheet-metal connections leading to and from the filters.
4. All doors on the clean-air side should be lined with felt to prevent infiltration of unclean air. All connections and seams of the sheet-metal ducts on the clean-air side should be as airtight as possible.
5. Electric lights should be installed in the chamber in front of and behind the air filter.
6. Filters installed close to an air inlet should be protected from the weather by suitable louvers, in front of which a large mesh wire screen should be provided.
7. Filters, other than electronic air cleaners, should have permanent indicators to give a warning when the filter resistance reaches too high a value.

Safety Requirements

An investigation of safety ordinances should be made by the engineer when the installation of an air cleaner of any considerable size is contemplated. It is possible that combustible filtering media may not be permitted in accordance with some existing local regulations. Combustion of dust and lint on a filtering medium is possible, though the medium itself may not burn.

ADSORPTION OF VAPORS OTHER THAN WATER

Many of the foreign gases in the atmosphere are selectively adsorbed by charcoal. Included are many of the organic gases, such as those emanating from animals and people, some of the gaseous constituents of combustion, alcohols, ketones, esters, and gaseous products of putrefaction.

Charcoals differ widely in their adsorptive capacity. Those which have marked adsorption characteristics, such as properly prepared coconut shell charcoals, are sometimes called *activated charcoals* or *activated carbon*.* These materials can adsorb approximately 50 percent of their own weight of many organic gases at 70 F. The charcoal may be used for a long time by reactivation at high temperatures, under which condition it gives up the adsorbed gases. Temperatures of approximately 1000 F are desirable for reactivation. Charcoals for use in air handling systems should be able to stand physical handling, including reactivation, without excessive loss by breakage or dusting.

As applied in air handling systems, the charcoal is placed in perforated metal containers which are grouped in frames and set in the air stream. The percentage removal of an organic gas, such as carbon tetrachloride, is 95 percent or above, when placed in intimate contact with the carbon at 70 F. In commercial apparatus there may be a bypass effect which depends on the physical arrangement of the charcoal containers. This bypassing reduces the percentage removed in the total gas passing through the adsorber. Resistance to air flow is usually selected within the general range of resistance of impingement filters.

The required quantity of recirculated air to be treated is determined by dividing the requirements for contaminant-free air, *minus the outdoor air*, by the fraction denoting the percentage removal of the gas in question in the adsorber bank which is to be used.

Adsorbers may be applied to reduce objectionable gases entering through the outdoor air inlet. They may also be used to reduce the odors caused by exhausts from processing. Adsorber beds, in all cases, should be protected from dust, free oil, and grease.

PART II—INDUSTRIAL AIR AND GAS CLEANERS

Industrial development and growth of industrial areas have had a cumulative effect upon the problem of controlling contaminants. Not only has the atmosphere in many cities become more polluted, but the intensity of pollution at the points of control has increased. Accompanying this increase in pollution there has been a growing consciousness of the need for more effective air cleaning among housewives, store managers, industrialists, and legal inspectors, and this has resulted in increasing severity of regulations pertaining to collection of dust and contaminants.

Air cleaning for the supply system is usually accomplished by means of some type of air filter. To prevent escape of industrial dust into the atmosphere, some type of collector is required. An industrial air-cleaning installation is designed to perform one or more of the following 6 functions:

1. Prevent a nuisance or physical damage to an individual, a plant, or adjacent property.
2. Prevent re-entry of contaminants to working spaces.
3. Reclaim usable material.
4. Reduce fire, explosion, or other hazards.
5. Permit recirculation of cleaned air to working spaces.
6. Allow utilization of cleaned gases for processes.

DEGREE OF AIR CLEANING REQUIRED

The amount of material which can be discharged to the atmosphere is established by local or state regulations prepared by pollution control, labor, or health departments. Standards are established to prevent injury to persons or property. The need for control within the plant is often of prime interest to the industrialist for protecting his equipment, improving his product, and providing a clean working environment. The re-entry of contaminants to working spaces is prevented by effective air-cleaning devices since only a clean effluent is discharged to the ambient atmosphere. Public nuisance complaints often occur even when the effluent concentration discharged to the atmosphere is below the permissible limits of concentration and visibility. Plant location, contaminants involved, and meteorological condition of the areas must be evaluated in addition to existing regulations or codes of good practice.

Fire and explosion hazards are created by combustible dusts, vapors, or gases. Safety may be affected by loss of visibility, settlement, or accumulation of various materials. The cleaning device can prevent these hazards by providing an effluent below the inflammation limit, visibility index, and point where settlement or accumulation will occur.

The degree of air cleaning for the recovery of usable material is a matter of economic evaluation which will vary with such factors as quantity and value of material collected, capital cost, and operating cost.

Air cleanliness must be of the highest order where toxic materials are involved and the cleaned air is recirculated to the workroom. Such recirculation is considered poor practice and is prohibited by many regulations where toxic materials are involved, except for those cases where discharge to atmosphere is impossible or decidely impracticable. Where air is recirculated, its contamination must not exceed the established maximum allowable concentrations listed in Chapter 7. Usual requirements are a fraction, often 1/5 to 1/2, of this standard, depending on: regulation involved, air quantities recirculated in relation to the cubical content of working space, and the presence of other exhaust systems discharging to the atmosphere.

Cleaning requirements for the utilization of cleaned gases are dependent on the process and are generally of the highest order (usually comparable to atmospheric air cleanliness).

Regardless of standards it is good practice to install the most effective collection equipment available in the light of practical operation features, installation, and equipment costs. This is warranted because the required degree of cleaning is increasing continually.

FACTORS AFFECTING SELECTION

Selection of an industrial air cleaner for a given application requires an evaluation of the following 5 considerations:

1. Concentration, particle size, and size distribution of the contaminant.
2. Degree of cleaning required.
3. Conditions of air or gas stream with reference to temperature, moisture content, and chemical composition.
4. Characteristics of the contaminant, corrosiveness, solubility, adhesion, or packing, and its specific gravity, surface, and shape.
5. Methods of disposal or salvage that meet the conditions imposed by material, process, or plant location.

In view of the range of these variables, all available test data, application experience, and manufacturers' performance data should be considered.

TYPES AND APPLICATION

There are a number of principles involved in the operation of industrial air and gas cleaners. They are illustrated by the following types of cleaners:

1. *Inertial Separators* (Gravitational or inertial force principle): Baffle Chambers, Centrifugal Collectors.
2. *Scrubbers and other Wet Collectors:* Air Washers, Wet Filters or Packed Towers, Spray Towers, Centrifugal or Inertial Scrubbers, Venturi Scrubbers.
3. *Filters:* Fabric Filters, Ultra or Absolute Filters.
4. *Electrostatic Precipitators.*
5. *Adsorbers, Absorbers, and Combustion Devices.*

Units employing these principles may apply them singly or in combination. The important characteristics of any unit are: its collection efficiency, resistance to air or gas flow (power requirements), ability to maintain specified air or gas flow during its operating cycle, and maintenance requirements.

In relation to industrial air cleaners, efficiency of collection is generally presented on a basis of weight removal. In many instances, collection efficiency is expressed in terms of weight removal in specific particle size ranges. Occasionally efficiencies are given in terms of visibility or photometric evaluation. Under other circumstances efficiencies are stated by particle count. For most process recovery problems and air pollution control, the weight basis has been adequate. When loss of visibility is the major problem, light opacity measurements may be required.

From a hygienic viewpoint where air is to be recirculated, only the contaminant concentration in the recirculated air stream, rather than the collection efficiency, should be considered.

Correlation between the various efficiencies is poor due to the wide variation in particle size, size distribution, and shape.

Resistance or pressure drop through any given collector is related to the operating principle, and varies about as follows: low resistance collectors—up to 2 in. water; medium resistance collectors—2 to 8 in. water; high resistance collectors—8 to 25 in. water.

In mechanical centrifugal collectors of the wet or dry type (power driven impellers or rotors) resistance alone is not of significance since these devices are also air movers and therefore the factor of paramount importance is the overall power consumption of the unit. This is also true of certain types of wet collectors which require auxiliary pumping power.

The ability of any unit to maintain specified air or gas flow during its operating cycle is also a function of its operating principle. For example, inertial collectors of the simple and centrifugal type are able to maintain constant resistance regardless of concentration and particle size. However, adhesive materials may collect at the entry, throughout the unit, and at the exit, and thus cause a reduction in air flow and serious maintenance problems. With the exception of packed towers and wet filters, wet collectors operate at constant resistance throughout their operating cycles. However they may also be adversely affected by adhesive or corrosive materials. Wet filters or packed towers will eventually become clogged because of retention of inert dusts. Units employing fabric or other filter media will have resistance influenced by the concentration, particle size, shape, and packing characteristics of the aerosol. Their overall resistance during the operating cycle may increase gradually over a known or determined period. Auxiliary cleaning devices such as separate compartments, alternately shaken, reverse air flow, or reverse jet air flow may maintain constant or nearly constant resistance or insure only small fluctuations in air or gas flow. Electrostatic precipitators have the outstanding characteristic of maintaining a constant air flow with an accompanying low resistance. Adsorbers behave similarly to packed towers in that they provide constant resistance until the voids become seriously plugged with inert dusts. This generally requires a matter of several months on normal applications.

It has been found that inertial separators require the least maintenance. Simple wet collectors rank next in this respect. Complex wet collectors and fabric arrestors require maintenance on a definitely scheduled basis. Electrostatic precipitators require the usual maintenance connected with high-voltage equipment and may vary widely. The maintenance of adsorber type units is minimal and is based primarily on the amount of inert material encountered.

INERTIAL SEPARATORS

There are many types in this group. These range from simple settling chambers utilizing gravitational forces, to complex centrifugal devices using abrupt directional changes or motor driven rotors. They all have the desirable features of constant resistance during their operating cycle and do not require any special construction for temperatures under 750 F. Usually they require less maintenance than other types of collectors.

Table 1 . . . . Inertial Separator Characteristics

Kind	Efficiency Particle Size Range In Microns			Resistance (Pressure Drop)	Description and Remarks
	>100	100–10	<10		
Settling and baffle chambers	Medium	Low	Low	Low	A large chamber to reduce velocity to permit settling. Auxiliary baffles are sometimes used to improve performance. Requires large space.
Large diameter cyclones	High	Medium to low	Low	Low	Considered as single body over 3 ft in diameter. Gases and material must be dry. Air-tight dust bins or continuous removal of material must be provided. Space requirements moderate.
Refined design centrifugals	High	High	Medium to low	Medium	Available as multiple small-diameter tubes, scroll-shaped bodies, and multiple louvers. Gases and material must be dry. Air-tight dust bins or continuous removal of material must be provided. Space requirements moderate.
Integral rotor type	High	High	Medium to low	See remarks	Rotor acts as air mover and separating element. Usually limited to dry granular materials and dry gases. Low space requirements. No external pressure loss involved but power requirements are higher than usual exhauster.
Slotted scroll fan	High	Medium to low	Low	See remarks	Separation takes place in fan scroll. No external loss involved but power requirements are higher than usual exhauster.

On the basis of their important characteristics inertial separators may be classified according to Table 1.

SCRUBBERS AND OTHER WET COLLECTORS

In these collectors many methods are used to wet the contaminant and remove it from the air stream. A number of wet collectors are similar in design to dry collectors with the addition of sprays or liquid contactors. The mechanism of separation in wet collectors depends on inertial forces such as impaction and impingement, and in some cases, diffusion.

Wet collectors do not require any special construction for high temperatures. They are suitable for use on mixed contaminants such as gases and solid or liquid particles. The collection of dust in a wetted form eliminates a secondary dust problem in disposal of collected material. Protection against corrosion and freezing may be necessary and disposal of waste liquids may become a problem. With the exception of the packed towers and wetted filters they all have a constant resistance during their operation.

Table 2 . . . . Characteristics of Wet Collectors

Kind	Efficiency Particle size range in microns			Resistance In. Water	Water Rate In Gallons Per 1000 Cu Ft Gas	Maintenance	Description and Remarks
	>10	10–1	<1				
Air washers	High	Medium	Low	Low (less than 1)	5–15	Medium	Composed of one or more banks of low-pressure sprays and flooded eliminator plates. Space requirements are large. Water is usually recirculated.
Wet filters	High	High	Medium	Low (less than 1)	3–8	Medium	Composed of one or more banks of low-pressure sprays flooding packed cells of glass or other fibers. Space requirements are large. Limited to low loadings of solid particles.
Packed towers	High	High	Medium	High	5–10	High	Composed of granular or ceramic packing. Space requirements large. Generally limited to gas absorption.
Spray towers	High	Medium	Low	Low	5–10	Low	Usually a vertical chamber equipped with low-pressure water sprays at the top or along the walls. Baffle plates may be provided for contacting. Space requirements medium. Usually applied for hot gases.
Centrifugal or inertial scrubbers	High	High	Medium to high	Medium (see remarks)	½–40	See remarks	This classification includes most of the commercial wet collectors. They may utilize singly or in combination, low- or high-pressure sprays, wetted plates, rotating members, liquid gas contact by induction or centrifugal action. Some of these units (disintegrators) have high overall power requirements for driving the rotor while others have high power consumption for raising water pressures for spray nozzles. Maintenance varies from low to high depending on component parts. Simple gas liquid contact units would be classified as low. Units having rotating members may vary from medium to high depending on the type of service.
Venturi scrubber	High	High	High	High (12–25)	5–10	Medium	Consists of a simple venturi with low-pressure sprays at throat followed by a cyclonic type separator. Space requirements are low. Unit functions independent of grain loadings. High velocity at venturi throat may cause erosion problems.

On the basis of their important characteristics wet collectors may be classified according to Table 2.

FILTERS

Filters for industrial air and gas cleaning consist primarily of cloth dust collectors and the ultra or absolute type of filter.

Cloth dust collectors are generally constructed in the form of housings with cloth envelopes drawn over wire screen frames or with vertical cloth bags or tubes. Cloth filters depend to a great extent on the accumulated dust

Table 3 . . . . Characteristics of Fabric Filters

Kind	Efficiency, Particle size range in microns: >10	10 to 1	<1	Resistance	Filtering Velocity fpm	Maintenance
Cloth collector (shaken or rapped)[a]	High	High	Moderate to high	Medium	1–6	Medium to high (periodic)
Cloth collectors (reverse air or jet cleaning)[b]	High	High	Moderate to high	Medium to high	10–35	Medium to low
Absolute type[c]	High	High	High	Medium	3–14	None (see remarks)[c]

[a] Space requirements large. Requires scheduled cleaning.
[b] Space requirements medium. Cleaning constant during operating cycle. Added horsepower used for reverse jet action.
[c] Space requirements small. Requires replacement of unit when predetermined resistance is attained. Ordinary cleaning not possible.

layer for their effectiveness. Their basic or initial efficiency depends upon the type of fabric employed; felted media such as wool gives higher initial removal than woven fabrics. The increase in resistance to air flow, resulting from the accumulated dust layer, is related to rate of flow, concentration, and characteristics of the contaminant. Removal of the collected material is accomplished through periodic shaking by single mechanisms or combinations of them employing rapping, or through reversal of air flow, or by means of high-velocity reverse air jets. With shaking or rapping, air flow is not constant and filtering velocities are usually held under 6 fpm. With reverse jets or reverse air flow, substantially constant air flow is maintained and filtering velocities are generally higher (up to 35 fpm). Filtering velocities in all fabric collectors are highly dependent upon the aerosol, fume, or dust being collected.

The ultra or absolute type of filter[2] may consist of pleated cellulose asbestos paper, deep sand beds, composite glass wool layers, compressed glass fiber in the form of paper, or batts and resin wool. The use of this type of filter is limited to concentrations in the range of outdoor air where very high efficiencies, greater than 99.95 percent are required.

Fabric filters are not suitable for high temperatures, above 300 F, acid mists or vapors, or mixed contaminants such as wet corrosive gases and solid or liquid particles. Cotton fabrics are limited to 180 F, whereas wool is limited to temperatures under 200 F. A wide variety of natural and synthetic filter media are available for various temperature and corrosion conditions but caution must be exercised relative to rupture, high temperatures, and corrosive gases. When properly selected, efficiencies are often high enough to permit recirculation of air except, possibly, when handling toxic contaminants. On the basis of their important characteristics, fabric filters may be classified according to Table 3.

ELECTROSTATIC PRECIPITATORS

Electrostatic precipitators for industrial air and gas cleaning differ materially from the low-voltage designs described in Part I of this chapter, although the principles of operation are similar. For industrial concentrations, it is obvious that more severe demands are made upon methods of cleaning the collector, disposal of collected material, and servicing practices. Low-voltage cleaners for industrial loadings to date do not have sufficient inherent dust-holding capacity. One exception in the field of exhaust systems is that of the oil-mist collector, which functions satisfactorily on a liquid aerosol with the conventional low-voltage type (2 stage) electrostatic precipitator.

Industrial precipitators employ an assembly of parallel collector electrodes of various constructions, including corrugated plate, rod curtains, or perforated plate. Air flow is usually horizontal, although special construction may permit vertical air flow. High-voltage collectors were made in the form of vertical pipes, and are used for high operating pressures and for *wet-collector* designs where water continuously flows downward inside the pipe walls of the collector electrode. The negative discharge electrodes or rods are accurately centered between the usual 9 in. collector electrode spacing, the latter being of positive charge. Precipitation occurs in a single stage wherein ionization and collection are carried on simultaneously throughout the unit, and depend on high potentials of 25,000 to 75,000 volts. The high-voltage direct current is supplied by either mechanical or electronic rectifiers.

High efficiencies are obtained by allowing suitable time for contact in the collector zone, and by proper ratio of air flow velocity to that of transverse velocity of the negatively charged particles toward the positively charged collector plates.

Air velocities vary from 240 fpm to 480 fpm with a constant pressure drop of less than ½ in. water. Since the maximum efficiency is obtained with the finer particles at low concentrations, and the precipitation is not effective for large particles, inertial separators are frequently used in series with these units.

Attention should be directed to the prohibition of air recirculation to occupied spaces. High-voltage cleaning equipment produces ozone and nitrogen oxides in excessive quantities. The major source of maintenance usually will be in conjunction with the high-voltage rectifying equipment rather than the collector. Although usually used for elevated temperature work, unless the gases are preconditioned, temperatures in excess of 700 F should be avoided.

Collector electrode cleaning is accomplished by a rapping device, either electrical or pneumatic, without interrupting operation. The dry plate surfaces release the dust into hoppers below the plates. Some wet cleaning methods are also used.

Electronic precipitators require large spaces, maintenance may vary widely, and initial costs are high. Efficiency is a function of the aerosol encountered and usually ranges above 90 percent.

ABSORBERS, ADSORBERS, AND COMBUSTION DEVICES

One of the important problems in industrial air and gas cleaning is the removal of soluble, insoluble, or combustible gases and vapors from fluid streams before their discharge to the atmosphere. These contaminants may create problems which are distinct from the particulate removal discussed previously. It has been pointed out that scrubbers and other wet collectors may remove gases and particulate matter from contaminated air streams. However, the units

described are intended primarily for particulate removal with the exception of the spray or packed towers or wet cell devices.

Absorption devices are intended primarily for the removal of readily soluble gases which can be removed by simple absorbing agents such as water or alkali; for example, the removal of hydrochloric acid gas by a caustic spray.

Adsorption apparatus is intended primarily for removal of organic vapors in either high concentration (solvent recovery) or low concentration (odor removal). The first process is usually carried on as a matter of economic recovery. In the second process, the concentration of the contaminant must be low enough so that the adsorbent will have a reasonable life before replacement or reactivation.

Combustion, either total or catalytic, may be used to destroy organic compounds, either gaseous or particulate, which create odor problems. The use of catalysts to reduce the temperature needed for combustion is comparatively recent in this field. In some instances the heat of combustion may be recovered for utilization. The temperature necessary for destruction of compounds varies with the nature of the contaminant. However, partial combustion may alter the compound to render it innocuous. The temperature required for complete destruction is usually above 1100 F. Catalysts may only require heating to 500 F or may operate without heating if the gas temperature is above 500 F.

Absorbers

These may consist of spray chambers, packed towers, or wet-cell washers through which an absorbing agent is recirculated. The gases collected may be converted to insoluble salts or usable acids and other compounds. The performance of these devices depends upon several factors such as the solubility of the gas, its vapor pressure, its rate of reaction with the absorbent, the velocity through the collector, and area of the absorbing surface either as spray droplets or wetted media. These are generally designed for the specific purpose intended. They usually are intended to collect over 95 percent of the contaminant. Resistance depends upon the particular design and ranges from 1 to 10 in. of water in most installations. Their resistance during their operating interval is constant unless serious plugging of packing occurs. Spray units are seldom affected by plugging. Packed absorption devices require a precleaner to remove particulates if long trouble-free service is desired. Maintenance of spray nozzles is a function of the degree of atomization used and of conditions encountered, such as purity of spray liquids.

Protection against corrosion and freezing may be necessary and disposal of waste liquids may create secondary problems.

Adsorbers

These units consist of a chamber filled with a granular adsorbent. The adsorbent may be activated charcoal, silica gel, alumina, and other treated solids. Activated charcoal is the most common and has the highest retention per unit weight for organic solvents. Performance of adsorbent beds depends upon their thickness, velocity of the gas passing, mesh size (surface area) of adsorbent, and temperature of the vapor being removed. They can be designed to obtain almost complete removal of organic vapors and some inorganic gases. They may be reactivated by the use of steam or heat. Resistance depends upon the adsorbent mesh size, depth, and velocity. It may range from less than one inch of water to several inches of mercury.

Adsorbent beds are usually limited, because of their cost, to applications where recovery may be economically feasible. Removal of particulate matter by use of precleaners is necessary if significant loadings are involved as the adsorbent voids are readily plugged. They are not suitable for high temperature conditions. Silica gel and alumina exhibit a strong preference for water vapor whereas charcoal does not.

Combustion Devices

This group of industrial air cleaners entails the use of high temperatures or catalysis combined with some elevation in temperature to destory or decompose organic and some inorganic gases which create obnoxious odors. The devices are divided into those in which combustion is obtained by use of liquid or gaseous fuels and the secondary air for combustion is provided by the contaminated air. In essence, the simplest form involves passing the contaminated air through the combustion chamber of a boiler or fire box. Special fire boxes or brick checkerwork may also be employed. Catalytic devices consist of noble metal packed into frames in the form of ribbons or screens or may be ceramic granules coated with noble metal catalysts. The frames or packing are placed in a housing over which the gas stream to be decontaminated is passed. The contaminants break down and are reduced to elemental gases of an innocuous nature. Catalysts reduce the amount of heat necessary but may become contaminated by sulfur or other elements.

The performance of a combustion device is dependent upon the retention period of the contaminant in the high temperature zone. Removal or destruction is thus dependent on velocity and surface area of the heat-transfer device. If properly designed, obnoxious odors are removed.

The resistance of the direct combustion unit is negligible since it is an integral part of the fuel-combustion system. Catalytic devices behave in a manner comparable to packed beds or filters except that they are not plugged by organic particulates but may be affected by inorganic solids.

Maintenance of these types of units is essentially dependent upon the contaminants encountered. The direct-combustion system requires little care other than that required by ordinary fuel-burning equipment. Catalytic units become contaminated slowly and will require removal for reactivation at certain intervals depending upon the application.

Economic factors may limit the use of direct-combustion methods unless there is a demand for the heat generated. They may best be applied to processes employing combustion. Catalytic units require temperatures of at least 500 F and consequently it may be necessary to preheat the contaminated air to this value.

REFERENCES

[1] S. S. Leopold and C. S. Leopold: Bronchial asthma and allied allergic disorders (*Journal of the American Medical Association*, March 7, 1925, p. 731).

[2] L. H. Criep, M.D., and M. A. Green, M.D.: Air cleaning as an aid in the treatment of hay fever and bronchial asthma (*Journal of Allergy*, January 1936, p. 120).

[3] O. M. Lidwell: The bacterial filtration efficiency of an electrostatic air cleaner (*Journal of the Institution of Heating and Ventilating Engineers*, London, June 1951, p. 139).

[6] R. A. Nielsen: Dirt pattern on walls (ASHVE TRANSACTIONS, Vol. 46, 1941, p. 247).

[8] P. Drinker and T. Hatch: *Industrial Dust* (McGraw-Hill Co., New York).

[9] *Code for Testing Air Cleaning Devices Used in General Ventilation, Section I, Unit or Panel Type Air Filtering Devices* (Air Filter Institute, 1953).

[7] *ASHVE Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work* (ASHVE TRANSACTIONS, Vol. 39, 1933, p. 225).

[8] R. S. Dill: A test method for air filters (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 379).

[9] H. L. Barnebey: Activated charcoal for air purification (ASHAE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, March 1958, p. 153).

[10] *Laboratory Design for Handling Radioactive Materials* (Building Research Advisory Board, National Research Council, Washington, D. C., *Research Conference Report*, 1952).

BIBLIOGRAPHY

H. C. Murphy: Design and application of oil-coated air filters (ASHVE TRANSACTIONS, Vol. 33, 1927, p. 73).

W. G. Frank: Operation and maintenance of air filters (*Heating, Piping and Air Conditioning*, May 1931, p. 378).

W. G. Frank: Size and characteristics of air-borne impurities (*Heating, Piping and Air Conditioning*, January 1932, p. 35).

O. Wechsberg: Fundamental principles in the design of dry air filters (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April 1933, p. 217).

H. E. Ziel and Henry Sleik: The economic factors in converting recirculated air for ventilation (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July 1943, p. 367).

F. B. Rowley and R. C. Jordan: ASHVE RESEARCH REPORT No. 1094—Air filter performance as affected by kind of dust, rate of dust feed, and air velocity through filter (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 415).

F. B. Rowley and R. C. Jordan: ASHVE RESEARCH REPORT No. 1122—Air filter performance as affected by low rate of dust feed, various types of carbon, and dust particle size and density (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 339).

F. B. Rowley and R. C. Jordan: ASHVE RESEARCH REPORT No. 1145—The effect of lint on air filter performance (ASHVE TRANSACTIONS, Vol. 46, 1940, p. 25).

F. B. Rowley and R. C. Jordan: ASHVE RESEARCH REPORT No. 1169—Comparison of the weight, particle count and discoloration methods of testing air filters (ASHVE TRANSACTIONS, Vol. 47, 1941, p. 29).

F. B. Rowley and R. C. Jordan: ASHVE RESEARCH REPORT No. 1187—Economical air velocities for mechanical air filtration (ASHVE TRANSACTIONS, Vol. 47, 1941, p. 391).

F. B. Rowley and R. C. Jordan: ASHVE RESEARCH REPORT No. 1218—Overloading of viscous air filters during accelerated tests (ASHVE TRANSACTIONS, Vol. 48, 1942, p. 437).

G. W. Penney: A new electrostatic precipitator (*Electrical Engineering*, January 1937, p. 159).

H. E. Corbitt and N. J. Clark: Electrostatic precipitation for aircraft (*Aero Digest*, December 1940, p. 132).

C. E. Miller: Pointers on selecting equipment for industrial gas cleaning (*Chemical and Metallurgical Engineering*, March 1938, p. 132).

W. A. Schmidt and E. Anderson: Electrical precipitation (*Electrical Engineering*, August 1938, p. 332).

A. W. Simon and L. C. Kron: Electrical precipitation (*Electrical Engineering*, February 1932, p. 93).

E. Anderson: Some factors and principles involved in the separation and collection of dust, mist, and fume from gases (*American Institute of Chemical Engineers Transactions*, Vol. 16, 1924, p. 69).

R. B. Rathbun: Electrical precipitation of solids from smelter gases (*American Institute of Electrical Engineers Transactions*, Vol. 41, 1922, p. 815).

A. C. Stern, J. Baliff, A. E. Perina, R. Crowley, B. Feiner, and A. A. Urbano: Characteristics of unit dust collectors (ASHVE TRANSACTIONS, Vol. 52, 1946, p. 237).

J. M. Kane: Operation, application and effectiveness of dust collection equipment (*Reference Section, Heating and Ventilating*, August 1952).

C. A. Lapple: Chapter 9 (*Air Pollution Abatement Manual*, Manufacturing Chemists Association).

S. K. Friedlander, L. Silverman, P. Drinker, and M. W. First: *Handbook on Air Cleaning* (U. S. Atomic Energy Commission, Washington, D. C.).

M. W. First, R. Moschella, L. Silverman, and E. Berly: Performance of wet-cell washers for various aerosols (*Industrial and Engineering Chemistry*, Vol. 43, 1951, p. 1363).

J. H. Perry: *Chemical Engineers Handbook* (McGraw-Hill Co., New York, 1950, 3rd ed.).

American Industrial Hygiene Association Quarterly, March 1950.

R. J. Ruff: Design factors in catalytic fume elimination (*Heating and Ventilating*, September 1953, p. 84).

A. Nutting and R. F. Logsdon: New air filter code (*Heating, Piping and Air Conditioning*, June 1953, p. 77).

CHAPTER 25

SOUND CONTROL

Acoustical Terminology; Apparatus for Measuring Sound; Approaches to the General Problem of Noise Control; Criteria for Noise Control; Kinds of Noise; Noise Generated by Fans, Grilles, and other Sources; Sound Attenuation in Ducts; Determination of Room Levels; Cross Transmission between Rooms and through Duct Walls; Controlling Vibration from Machine Mountings

THE NOISE created by heating, ventilating, and air-conditioning equipment has become an important factor in modern building design. Related to this, and of equal importance, is the problem of the transmission of speech or music from one part of the building to another through ventilating and air-conditioning ducts. The architect and the acoustical engineer cooperate to produce rooms that are satisfactory for speech, music, or other intended uses. The ventilation and acoustical engineers cooperate to ventilate and air condition these rooms to be physically comfortable without adding noise in excess of established requirements. This chapter is planned to supply part of the information needed to achieve adequate quietness in rooms that are ventilated and air conditioned through supply ducts. The quieting of unit heaters and coolers is assumed to be the job of the manufacturers and is not covered here. Remaining information on cost, availability, durability, and ease of installation can come only through experience with practical installations.

ACOUSTICAL TERMINOLOGY[1]

Sound Power W is the power in watts produced by a source of sound. This power may be (*a*) the total power radiated by the source over its entire frequency range, (*b*) the power radiated in a limited frequency range, or (*c*) the power radiated in each of a series of contiguous frequency bands. The frequency range or frequency band should be clearly stated.

Sound Intensity I is the power radiated in a specified direction through unit area normal to this direction, e.g., watts per square meter, watts per square foot, or watts per square centimeter.

Sound Pressure p is the root-mean-square incremental pressure produced when a sound wave passes through an otherwise undisturbed medium. The unit is the microbar (1 dyne per sq cm or 0.1 newton per sq m).

Frequency f is the rate of repetition of a periodic phenomenon. The frequency is the reciprocal of the period, or the time necessary for the phenomenon to repeat. The frequency f of a single tone sound wave is equal to the ratio of the speed of sound c to the wavelength λ of the tone. $f = c/\lambda$, or $c = f\lambda$. The unit is the cycle per second, or cps.

Decibel db is a dimensionless unit for expressing the ratio of two numerical values (usually electrical or mechanical power) on a logarithmic scale. The number of decibels is ten times the logarithm to the base 10 of the numerical ratio of the two quantities. For example, let W_1 and W_2 designate two powers; or I_1 and I_2 designate two sound intensities, and let M designate the number of corresponding decibels, then

$$M = 10 \log_{10} \frac{W_1}{W_2} \text{ decibels} \quad (1)$$

or,

$$M = 10 \log_{10} \frac{I_1}{I_2} \text{ decibels} \quad (2)$$

Alternatively, the ratio of two sound pressures squared is expressed in the same units,

$$M = 10 \log_{10} \frac{p_1^2}{p_2^2} = 20 \log_{10} \frac{p_1}{p_2} \text{ decibels} \quad (3)$$

These relations tacitly assume that: $W = p^2/Z$, which means that Z, the acoustic impedance, is real, and that for p_1 and p_2, $Z_1 = Z_2$. This is not usually the case, but because sound pressure is the easiest variable to measure in a sound field, the relation:

$$M = 20 \log_{10} \frac{p_1}{p_2} \text{ decibels} \quad (4)$$

is commonly used. Thus, in a sense when used in this way, the decibel is redefined on the basis of a pressure squared ratio rather than a power ratio.

Octave Frequency Bands. The frequency range of a noise is frequently broken up into octave frequency bands where, in principle, the upper frequency is twice the lower frequency of the band as shown in the following table.

Band Number	Usual Frequency Limits	Approximate Geometric Mean Frequency
1	20 to 75 cps*	40
2	75 to 150	105
3	150 to 300	210
4	300 to 600	425
5	600 to 1200	850
6	1200 to 2400	1700
7	2400 to 4800	3400
8	4800 to 10,000*	6900

* The highest and lowest bands as usually used, contain somewhat more than one octave as indicated.

Sound Power Level PWL is ten times the logarithm to the base 10 of the ratio of the sound power to the reference power of 10^{-13} watt. Thus, Power Level in the English system[2]

$$\text{PWL} = 10 \log_{10} \frac{W}{10^{-13}} \text{ dbe} \quad (5)$$

where

W = sound power, watts.

The unit is the *dbe*. The symbol *dbe* means that the sound

power level is convenient to use in the English system and that 10^{-13} watt is the reference power.

Sound Intensity Level (L_I) is ten times the logarithm to the base ten of the ratio of the sound intensity I to the reference intensity of 10^{-16} watt per sq cm (or its equivalent of 1 picowatt per sq m). Thus

$$L_I = 10 \log_{10} \frac{I}{10^{-16}} \text{ dbi} \quad (6)$$

where

I = watts per square centimeter

or

$$L_I = 10 \log_{10} \frac{I}{10^{-12}} \text{ dbi} \quad (7)$$

where

I = watts per square meter.

The unit is the *dbi*. Either formula gives the same result.

Sound Pressure Level L_p is twenty times the logarithm to the base 10 of the ratio of the sound pressure p to the reference sound pressure of 0.0002 microbar, i.e.,

$$L_p = 20 \log_{10} \frac{p}{0.0002} \text{ dbc} \quad (8)$$

where

p = sound pressure, microbars (dynes per square centimeter).

The unit is the *dbc*, where *c* stands for the C scale or flat scale on the sound level meter.

Loudness N is the intensive subjective attribute of an auditory sensation, in terms of which sounds may be ordered on a scale extending from soft to loud. The unit is the sone. The number of sones is directly proportional to the loudness. By definition, a 1000 cps pure tone with a sound pressure level of 40 dbc has a loudness of 1.0 sone.

Loudness Level (L_N) of a sound is numerically equal to the sound pressure level L_p of a simple tone of frequency 1000 cps that is judged by the listeners to be equivalent in loudness. The unit is the phon. In this text, for a 1000 cps tone, L_N in phons is assumed to be related to N in sones by the formula

$$L_N = 33 \log_{10} N + 40 \text{ phons} \quad (9)$$

This approximation is assumed valid[3] for $N > 1$.

Sound Level at a point in a sound field is the reading in decibels of a sound level meter constructed and operated in accordance with the latest edition of *American Standard Sound Level Meters for the Measurement of Noise and Other Sounds*.[4] The reading is taken with the proper weighting network switched into the circuit. The unit is dba, if the reading is taken on the A scale; dbb if taken on the B scale, and dbc if taken on the C or flat scale. The use of these scales is discussed later.

Attenuation is ten times the logarithm to the base 10 of the ratio of the sound intensities at any two points along any continuous acoustic path in the direction away from the source. The unit is the decibel. For a signal confined to a channel or duct, the intensity values are averaged over the cross-section.

Transmission Loss TL is ten times the logarithm to the base 10 of the sound energy incident on an obstruction in an acoustic path to the sound energy that is transmitted through the obstruction. The unit is the decibel.

Noise Reduction NR is the difference in decibels of the sound pressure levels at two points along an acoustic path in the direction away from the source; alternatively, it is the difference in decibels of the sound pressure levels existing at a single point before and after the addition of acoustic treatment to the path.

APPARATUS FOR MEASURING SOUND

The measurement of sound or noise is usually accomplished by means of (*a*) a sound level meter SLM consisting of a microphone, an amplifier, a variable attenuator, weighting networks, and an indicating meter which reads directly in decibels, and (*b*) an octave band analyzer OBA which is a set of filters for determining the sound pressure level of the sound being measured in each of eight octave-frequency bands. The approved sound level meter and octave band analyzer should comply with the specifications in the latest version of the *American Standard Sound Level Meters for Measurement of Noise and Other Sounds*, Z24.3–1944, and *American Standard Specifications for an Octave-Band Filter Set for the Analysis of Noise and Other Sounds*, Z24.10–1953, published by the American Standards Association.

The SLM is designed to indicate either the sound pressure level L_p or the sound level (if weighting networks are used) above the standard reference level of 0.0002 microbar. The SLM itself has three weighting networks, which are approximations to the equal loudness contours for pure tone sounds of three discrete loudness levels. The equal loudness contours give the L_p of a 1000 cps pure tone that sounds equally as loud to the average listener as the tone whose loudness level is desired.[3]

Unfortunately, standard sound level meters do not give an indication of the loudness of more complex noises. To serve as a basis for determining the loudness of more complex noises, measurements of complex sounds are taken on the flat scale (C scale) of the SLM using the OBA to determine the distribution of the sound pressure level as a function of frequency. Computations of loudness and loudness level are made from these data.

Allowable deviations in response and acceptable tolerances recognized in the standard for the sound level meter are: ±0.5 db at 80 cps to 630 cps; +2.0, −2.5 db at 25 cps; ±2.5 db at 2000 cps; ±6.0 db at 8000 cps. Calibration of the SLM and OBA should preferably be made before, during, and after each use by a calibrated loudspeaker.[3] In turn, the loudspeaker should be calibrated at the factory at frequent intervals. Instructions and precautions in the use of sound level meters are given in Reference 2.

APPROACHES TO THE GENERAL PROBLEM OF NOISE CONTROL

It is usually necessary to think of noise control problems as composed of three parts: (*1*) the source, (2) the path, (3) the receiver.

It is sometimes possible to reduce the noise at any or all of these parts. However, in construction design, the engineer is usually able to do something only with the path.

The source may be any piece of equipment or person generating audible noises. The transmission path may be a path directly through air or may be a path which includes solid structure such as walls, floors, pipes, duct walls, or air within ducts in various combinations. The receiver is usually a person who might be disturbed by the noise; however, the receiver might also be a delicate machine or manufacturing process that must be isolated from vibration.

The acoustic output of a source is ideally specified by its

total sound power level and the sound power level in each of a group of frequency bands, preferably octave frequency bands. Where the source is not confined to a duct or the like, it may also be necessary to specify the source directivity, i.e., the relative amount of sound radiating in each direction of interest. In general, the specification of the acoustic output of a source by a single number is not adequate for engineering design of noise control measures. The sound power outputs of fans and grilles as noise sources are discussed in later sections.

Acoustic losses along a path from source to receiver normally exist in any building structure. These losses are a function of frequency and can be defined adequately when expressed as losses in each of eight octave bands. Losses for common structures, and for ducts of various sizes and shapes, as well as for some packaged sound attenuation devices, are published. Some values particularly applicable to ventilating problems are given in a later section.

The response of a person as a receiver has been determined under certain conditions. Noise levels which are found acceptable to the average person under specified conditions are called *noise criteria*. Several frequently used noise criteria are summarized in a later section. The noise criteria to be used must be chosen with regard to the type of activity to be carried on in the space being considered. Where delicate machines or manufacturing processes are the receiver, special noise criteria must be determined to fit the need. Noise criteria should best be specified as sound pressure levels in octave frequency bands.

A good procedure for attacking a noise control problem in a ventilating system will be illustrated by reference to Fig. 1. The sound power level of the fan is expressed in dbe. Sound attenuation in the duct is composed of losses in the unlined and lined portions of the duct, losses at the bend, and end reflection losses. Added attenuation of the fan noise is achieved by adding additional lining to the duct or providing package attenuation units. Noise is also produced at the grille and is expressed in dbe. All losses (attenuation) are expressed in decibels. The procedure follows:

Step 1. Determine the sound power level of the grille noise either from published data, or from actual measurements, or from approximate data given in the section on Noise Generated by Grilles.

Step 2. Determine the sound pressure levels in octave bands at a specified listener's position due to grille noise alone which is radiated from the end of the duct, using the acoustical properties of the room and the position of the listener (see the following section on Determination of Room Levels).

Step 3. Determine the allowable sound pressure level in octave bands (the criterion) at the listener's position either from published standard data (see the following section on Criteria for Noise Control) or from discussions with the architect or the ultimate user of the room under design.

Step 4. Subtract the criterion levels from the sound pressure levels at the listener's position (in decibels) to determine that the levels at least meet the criterion in each octave band. If the grille noise is higher than the criterion in any octave band, a larger grille size must be used.

Note: The total noise in the room is due to both the grille noise and the fan noise. If the grille noise levels just meet the criterion, then the fan noise emitted in the room must be at least 5 db below the criterion. Alternatively, the grille noise levels and the fan noise levels in the room could each be 3 db below the criterion, because when two noises of the same intensity are combined, the total level is 3 db higher than either. As a third choice, the fan noise could just meet the criterion, and the grille noise be at least 5 db below the criterion.

Step 5. Obtain the sound power level in octave bands of the fan, either from published data, or from actual measurements, or from approximate formulas given in the following section on Noise Generated by Fans.

Step 6. Determine the existing attenuations in the path, for example, natural duct losses, losses at bends, and reflection losses at the outlet (see the following section on Sound Attenuation in Ducts).

Step 7. Determine the net source power levels by substracting values from Step 6 from Step 5.

Step 8. Determine the sound pressure levels in octave bands at a specified listener's position using as basic data the net source power level from the duct, the acoustical properties of the room, and the position of the listener as done in Step 2.

Step 9. Subtract the criterion levels from the sound pressure levels at the listener's position (in decibels) as done in Step 4 to obtain the required amount of noise reduction NR in each of the eight octave bands (refer again to note following Step 4).

Step 10. Choose an economical means of providing the required NR. The noise reduction required usually must be obtained by treating the ducts between the source and the receiver either by adding an absorbing lining or by inserting a package unit (see Fig. 1) of known attenuation as a function of

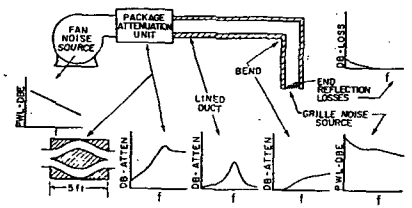


Fig. 1 . . . . Typical Ventilation System Problem

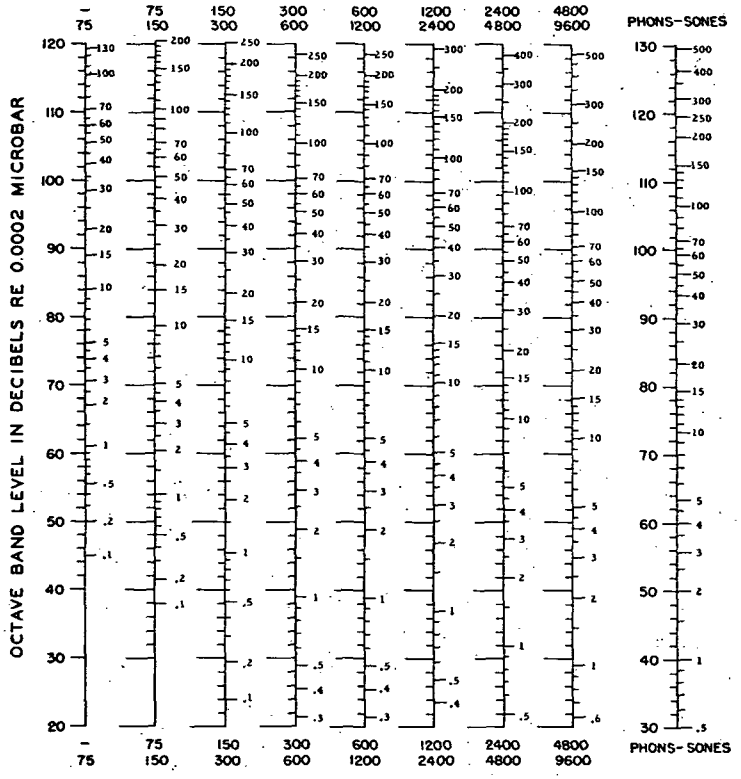
frequency. Because noise may be induced by vibration, a consideration of the fan mounting may be necessary. Also, if the fan room is adjacent to a quiet space, the construction of the fan room walls to provide adequate noise reduction should require attention. It is usually good policy to provide slightly more acoustical treatment than is necessary, thus obtaining a safety factor. A safety factor often used is 5 db greater noise reduction than would be indicated by the above procedure.

Frequently the controlling noise source in a building with a central station system is sound transmission through the walls of the apparatus room or from a duct plenum. Where such a room or plenum is near occupied rooms, the noise transmitted through the walls must be determined. The combined sound pressure levels resulting at a listener's position should be compared with the criteria as done in Step 4 for noise produced by grilles.

CRITERIA FOR NOISE CONTROL

Criteria for noise control in building spaces have evolved in the United States from several large scale studies and from the experiences of engineers who have recorded favorable and unfavorable reactions to known background noises in many types of rooms.[5]

The acoustical engineer should carefully question the user of the building to make certain of the kind of uses that he has in mind. For example, a coliseum used for sports only, need not be as quiet as a coliseum also used for concerts. As another example, a hall for band concerts need not be as quiet as one for chamber music. In fact, some rooms are better if they are not too quiet. It has been found that libraries, drafting rooms, and study rooms that are too quiet are not so desirable as they would be with moderate noise levels. In very quiet rooms of these types, sounds such as whispering, turning of pages, creaking of the floor, coughing, closing of doors, etc., act to distract the occupants and to make them complain of noise.



The calculated total loudness in sones can be converted to loudness level in phons by means of the nomogram at the right.
(From Reference 3. Used by permission.)

Fig. 2 .... Nomograms Relating L_s in Octave Bands to Loudness in Sones

Introduction of a continuous spectrum noise properly distributed as a function of frequency produces a *quieter* condition even though the background noise is actually higher. Rumbles, rattles, audible single frequency components, beats between components, and intermittency in a noise often make it objectionable even though the noise as measured seems to meet the criterion.

Noise Specification

There are at least four possible ways of specifying the noise. The first way is simply to give the octave band spectrum of the noise. This method provides the information that is essential to the design engineer and no substitute can suffice. The other three of these methods of specifying noise are single-number numerical quantities that indicate something about the influence of the noise on the average human being. These three methods are useful as summaries of the octave band spectrum, but, taken separately, they are of limited value to the design engineer. The definitions of each of four commonly used ways of specifying noise are as follows:

1. *Octave Band Spectrum*. The octave band spectrum is comprised of the readings of sound pressure level in the eight octave frequency bands (see the section on Acoustical Terminology). The readings are taken with the sound level meter weighting networks set on *flat* (C-scale) and the units are decibels (dbe). The specification of a complete octave band noise spectrum is necessary for satisfactory acoustic design in all cases.

2. *Speech Interference Level* SIL. The speech interference level is a single number found by taking the arithmetic average of the sound pressure levels in the three octave frequency bands 600–1200, 1200–2400, and 2400–4800 cps. The unit is the decibel (db). The speech interference level indicates the ease of speech communication in the presence of the noise so rated.

3. *Loudness Level* L_N. The loudness level is a single number computed from octave band levels according to the procedure following. The unit is the phon.

To obtain the loudness level in phons, obtain the sound pressure levels in the eight octave bands and read from Fig. 2 the loudness in sones for the respective bands. The sum of the loudnesses will be:

$$\Sigma N = N_1 + N_2 + N_3 + N_4 + N_5 + N_6 + N_7 + N_8 \qquad (10)$$

The loudness in *sones* of the total noise is given by

$$N = N_m + 0.3\,(\Sigma N_i - N_m) \qquad (11)$$

where

N_m = the loudness of the loudest band.

ΣN_i = the sum of the loudnesses of all eight bands.

To convert from loudness in sones to loudness level in phons, use Equation 9 or the nomogram at the right-hand side of Fig. 2.

The loudness level in phons indicates the subjective loudness but does not indicate the ease of speech communication.

4. *Sound Level*. The sound level is a single reading taken on the sound level meter with the meter operated as follows:[2, 6]

Sound-Level Range	Weighting	Unit
Below 55 db.....	A	dba
55–85 db.....	B	dbb
Above 85 db.....	C (Flat)	dbc

Sound level measured with the A-scale can be correlated most closely with speech interference level but the two are not numerically the same. The C-scale is *always* used with the octave band analyzer and is used where otherwise specified.

Criteria

Noise criteria have been developed from detailed studies and engineering experience in a number of architectural spaces.[4] The acceptability of background noise for various types of activities normally performed in rooms and enclosures can be related to two principal characteristics of the noise: (1) Speech interference level, SIL, and (2) Loudness level, L_N. Definitions of these two quantities are given in the preceding section on Noise Specification. The speech interference level is governed by the amount of high-frequency noise (between 600 and 4800 cps) and the loudness level is governed by both the high- and low-frequency noise.

For most types of spaces the loudness level in phons is 22 to 25 units higher than the speech interference level in dbc. However, for factories, sports coliseums, restaurants, and similar spaces where cost of acoustic treatment dictates a maximum amount of compromise, the loudness level may rise as much as 30 units above the speech interference level.

Acceptable background noise criteria have been established for these two conditions and are presented in Figs. 3 and 4. Each curve is designated by a number indicating the speech interference level obtained. The numbers in the first column of Table 1 and Table 2 identify the curve which defines recommended noise levels for the various types of spaces listed.

With regard to the choice between Figs. 3 and 4, Fig. 3 should be used whenever possible, particularly where there is a possibility that the sound may be fluctuating, such as often occurs with turbulence in a ventilating system. Fig. 4, or numbers in the range between Figs. 3 and 4, may be used when economic factors dictate a compromise and when there is a definite assurance that the noise will not be of a fluctuating type.

In engineering design, all eight octave band levels selected from the curves of Fig. 3 or 4 should be used. In construction specifications, reference may be made to THE GUIDE and the selected *NC* or *NCA* curve may be referred to by number, e.g., *NC*-35.

Where octave band analyses of the noise cannot be made and a rough approximation of accepted criteria must be obtained by use of a sound level meter only, the A-scale values which would be read for the noise spectra specified in Column 1 are given in Column 2 for comparison purposes. It should be understood, however, that an A-scale reading gives a measure of the overall noise using a weighted network; it does not specify the spectrum. For example, the noise spectrum shown as *NC*-25 in Fig. 3 would give an overall A-scale measurement of 35 dba, but a measured value of 35 dba does not mean that the actual noise measured has the same spectral shape or is as satisfactory as the specified *NC*-25 spectrum. The lack of octave band analysis can be partly offset by a subjective observation; if the noise sounds unnatural because of excessive amounts of high pitched or low pitched noise or has particularly noticeable amounts of pure tones, the spectrum will probably depart widely from the specified one. In such cases it is advisable to make an octave band analysis of the noise, if possible.

KINDS OF NOISE

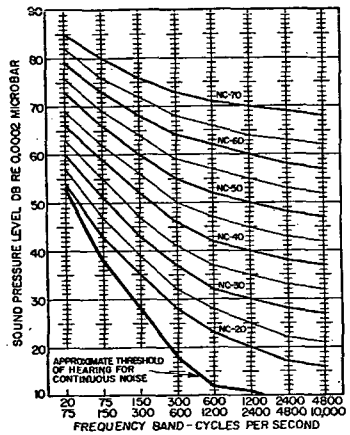
The noise introduced into a room or building by ventilating or air-conditioning equipment may be divided into two general kinds, depending on how it reaches the room:

1. *Noise transmitted through the ducts.*
   - *a.* From equipment such as fans, motors, sprays, etc.
   - *b.* From outside, and transmitted through duct walls into air stream.
   - *c.* From duct wall vibrations, transmitted into air stream.
   - *d.* From air currents, including eddying noises.
   - *e.* Cross talk and cross noises between rooms connected by the same duct system.
   - *f.* From dampers and air valves located near branch duct take-offs and outlets.
   - *g.* Noise produced by grilles or diffusers.
2. *Noise transmitted through the building construction.*
   - *a.* From machine mountings as vibration.
   - *b.* From equipment through room wall surfaces.

The following text will present data and discussion of methods whereby solutions of the noise problem can be obtained when the allowable room noise level, and the path through which the noise reaches the room, are known.

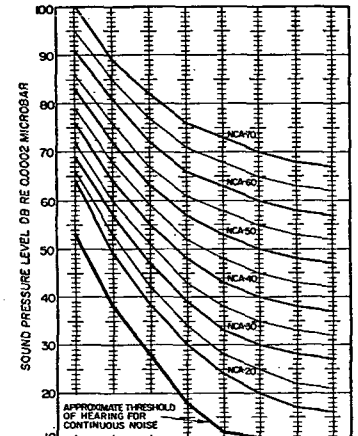
NOISE GENERATED BY FANS

Aerodynamic noise from all types of fans may be broadly divided into a rotational component and a vortex component. The rotational component is associated with the impulse given to the air each time a blade passes a given point and is hence a series of discrete tones at the fundamental blade passing frequency and harmonics thereof. The vortex component of noise is largely due to the shedding of vortexes from the fan



Each NC curve has a loudness level in phons that is 22 units greater than the Speech Interference Level in decibels (expressed by the NC number of the curve). These curves should be used in design or specifications wherever a favorable relation between the low frequency and the high frequency portion of the spectrum is desired (see text).

Fig. 3 .... Criterion Curves for Use with Table 1 in Determining the Permissible (or Desirable) L_p Values in 8 Octave Frequency Bands



Same as Fig. 3 except that each NCA curve has a loudness level in phons that is 30 units greater than the Speech Interference Level in decibels (expressed by the NCA number of the curve). These curves may be used in design or specifications in place of those of Fig. 3 wherever economy dictates a maximum compromise and where, in addition, the fan and turbulence noise is steady and free of low frequency pure tone components (see text).

Fig. 4 .... Criterion Curves for Use with Table 1 in Determining the Permissible (or Desirable) L_p Values in 8 Octave Frequency Bands

blades. It is random in character and has a continuous spectrum over a wide range of frequencies determined by the fan geometry and operation. It may also have many nonharmonic single-frequency components determined by blade geometry and local air velocity. Since the laws of generation of these two types of noise are different, they will vary in importance for different types of fans and operating conditions.[6, 7]

In addition to aerodynamic noise, there are usually several non-aerodynamic sources of noise in equipment involving fans. Such sources include noise resulting from unbalance, bearing noise, brush noise, magnetic noise, and belt noise.

The number of blades of a centrifugal fan generally is governed by optimum air-flow design. The noise generation decreases but slightly for more than the optimum number of blades.

Powerful tones of definite pitch may be generated in fans when the wakes from the impeller blades impinge on an obstruction such as a badly located or shaped cut-off in a centrifugal fan, or struts or motor supports which are badly shaped and too close to the impeller of an axial flow or propeller fan. This interference noise has the same frequency characteristics as rotation noise.[8]

A shroud around a propeller fan may serve to reduce noise considerably if it is working properly. Such reduction is generally most effective at the higher harmonics. However, if the flow breaks down over part of the shroud, the noise may become considerably worse than for an unshrouded case.

As the operating pressure across axial fans is increased the maximum sound intensity is shifted from the fundamental to higher harmonics. This effect is not observed for centrifugal fans.

Effect of Fan Parameters on Noise vs. Performance

Fan Laws. The principal laws governing the mechanical performance of a fan, whether centrifugal or axial, are fairly well established and are given in Chapter 22.

Fixed Point of Rating. A fixed point of rating of a given size fan at a given speed is one for which the capacity (volume flow) is chosen as a given fraction of the free delivery capacity. Having chosen this capacity ratio, the ratio of pressure to the static no-delivery pressure is determined. This fixed point of rating will have a fixed efficiency within the limitation of certain manufacturing details. The fan laws serve to relate the performance of two members of a symmetrical series of fans operating at the same fixed point of rating whether at the same or different speeds.

Correlation of Fan Noise with Size, Static Pressure, Speed, and Capacity. The empirical relations between the noise generated by a fan and its size, static pressure, speed, and capacity are as follows:

Change in overall sound power level varies as:

$$70 \log_{10}(\text{size}_2/\text{size}_1) + 50 \log_{10}(\text{speed}_2/\text{speed}_1) \quad (12)$$

$$20 \log_{10}(\text{size}_2/\text{size}_1) + 25 \log_{10}(\text{pressure}_2/\text{pressure}_1) \quad (13)$$

$$10 \log_{10}(\text{capacity}_2/\text{capacity}_1) + 20 \log_{10}(\text{pressure}_2/\text{pressure}_1) \quad (14)$$

These relations also apply only for a fixed point of rating.

A double width fan is essentially two fans of the same size, speed, and sound pressure level and therefore its sound power level will be $10 \log_{10} 2$ or 3 db greater than for the single one.

Example 1: Consider a 36½-in. diameter fan operating at 700 rpm. If it produces a noise level of 70 dbc what will be the level if a 49-in. diameter fan operating at 1000 rpm is substituted for it?

Solution: From Equation 12,

$$\text{Change in level} = 70 \log_{10} \frac{49}{36.5} + 50 \log_{10} \frac{1000}{700} = 16.8 \text{ db}$$

$$\text{Noise level of second fan} = 70 + 16.8 = 86.8 \text{ dbc}$$

Fan noise does not follow equations exactly. The radiated sound power of a centrifugal fan, for example, is not all concentrated on the blade frequency. While the overall sound power level seems to follow these equations fairly closely, the various components in the different frequency bands may show greater deviation. In the case of axial flow fans there is a wider range in characteristics of the vortex noise, so it would be expected that a slightly greater deviation from these equations exists than for centrifugal fans. This is particularly true as speed is increased. The fundamental or blade frequency exists over a wider range of pressure in the case of the centrifugal fan. In the axial fan, the harmonics are more easily excited and often the second and third harmonics exceed the level of the fundamental.[8]

Specific Plots

Curves which describe the characteristics of an entire series of similar fans can be plotted by using dimensionless coordinates.[7] To describe the pressure characteristics the pressure coefficient ψ can be plotted against the flow coefficient ϕ as shown in Fig. 5, where

$$\psi_t = \frac{p_t}{\left[\frac{(\pi)(D)(\text{rpm})}{4005}\right]} = \frac{\text{total pressure}}{\left(\begin{matrix}\text{wheel peripheral} \\ \text{velocity pressure}\end{matrix}\right)} \quad (15)$$

Table 1 .... Recommended Noise Criteria for Rooms[a]
Noise levels to be Measured in Unoccupied Rooms

Type of Space	Recommended Noise Criterion Curve	Computed Equivalent Sound Level Meter Readings[a] Weighting Scale-A dba
Broadcast studios	NC 15–20	25–30
Concert halls	NC 20	30
Legitimate theaters (500 seats, no amplification)	NC 20–25	30–35
Music rooms	NC 25	35
Schoolrooms (no amplification)	NC 25	35
Conference rooms for 50	NC 25	35
Apartments and hotels	NC 25–30	35–40
Assembly halls (amplification)	NC 25–30	35–40
Homes (sleeping areas)	NC 25–35	35–45[b]
Conference rooms for 20	NC 30	40
Motion picture theaters	NC 30	40
Hospitals	NC 30	40
Churches	NC 30	40
Courtrooms	NC 30	40
Libraries	NC 30	40
Small private offices	NC 30–35	40–45
Restaurants	NCA 45	55
Coliseums for sports only (amplification)	NCA 50	60
Stenographic offices (typing and business machines)	NC 50	60
Factories	NCA 40–65	50–75

Note: Each noise criteria curve is a code for specifying permissible sound pressure levels in eight octave bands. It is intended that in no one frequency band should the specified level be exceeded. The computed equivalent dba numbers in the right-hand column are presented for information only and are no longer recommended for use in specifications. Ventilating systems should be operating and outside noise sources, traffic conditions, etc., should be normal when measurements are made.

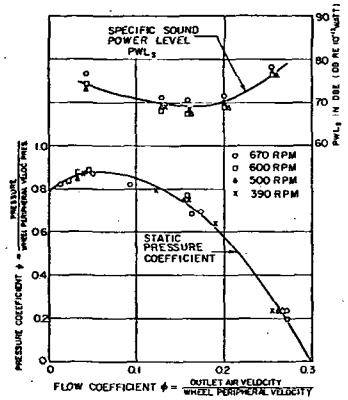
[a] If the noise is predominantly in the speech bands so that the spectrum does not have the shape of the recommended noise criterion curve, the recommended dba numbers should be lower by 5 dba.

[b] Room air conditioners made prior to 1957 commonly produce levels of 50 to 60 dba in sleeping areas.

Table 2 .... Recommended Noise Criteria for Offices[a]

NC Curve of Fig. 3	Communication Environment	Typical Applications
NC20 to NC30	Very quiet office—telephone use satisfactory—suitable for large conferences.	Executive offices and conference rooms for 50 people.
NC30 to NC35	Quiet office; satisfactory for conference at a 15-ft table; normal voice 10 to 30 ft; telephone use satisfactory.	Private or semiprivate offices, reception rooms, and small conference rooms for 20 people.
NC35 to NC40	Satisfactory for conferences at a 6–8 ft table; telephone use satisfactory; normal voice 6 to 12 ft.	Medium-sized offices and industrial business offices.
NC40 to NC50	Satisfactory for conferences at a 4–5 ft table; telephone use occasionally slightly difficult; normal voice 3 to 6 ft; raised voice 6 to 12 ft.	Large engineering and drafting rooms, etc.
NC50 to NC55	Unsatisfactory for conferences of more than two or three people; telephone use slightly difficult; normal voice 1 to 2 ft; raised voice 3 to 6 ft.	Secretarial areas (typing), accounting areas (business machines), blueprint rooms, etc.
Above NC55	Very noisy; office environment unsatisfactory; telephone use difficult.	Not recommended for any type of office.

[a] Noise measurements made for the purpose of rating the noise level in an office by comparison with these criteria should be performed with the office in normal operation, but with no one talking at the particular desk or conference table where speech communication is desired (i.e., where the measurement is being made). Background noise with the office unoccupied should be lower by approximately 5 to 10 units.



These data are for a particular fan. The quantities ψ_s, ϕ, and PWL$_s$ are discussed in the text. Such curves apply to all fans in a similar series.

Fig. 5 . . . . Specific Fan Characteristic Curves, Including Specific Sound Power Level

$$\psi_s = \frac{p_s}{\left[\frac{(\pi)(D)(\text{rpm})}{4005}\right]} = \frac{\text{static pressure}}{\left(\begin{matrix}\text{wheel peripheral}\\ \text{velocity pressure}\end{matrix}\right)} \quad (16)$$

$$\phi = \frac{(\text{cfm})/(\text{Outlet Area})}{(\pi)/(D)(\text{rpm})} = \frac{\text{outlet velocity}}{\text{wheel peripheral velocity}} \quad (17)$$

where

D = wheel diameter, feet.
p_s = static pressure.
p_t = total pressure.

Such curves apply to all fans in a similar series, i.e., a series of fans differing in size only while preserving complete similarity in shape by having all linear dimensions changed in the same proportions.

An overall specific sound power level PWL$_s$ in dbe as shown in Fig. 5 can also be defined that will be applicable to all fans in a family.[7] It is obtained by means of Equation 18 which follows:

$$\text{PWL}_s = \text{PWL} - 10\log_{10}[q \cdot p^2] \text{ dbe} \quad (18)$$

where

q = quantity of air discharged, cfm.
p = pressure, inches of water.
PWL = sound power level actually measured.

Since static pressure p_s is more frequently measured for fans than total pressure p_t, p_s may be used under most normal operating conditions (when the difference between p_s and p_t is less than 20 percent) and the error will then be less than 1 db. Near free discharge, p_s becomes zero and p_t must be used. In other words, PWL$_s$ is the sound power level that a similar (although hypothetical) fan would produce when operating at 1 cfm and a total pressure of 1 in. of water. As an example, the specific power level for one fan tested is shown on Fig. 5.

Although few data are available to confirm the conclusion, it is indicated that such a plot also holds throughout a series of fans for all eight of the frequency bands provided rotor or blade passage frequencies are not predominant.

Approximate Overall Sound Power Level

In practice, data of the type shown in Fig. 5 are usually not available. Hence, in design an estimate has to be made of the overall sound power level and of the shape of the noise spectrum. Tests on centrifugal fans ranging in size from 0.01 hp to 100 hp show that the sound power level in a duct *either* upstream or downstream from the fan can be found within ±4 db from the simple relation:[9, 10]

$$\text{PWL} = 100 + 10\log_{10}W_E + 10\log_{10}p \text{ dbe} \quad (19)$$

where

W_E = nameplate electrical horsepower of the motor driving fan (assuming that the motor is operating near rated hp).
p = pressure head, inches of water.
dbe = number of decibels re 10^{-13} watt.

(See discussion following Equation 18.)

The assumption is made that the fan is being operated near the *knee* on its operating characteristics, i.e., near maximum efficiency or near the minimum in the curve of specific sound power (upper curve on Fig. 5).

By assuming a fan mechanical efficiency of 50 percent, Equation 19 can be put in the form of Equation 18 with PWL$_s$ = 65 dbe, i.e.,

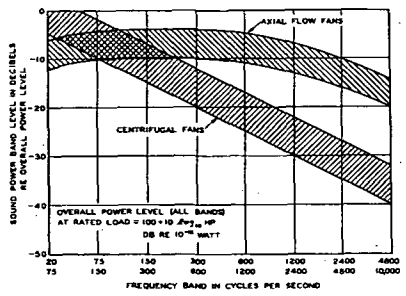
$$\text{PWL} = 65 + 10\log_{10}q + 20\log_{10}p \text{ dbe} \quad (20)$$

where

q = air quantity, cfm.
p = pressure head, inches of water.

(See discussion after Equation 18.)

It should be emphasized that Equations 19 and 20 are intended to be used only when actual noise data are not available. All fans of the same power input are not equally noisy. The example illustrated in Fig. 5 demonstrates this fact since the minimum value of PWL$_s$ for *this particular fan* is 69 db, 4 db higher than the value used in Equation 20. It is on the edge of the stated limit of approximation ±4 db.



The overall power level is first determined from Equation 19 or 20 and is added algebraically to the ordinate in Fig. 6. Acoustics by L. L. Beranek (McGraw-Hill, 1954 Second Printing). Used by permission.

Fig. 6 . . . . Chart for Determining the Sound Power Band Levels for Ventilating Fans of Two Types

Approximate Octave Band Spectrum

Within about ±4 db approximation, and assuming no turbulence in the duct or the coupling between it and the fan, the octave band spectrum for a centrifugal fan, relative to the overall sound power level, is given in Fig. 6.

Limited data on vane-axial fans,[11] where sound pressure level measurements were made in the discharge duct, indicate that the overall sound power level is about the same as that for centrifugal fans. However, the spectrum shape of the vane-axial fan noise has a shape something like that shown in Fig. 6.

More recent free field measurements on axial fans indicate that perhaps the drop in level at the high and low ends of the spectrum are more pronounced.[8] The solid line in Fig. 7 is typical of the sound power spectrum for the axial fans reported, as taken directly from the free field measurements. Application of a correction for the end reflection losses from the discharge opening (see Fig. 11) gives the dashed curve in Fig. 7, which corresponds to the relative sound power level in the duct. This spectrum shape for the axial fan appears to be based on more extensive data than those of Fig. 6.

NOISE GENERATED BY GRILLES AND DIFFUSERS

Grille Noise

The subject of grille noise has not been covered thoroughly in the literature. The information given in this chapter has been taken *from the advertising literature of several grille* manufacturers and from recent experiments at the Massachusetts Institute of Technology.

In general, grille noise at the lower frequencies is dependent primarily on the size and shape of the grille opening, on the mass flow per minute, and on the temperature of the air. At the higher frequencies, the noise is determined primarily by the area of the grille, the velocity of air through the free (open) areas (not the core or nominal area), and the pressure drop across the grille.

For two simple types of grille, stamped and deflector vane, power levels are given in Fig. 8. These power levels are given in terms of the speech interference level, PWL$_{SIL}$. Conversion to the speech interference level in db in the room at a listener's position is described in the section on Determination of Room Levels. In the absence of more complete information, it may be assumed that the power levels of the noise produced by the grilles themselves at low frequencies are inconsequential.

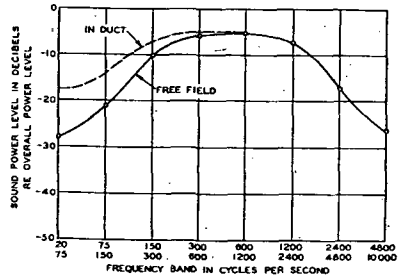
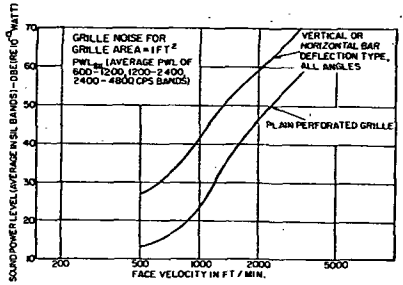


Fig. 7 . . . . Sound Power Spectrum for Single-stage Axial Fan[8]



The noise is expressed as the sound power level in dbe in each of the three speech interference bands (PWL$_{SIL}$).

Fig. 8 . . . . Grille Noise for a Grille Area of One Square Foot

Fig. 8 is for a grille core area of 1 sq ft. Correction to other areas of grille is made by the relation:

$$\text{Decibel addition} = 10\log_{10}A \text{ db}$$

where

A = core area, square feet.

Values of the decibel addition for various values of A are given in Table 3.

Table 3 . . . . Decibel Correction for Area

A sq ft	Decibel Addition	A sq ft	Decibel Addition	A sq ft	Decibel Addition
0.5	−3	8	9	70	18
1	0	16	12	100	20
2	3	20	13	200	23
4	6	40	16	400	26

Example 2: Assume a deflector vane type of grille with a core area of 2 sq ft. Determine the sound power level in the speech interference bands for an air velocity of 2000 fpm.

Solution: From Fig. 8, the PWL$_{SIL}$ for a 1 sq ft deflection grille is 59 dbe. From Table 2, an addition of 3 more dbe is required, so that PWL$_{SIL}$ = 62 dbe.

Diffuser Noise

Aside from the information furnished by diffuser manufacturers, to their users, on sound levels generated by air diffusers, few other data are available. Recently, however, some test results on several types of ceiling air diffusers have yielded a satisfactory correlation of the sound generating characteristics of these devices.[12] The data, covering ceiling diffusers ranging in size from 4- to 18-in. neck diameter, show that the

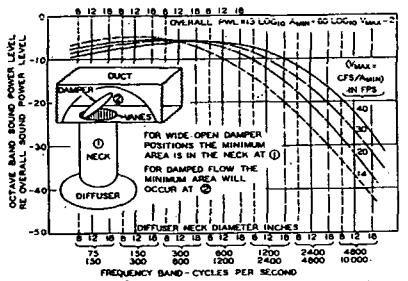


Fig. 9 .... Octave Band Spectrum of Ceiling Diffusers

overall sound power level (from 75 to 10,000 cps) can be found within ±2 db by the equation:

$$PWL_7 = 13 \log_{10} A_{min} + 60 \log_{10} V_{max} - 2 \text{ dbe} \qquad (21)$$

where

PWL_7 = overall sound power level in the seven octave bands from 75 to 10,000 cps, decibels *re* 10^{-13} watt.

A_{min} = minimum flow area for the air passing through the diffuser, square feet. For undamped flow the minimum area will normally occur in the diffuser neck, whereas for damped flow it will occur at the damper position (see Fig. 9).

V_{max} = *maximum air velocity* (q/A_{min}), *feet per second.*

q = quantity of air discharged, cubic feet per second.

For ceiling diffusers of cone-type construction used in conjunction with deflecting vanes and a damper, Fig. 9 presents the approximate octave band spectrum relative to the overall sound power level as given by Equation 21. The curves of Fig. 9 agree within approximately 3 db of the octave band data collected on 8-, 12-, and 18-in. neck diameter ceiling diffusers.

NOISE GENERATED BY OTHER SOURCES

Objectionable noise may be produced by fan motors, pumps, compressors, etc. The designer must give attention to these sources of noise. It is not possible to predict the noise, because units from different manufacturers differ greatly in the noise produced.

Objectionable noise is also generated in the duct system by excessive air velocity or poor duct fittings. No systematically taken data have been published from which conclusions can be drawn. The reader is referred to Chapter 21 Air Duct Design for good design practice.

Table 4 .... Approximate Attenuation in Straight Sheet-Metal Duct Runs in Decibels Per Foot

Duct	Size, Inches	All Frequencies
Small	6 x 6	0.1
Medium	24 x 24	0.05
Large	72 x 72	0.01

SOUND ATTENUATION IN DUCTS

Unlined Straight Sheet-Metal Ducts

The attenuation of sound in straight sheet-metal ducts is a function of the length, shape, and size of the duct. If a thermal insulating material is applied to the outside of the duct this will also affect the attenuation characteristics.[13] Attenuation values in common use are given in Table 4.[14]

A recent investigation conducted at the Massachusetts Institute of Technology indicates that the attenuation values given in Table 4 are probably satisfactory above a frequency of 500 cps, but are in most cases significantly lower than the observed values of attenuation below this frequency.[13] Only two sizes of ducts were tested, a 12 x 12 in. and a 12 x 24 in. However, these ducts were investigated not only bare, but also with two types of glass fiber thermal insulation applied to the outside of the duct. The first type used was a 1-in. thick glass fiber insulation with a density of 0.75 lb per cu ft. The second type was a glass fiber semirigid 1-in. board with a density of 6 lb per cu ft. Attenuation values were determined by plotting SPL readings made at 3-ft intervals over a total duct length of 40 ft. The results are presented in Table 5.

It should be noted from Table 5 that the addition of thermal insulation to the outer duct walls had little or no effect on the attenuation in the four octave bands above 600 cps for the ducts tested, and that the values are substantially in agreement with Table 4. In the lower octave bands, however, the effect of added thermal insulation on attenuation is indeed striking. More data on larger size ductwork is needed to confirm these results.

Branches

At a junction where a duct divides or has side branches or outlets, the sound power traveling down the duct divides.

The decibel reduction in sound power from the main duct

Table 5 .... Octave Band Attenuation of Bare and Thermally Insulated Straight Sheet-Metal Ducts

Duct Size Inches	Type of Insulation	Density	Attenuation—db per ft							
			Frequency Band—Cycles per Second							
			20–75	75–100	150–300	300–600	600–1200	1200–2400	2400–4800	4800–10,000
12 x 12	1-in. Glass Fiber, blanket type	—	0.26	0.26	0.20	0.08	0.08	0.12	0.12	0.16
		0.75	0.40	0.40	0.32	0.16	0.08	0.24	0.12	0.16
		6.0	0.66	0.66	0.50	0.26	0.08	0.24	0.12	0.16
12 x 24	1-in. Glass Fiber, semirigid board type	—	0.30	0.36	0.16	0.06	0.12	0.12	0.12	0.16
		0.75	0.38	0.44	0.22	0.10	0.12	0.12	0.12	0.16
		6.0	1.04	0.56	0.28	0.16	0.12	0.12	0.12	0.16

Table 6 .... Power Level Attenuation at Duct Branches or Outlets

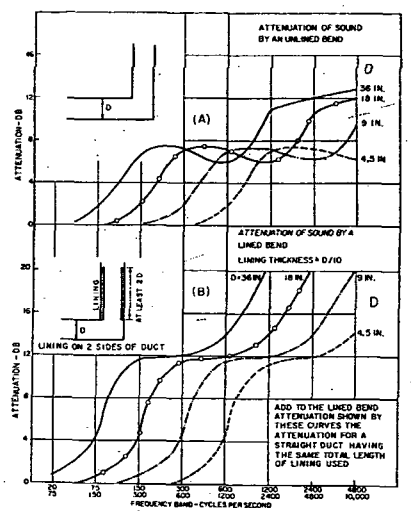
Ratio Branch Duct Area / Sum of Branch Areas	Attenuation db	Ratio Branch Duct Area / Sum of Branch Areas	Attenuation db
1.0	0	0.25	6
0.8	1	0.2	7
0.63	2	0.16	8
0.5	3	0.13	9
0.4	4	0.1	10
0.32	5		

to any branch may be calculated as 10 log₁₀ of the ratio of the branch area to the total area of all branches leaving the junction. Generally the total area of the branches leaving a junction is greater than that of the duct entering a junction; therefore, the area of the branch should not be divided by the area of the supply duct. In a complicated branched system, errors incurred by such a practice may accumulate to several db. Table 6 gives the db reduction for various ratios of branch duct area to the total branch duct area.

Square Corner Bends

Square corner bends can be quite effective in noise reduction provided they are widely separated in the system. The noise reduction for a single square corner bend with no turning vanes, unlined, is shown in Fig. 10 Part A. Turning vanes are usually employed in square corner ducts. When the width of the turning vanes is small, say less than one-fourth wavelength of the sound, they do not affect the noise reduction. At higher frequencies, they decrease the noise reduction substantially below that shown in Fig. 10 Part A.

When a square corner bend is lined with sound absorbing material, the lining should be extended several duct widths beyond the corner in the direction of sound transmission. For square bends lined in this manner the attenuation may be estimated from Fig. 10 Part B.



The attenuation is given for eight octave frequency bands with D in inches as the parameter.

Fig. 10 .... Attenuation in Decibels Provided by (A) Unlined Bend, and (B) Lined Bend in a Duct with the Lateral Dimension D

End Reflection Losses

All of the noise power that is produced in a duct by a fan (or by turbulence) does not radiate from the end of a duct into the room. Instead, part of the power is reflected back from the end toward the fan. The amount of noise power reflected is related to the product of the frequency of the sound and the length of a duct side L, assuming that the duct is square. If the duct is not square, the value of L is taken to be approximately equal to $\sqrt{L_x L_y}$. The end reflection loss is shown in Fig. 11 for two cases: (1) a duct ending in free space and (2) a duct ending flush with the wall.

Duct Lining Procedure

In the past, the most common method of obtaining sound absorption in ventilating systems has been to line the duct with absorbing material. It is usually more convenient to line all four sides of the duct, but a lining on one side over a longer length of the duct will, in general, give the same effect for the same area of applied acoustical material. Subject to certain restrictions, the attenuation of a fully lined duct to single-frequency sounds may be expressed by the approximate Equation 22.[15]

$$R = 12.6l \frac{P}{A} \alpha^{1.4} \text{ db} \qquad (22)$$

where

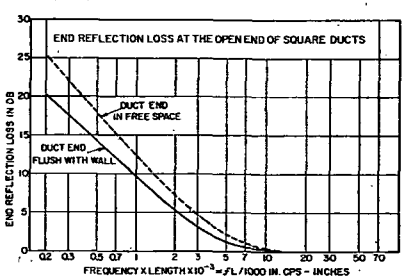
R = attenuation, decibels.

l = length of lined duct, feet.

P = perimeter of duct, inches.

A = cross-sectional area of duct, square inches.

α = absorption coefficient of lining (a function of frequency).[16]



The abscissa equals fL/1000 where f is the geometric mean frequency of an octave band in cps and L is the duct dimension in inches. Two cases are shown, with the duct end flush in the wall and with the duct end projecting into the room.

Fig. 11 .... End Reflection Losses in Decibels at the Open End of a Square Duct of Area L^2

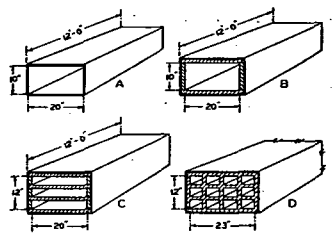
Table 7 .... Attenuation Data for Typical 1-in. and ½-in. Thick Duct Lining Board

Frequency Cycles Per Second	1-Inch Thickness			½-Inch Thickness		
	Absorption Coefficient α	$\alpha^{1.4}$	Attenuation db	Absorption Coefficient α	$\alpha^{1.4}$	Attenuation db
128	0.12	0.051	$0.64\, l \frac{P}{A}$	0.09	0.034	$0.43\, l \frac{P}{A}$
256	0.38	0.26	$3.2\, l \frac{P}{A}$	0.25	0.15	$1.9\, l \frac{P}{A}$
512	0.70	0.60	$7.6\, l \frac{P}{A}$	0.40	0.28	$3.5\, l \frac{P}{A}$
1024	0.80	0.73	$9.2\, l \frac{P}{A}$	0.72	0.63	$7.9\, l \frac{P}{A}$
2048	0.79	0.72	$9.1\, l \frac{P}{A}$	0.78	0.71	$8.9\, l \frac{P}{A}$

This formula was empirically developed for a set of duct sizes ranging from 9 x 9 in. to 18 x 18 in.; for cross-sectional dimension ratios of 1:1 to 2:1; for frequencies between 256 and 2048 cycles; and for absorption coefficients between 0.20 and 0.80. The duct lining material used was 1-in. rock wool sheet. The absorption coefficients of a material of this type in one-half and one-inch thickness are listed in Table 7.

It is also possible to calculate the absorption by a very complicated mathematical theory.[17, 18] Such calculations are in substantial agreement with Equation 22. This equation may be in error when applied to other types of duct lining and to duct sizes and shapes greater than those specified.

With every application, the use of sound absorptive material should be considered in the dual function of insulation and sound absorption. It has been shown theoretically that the reduction (in decibels per linear foot) of sound transmitted along a duct lined with sound absorbing material is related in a rather complicated manner to the size and shape of the duct, to the frequency of the sound, and to the sound absorbing characteristics of the lining. Experimental evidence likewise indicates that there is no simple formula involving the variables which will apply accurately to all cases. However, it may be stated generally that the attenuation in decibels at a given frequency is directly proportional to the length of lined duct. It decreases as the cross-sectional area increases, and increases as the aspect ratio is increased.



A. Unlined metal duct
B. Lined duct
C. Splitter plate type absorber
D. Cell-type absorber

The open area is the same for each case shown if 1 in. thick absorption material is assumed.

Fig. 12 .... Acoustic Treatment of Ducts

Rectangular Cells (Plate or Cell Absorbers)

If the available length of duct between the main duct (or the fan) to the grille is shorter than the length of lining indicated as necessary by Equation 22, the duct may be subdivided into smaller ducts as shown in Fig. 12, or it can also be even more subdivided by an egg-crate construction. In such a construction in which all the subdivided ducts are the same size, sound will be equally absorbed down each channel. It is, therefore, only necessary to calculate the sound attenuation of an individual channel. For this, Equation 22 is adequate, assuming that the thickness of the material is halved if it forms a common splitter between two cells.

When the number of splitter plates or cell partitions is large, the percentage free area of the gross duct size may be materially reduced. This leads to a further sound attenuation. Values of the attenuation possible, due to this cause, are given in Table 8.

Table 8 .... End Reflection of Plate or Cell Absorbers

Percentage Free Area of Absorber	50	40	30	25	20
Attenuation db	1	2	4	5	6

Plenum Absorption

In systems where individual ducts are directed to a number of rooms, and sound treatment is required in every duct, a sound absorption plenum on the fan discharge as shown in Fig. 13 will often prove the most economical arrangement.

Based on both experiment and ray acoustics theory, the following approximate expression has been derived for acoustic plenum attenuation.[19]

$$\text{db (attenuation)} = 10 \log_{10} \frac{1}{\frac{S_e \cos\theta}{2\pi d^2} + \frac{1-\alpha}{\alpha S_w}} \qquad (23)$$

where

α = absorption coefficient of the lining (dimensionless).
S_e = plenum exit area, square feet.
S_w = plenum wall area, square feet.
d = distance between entrance and exit, feet.
θ = the angle of incidence at the exit, i.e., the angle which the direction d makes with the normal to the exit opening.

For frequencies sufficiently high so that the dimensions of the plenum exceed about one wavelength, Equation 23 will predict the sound attenuation within a few decibels. At low frequencies, somewhat more attenuation is realized than will be computed by Equation 23, due to the existence of a resonant muffler action in the duct-plenum system.

Outlet Sound Absorbers

Outlet sound absorbers are rectangular or plate cells installed directly behind an outlet or they may be the lining of a pan or plaque outlet. They are particularly effective in the elimination of high frequency whistles which are generated by air flow in the ducts. They are also employed in large systems with long runs where only a few outlets near the fan require treatment. Frequently outlet cells are the only means of correcting existing noisy installations, as the duct sections directly behind the outlets may be the only sections accessible for treatment. (See Fig. 14.)

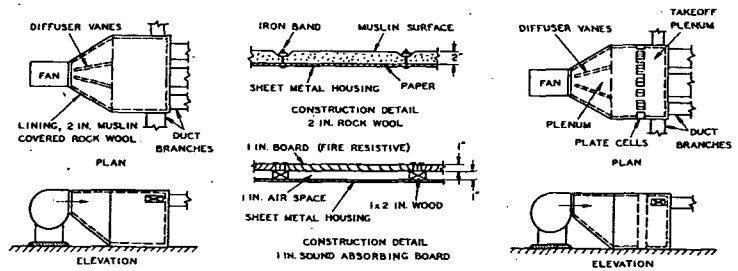


Fig. 13 .... Absorption Plenums With and Without Sound Cells

Package Units

Package units for attenuating sound in ducts are coming more and more into common use. These units have the advantage of providing, in a short length, a known amount of noise reduction with a minimum opportunity for errors in installation and for erosion of materials. The cost of inspection of the completed job is eliminated and the outside dimensions of the duct can be reduced as compared to a lined duct.

These units are available in sizes to fit 6 x 12 in. ducts up to 80 x 84 in. ducts. Charts are supplied by the manufacturers giving the attenuations and pressure drops for different types of units.

In Figs. 15 and 16, the noise reductions and static pressure drops for several types of package units are shown. Manufacturers' literature should be consulted for exact values.

Selection of the Absorptive Material

When a sound wave impinges on the surface of a porous material, a vibrating motion is set up within the small pores of the material by the alternating sound waves. As the ratio of the cross-sectional area of the pores to their interior surface is small, the resistance to the movement of air in the pores is large. This viscous resistance within the pores of the material converts a portion of the sound energy into heat. The decimal fraction representing the absorbed portion of the incident sound wave is called the absorption coefficient. Considerable absorption may also result, particularly in the low-frequency range, from the flexural vibrations of the duct. In the selection and application of the absorptive material, the following points should be considered:

1. For the absorption of the low-frequencies below 500 cps the material should be 2- to 12-in. thick. Thin materials, particularly when mounted on hard solid surfaces, will absorb only the high frequencies.

2. In order to provide as much low-frequency noise absorption as possible by means of flexural vibration, it is desirable to fasten the absorptive panels discontinuously. This result may be attained to some extent by spot cementing, but better results are obtained when it is possible to fasten the absorptive panels to furring strips, leaving an air space behind. However, the exact resonance characteristics of the panels, and thus their absorption, are so unpredictable that flexural vibration cannot be relied upon for a specific value of attenuation.

Requirements for a good sound absorption material are: (1) high absorption at low frequencies;[16] (2) adequate strength to avoid breakage; (3) fire resistance and compliance with national and local code requirements; (4) low moisture absorption; (5) freedom from attack by bacteria and algae; (6) low surface coefficient of friction; (7) particles should not fray off at the higher design velocities; and (8) freedom from odor when either dry or wet.

DETERMINATION OF ROOM LEVELS

In the previous sections on fan and grille noise, sound power levels were given in dbe referred to 10^{-13} watt. These quantities as they appear at the duct termination (grille) must be

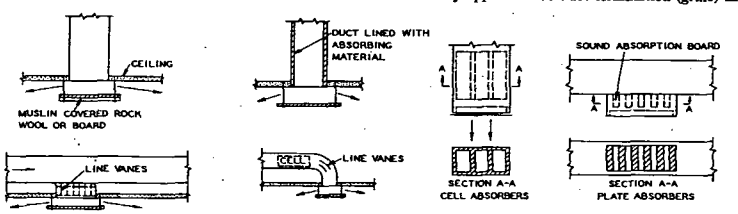


Fig. 14 .... Outlet Cells for Pan Outlets or Grilles

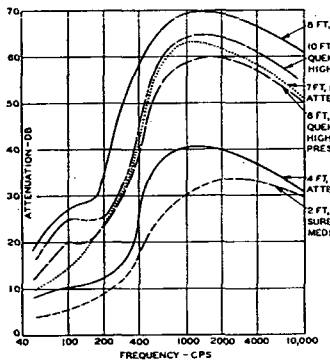
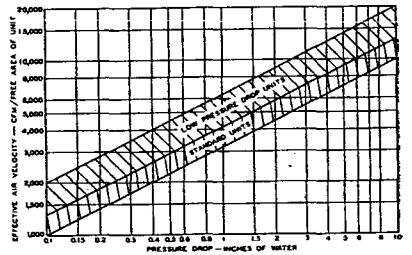


Fig. 15 .... Typical Values of Sound Attenuation Through Several Types of Package Units



Note: Early models of these units sometimes gave higher pressure drops than those shown above. Data of certified test laboratories on current unit should be consulted where possible.

Fig. 16 .... Approximate Pressure Drop Across Typical Package Units

converted to sound pressure levels in the room, because it is sound pressure to which the ear and the sound level meter respond. The sound pressure level at a distance r from the grille opening and at an angel θ with respect to an axis perpendicular to its surface is given by the equation[20]

$$L_p = \text{PWL} + 10 \log_{10}\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right) + 0.5 \text{ dbc} \qquad (24)$$

where

PWL = sound power level emitted at the duct termination and is the power summation of the attenuated fan noise and the grille noise in dbe (see *Example 3*).
Q = directivity factor and is a dimensionless function of θ.
r = distance from the duct opening, feet.
R = the room constant, square feet. (R increases with the amount of acoustical absorption in the room and generally depends somewhat upon frequency.)

In Fig. 17 four methods are shown for terminating a ventilating duct in a room. In each of these cases, the noise power radiates into the room. Low-frequency sounds radiate equally in all directions. High-frequency sounds tend to *beam* in the direction the duct opening is facing. The *magnitude of this* beaming effect is described mathematically by the directivity factor Q. After the sound at any frequency has reflected from a wall, it generally travels around the room many times to produce reverberant sound. This fact appears in Equation 24, wherein the first term in the parenthesis described the *direct sound (prior to reflection)* and the second term described the *reverberant* sound (the result of many reflections).

The directivity factor Q is of importance only when the listener is near the duct termination, i.e., when r is small. At large distances, i.e., when the first term becomes negligible compared to the second, the listener hears only the reverberant sound.

For the four duct positions of Fig. 17, approximate values of Q are given in Fig. 18 for $\theta = 0$ deg and for $\theta = 45$ deg. The abscissa is the product of frequency times L assuming that the duct is approximately square with an area of L^2 sq in. If the duct is rectangular, the value of Q should be computed for L equal to the smaller dimension.

The room constant R is defined by the equation

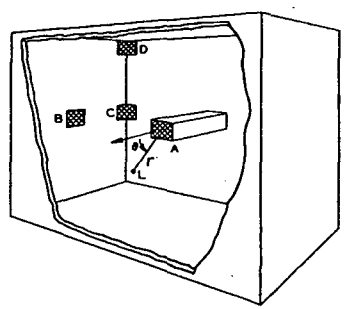
$$R = \bar{a}S/(1 - \bar{a}) \qquad (25)$$

where

$\bar{a}$ = average sound absorption coefficient (dimensionless) for the room at the mid-frequency of the band of noise being considered.
S = total area of the boundary surfaces of the room, sq ft.

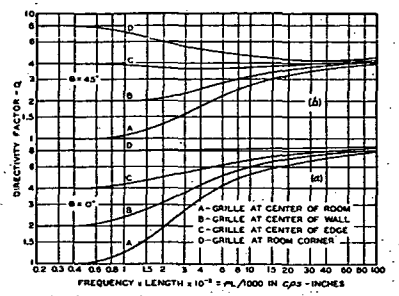
Where a room contains several surfaces, each with different absorption coefficients, the average absorption coefficient may be determined by the equation

$$\bar{a} = \frac{S_1 a_1 + S_2 a_2 + \cdots}{S_1 + S_2 + \cdots} \qquad (26)$$



A. Duct projecting in the room
B. Duct in the center of and flush with the wall
C. Duct in the center of one edge
D. Duct in the corner
The listener L is at distance r and angle θ from the duct opening.

Fig. 17 .... Four Typical Means for Terminating a Ventilating Duct in a Room



A. The value of Q is given for θ = 0, i.e., directly in front of the duct opening.
B. Same as for (A) except the value of Q is given for θ = 45 deg.
The frequency of f is in cps and the duct is assumed to be square with an area L^2 in square inches.

Fig. 18 .... Directivity Factor Q for the Four Duct Configurations of Fig. 17

where

S_1, S_2 = the areas of each type of absorbing material. a_1, a_2 etc., = the absorption coefficients of each type of material at the mid-frequency of the band of noise being considered. (Values of a for many materials are published by the Acoustical Materials Association.[14])

For estimating purposes, approximate values of the room constant R may be obtained from Fig. 19 as a function of room size for five classes of rooms.

The value of the second and third terms of Equation 24 (including 0.5 db) is plotted in Fig. 20 with distance r from the source as the abscissa, with the sound pressure level in decibels as the ordinate and the room constant R in sq ft as the parameter.

The directivity factor Q is given at the bottom left-hand side of the graph. The procedure for using the graph is as follows:

Determine the room constant R and the directivity factor Q for the particular direction θ of interest. Enter Fig. 20 at the bottom at the distance r. Then move diagonally to the left until the value of Q is reached. Then move vertically until the value R is reached. Then read the relative sound pressure level from the ordinate. An example is shown by the dotted line in Fig. 20 for $r = 7$ ft, $Q = 2$, and $R = 2000$ sq ft. The relative sound pressure level is added algebraically to the sound power level to yield the sound pressure level at the distance r and the angle θ in the room (see Equation 24).

It is seen from Fig. 20 that the noise from a ventilating duct is less at distances far from it than nearby. Moreover, at large distances, the levels produced in the room depend only on the power level of the source and the room constant and not on the directivity factor and the distance r.

Example 3: To illustrate the application of the noise criteria and control procedures outlined in this chapter, calculation will be made for the required treatment for the ventilation system supplying the library reading room of Fig. 21. This room has a total volume of 80,000 cu ft and may be considered as an *average room* for acoustic properties (see Fig. 19). The total air supply to the reading room is 6000 cfm which is equally divided between four 12 x 18 in. supply grilles located along one wall. The supply of outdoor air is from a central fan system which handles a total of 25,000 cfm at 2-in. water static pressure and requires a 15 hp motor.

Solution: It is necessary to determine the noise level in the reading room as a result of the ventilating system, both for the position of the nearest listener to a grille, and for a position remote from the supply grilles since it is impossible to tell beforehand which will be greater. The required attenuation for acceptable octave band noise levels will then be determined.

Analysis of Fan and Duct System

The acoustic power level of the fan should be computed from Equation 18 when the specific power level is available. In this example it is assumed that the specific sound power level is not known. Therefore, employ the approximate Equations 19 or 20. From Equation 19:

$$\text{PWL}_{(\text{duct AB})} = 100 + 10 \log 15 + 10 \log 2 = 114.7 \text{ dbe}$$

or from Equation 20:

$$\text{PWL}_{(\text{duct AB})} = 65 + 10 \log 25{,}000 + 20 \log 2 = 115 \text{ dbe}$$

which agreement is better than the accuracy expected from these equations.

Of this total acoustic power generated by the fan the main interest is in the part that reaches the reading room. At each duct branch the power will divide approximately as the ratio of the branch duct area to the total duct area after the branch. Thus at Station B (Fig. 21) the acoustic power delivered to the 18 x 48 in. duct, neglecting any attenuation in the system, will be

$$\text{PWL}_{(\text{duct BC})} = \text{PWL}_{(\text{duct AB})} + 10 \log (A_{\text{branch}}/A_{\text{total}})$$

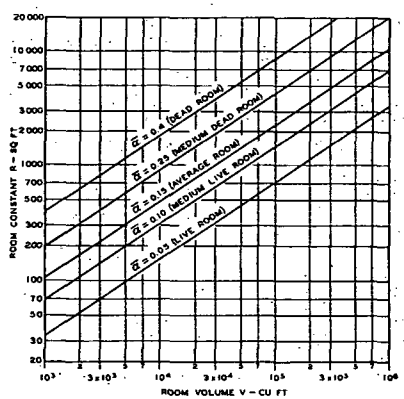
$$= 115 + 10 \log \left[\frac{18 \times 48}{(24 \times 48) + (18 \times 48)}\right] = 111 \text{ dbe}$$

At Station C the power division gives

$$\text{PWL}_{(\text{duct CD})} = 111 + 10 \log \left[\frac{12 \times 48}{(18 \times 24) + (12 \times 48)}\right]$$

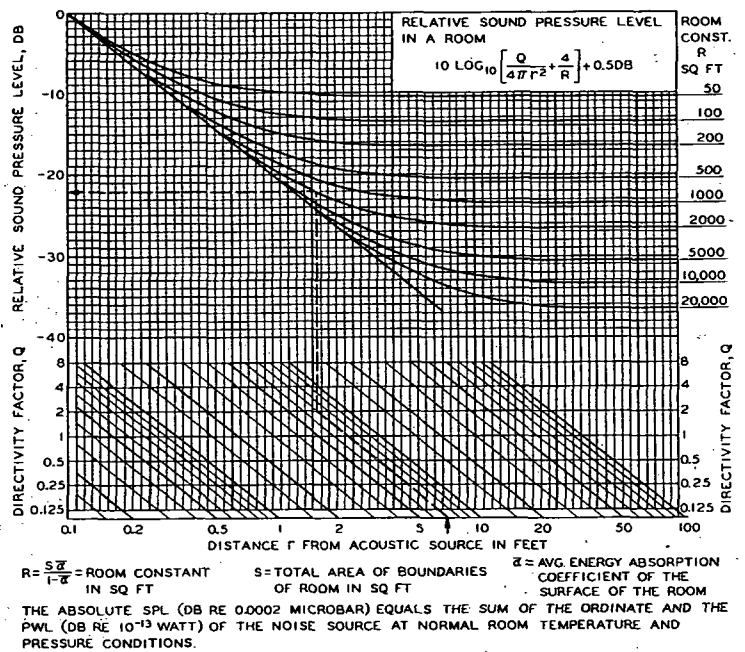
$$= 108 \text{ dbe}$$

Thus neglecting, at first, the attenuation in the connecting duct



These proportions give $S = 6.25V^{2/3}$. The parameter is the average sound-absorption coefficient for the room. The subjective ratings dead, live, etc., are the author's[20] and are not necessarily in standard use.
From Reference 20. Used by permission.

Fig. 19 .... Value of the Room Constant R as a Function of Room Volume for Rooms with Proportions of About 1:1.5:2



The listener's position is at a distance r and an angle θ with respect to the duct opening. Q is determined from θ and from the duct dimensions (see Fig. 18). To use the graph proceed as for the example shown by the dotted line for r = 7 ft, Q = 2, and R = 2000 sq ft. The sound pressure level in the room equals the sum of the ordinate and the sound power level. Normal room temperature and pressure conditions are assumed.

Fig. 20 .... Curves for Determining the Sound Pressure Level in the Room Relative to the Sound Power Level

system, the total acoustic power delivered to the room by the fan will be 108 dbe. Assuming that the branch fittings at *D*, *E*, and *F* are similar to that at *C* in Fig. 21, the unattenuated fan acoustic power delivered to each of the four 12 x 18 in. grilles will be approximately

$$PWL_{(\text{duct DB})} = 108 - 10 \log 4 = 102 \text{ dbe}$$

The octave band spectrum of the centrifugal fan noise is given by Fig. 6. This chart indicates that the power level in the first band (20–75 cps) will be about 102-1 or 101 dbe, and the power level in each succeedingly higher octave band will be 5 dbe less. These levels are entered in line 1 of the calculation Table 9.

The next step is to determine the natural duct attenuation between the fan discharge and the room supply grilles. From Table 4 the attenuation for straight sheet-metal ducts of *medium size* is 0.05 db per ft at all frequencies. Let us assume that all of the ducts of this system fall within this medium category. The total length of the duct run from Station *A* to Station *H*, Fig. 21, is 90 ft. The straight duct attenuation is, therefore, 0.05 (90) = 5 db, which is entered in line 2 of Table 9. It should be noted here that the ducts *CD* and *DH* of 12 x 48 and 12 x 15 in. dimensions, respectively, are of about the same size as those reported in Table 5. Hence if we assume the attenuation for these two *bare* ventilation ducts can be taken from Table 5, it will be found that the straight duct attenuation between *C* and *H* (35 ft) will be approximately 10 db in the lowest two octave bands, 6 db in the third band, and 3 db for all octave bands above 300 cps. That is, using Table 5 for ducts *CD* and *DH*, and Table 4 for ducts *AB* and *BC*, the first three columns in line 2, Table 9, would read 12, 12, and 9 db in that order, and the other columns would be unchanged at 5 db. It would appear, therefore, that the 5 db attenuation values for the lower three octave bands are probably overly conservative. The divided-flow fitting at *B* may be considered equivalent in attenuation to an 18-in. square bend. Values are read from Fig. 10, Part A and entered as line 3 in the calculation table. The rounded corners of the branch flow fittings at *C*, *D*, *E*, *F*, and *G* are assumed to have no attenuation. The 12 x 15 in. risers serving each of the wall grilles have a square bend (see Fig. 22), and attenuation values are read from Fig. 10, Part A and entered as line 4 of Table 9. The attenuation due to the end reflection loss at the grille face is given by Fig. 11. For the 12 x 18 in. grilles used here $L = \sqrt{12 \times 18} = 14.7$ in. Thus, for the first octave band (20–75 cps), which has a geometric mean frequency of 40 cps, $fL/1000 = (40 \times 14.7)/1000 = 0.58$ and the end loss is 13 db.

The attenuation in the other octave bands is similarly determined and entered in line 5 of Table 9. The total natural duct attenuation is now found by summing lines 2, 3, 4, and 5 of Table 9, and the result appears in line 6. This total attenuation *may now* be subtracted from the unattenuated power level in line 1 to give the estimated acoustic power level radiated into the room from the outlet grille at Station *H*. Line 7 of Table 9 gives the result.

Table 9 .... Summary of Calculations for *Example 3* on Ventilating Noise Levels in Libary Reading Room

Item	Octave Frequency Bands 20–75 cps	75–150 cps	150–300 cps	300–600 cps	600–1200 cps	1200–2400 cps	2400–4800 cps	4800–10,000 cps
1. $PWL_{(\text{duct DB})}$	101	96	91	86	81	76	71	66
2. Attenuation$_{(\text{st. duct})}$	5	5	5	5	5	5	5	5
3. Attenuation$_{(\text{bend B})}$	—	—	2	7	7	6	11	12
4. Attenuation$_{(\text{bend H})}$	—	—	1	4	7	7	8	10
5. Attenuation$_{(\text{end loss})}$	13	7	4	1	—	—	—	—
6. Total duct attenuation	18	12	12	17	19	18	24	27
7. $PWL_{(\text{grille outlet})}$	83	84	79	69	62	58	47	39
8. $fL/1000$ cps-in	0.5	1.3	2.5	5.1	10.2	20.4	40.8	81.6
9. Q	2	2	2.2	2.7	3.2	3.6	3.9	4.1
10. Rel. L_p at 7 ft	−22	−22	−22	−21	−20	−20	−20	−20
11. L_p at 7 ft	61	62	57	48	42	38	27	19
12. $PWL_{(\text{all grilles})}$	89	90	85	75	71	67	56	48
13. Rel. L_p far field	−26	−26	−26	−26	−26	−26	−26	−26
14. Reverberant L_p	63	64	59	49	45	41	30	22
15. Criterion L_p	60	51	43	37	32	30	28	27
16. Required attenuation	3	13	16	12	13	11	2	—
17. Attenuation 4 ft pkg. unit	7	11	14	30	40	40	38	33

Determining Room Sound Pressure Levels

The next step, then, is to calculate the sound pressure level at the position in the room occupied by the listener nearest to *any one grille, and for the overall reverberant sound pressure* level in the room due to the sound power radiated from all of the grilles. Fig. 22 shows that the position of the nearest listener is 7 ft from the grille. To determine L_p at this position, determine the directivity factor *Q* at 45 deg to the grille face; this is found from Fig. 18. The parameter $fL/1000$ is determined by taking the characteristic length *L* as the shortest side of the grille face or 12 in. Thus, for the first octave band, $fL/1000 = 40\ (12)/1000 = 0.48$. This value together with all the corresponding values for the other seven octave bands are entered in line 8 of Table 9. Line 9 of the table gives the corresponding directivity factors as read from Fig. 18, using curve *B* since the grille is flush with the wall but not at an edge or corner.

It was initially assumed that the library reading room had average acoustical properties, and consequently, the room constant *R* in sq ft may be read from the line marked *average room*

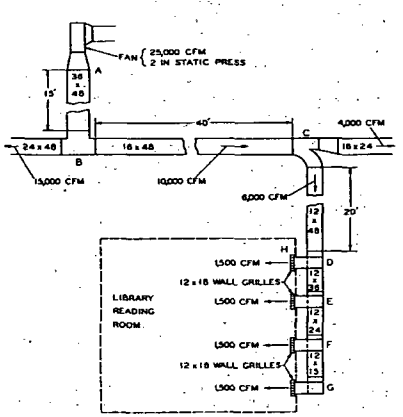


Fig. 21 .... Assumed Ventilating System for *Example 3*

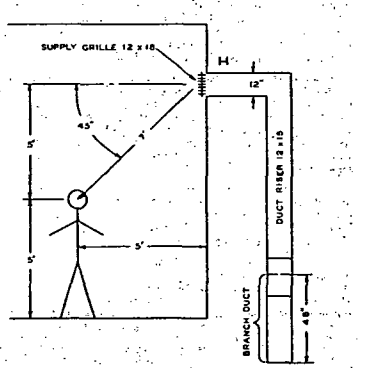


Fig. 22 .... Location of Nearest Listener to a Supply Grille

of Fig. 19. For a total room volume of 80,000 cu ft the result is R = 2000 sq ft. The relative sound pressure levels for an observer 7 ft from the grille may now be read from Fig. 20 making use of the directivity factors and the room constant as found. The dotted line in Fig. 20 shows that the relative L_p for the 20–75 cps band is −22 db. This relative L_p is entered in line 10 of Table 9. The sound pressure level at the 7-ft position from the grille is given by the addition of line 7 and line 10 in the calculation table. This result appears as line 11.

If there were only one grille in the room the calculation for room sound pressure level would be the only one required. However, since there are four grilles it is necessary to determine the general reverberant sound pressure level in the room due to the acoustic power radiated from these four sources to see if it is greater than the previous calculation. Neglecting the small additional duct attenuation which occurs in the somewhat longer duct runs to the other three room grilles, the total acoustic power radiated from the four grilles will be four times that from one grille. Hence the total acoustic power level from all grilles will be 10 log 4 or 6 db higher than from a single grille. In line 12 of Table 9, therefore, enter the PWL of all grilles by adding 6 db to the single grille value of line 7. The relative L_p for an observer in the reverberant sound field is read from the horizontal part of the R = 2000 sq ft curve of Fig. 20 as −26 db, and entered in line 13 of Table 9. The sum of line 12 and line 13 values from Table 9 gives, in line 14, the general reverberant sound pressure level in the room. Comparison of line 11 and line 14 shows that the latter is 2 db higher at the lower frequencies and, therefore, should be used in determining any noise treatment for the ventilating system or room.

Room Criteria

For this example the speech interference level criteria will be used. Examination of Table 1 shows that an acceptable noise level for libraries will be given by the *NC*-30 curve of Fig. 3. The values read from this curve are entered as line 15 of Table 9. The difference between these criterion sound pressure levels and the computed L_p in line 14 of the table gives the required attenuation for the ventilating system. This final result appears as line 16 in Table 9.

Treatment

The type of treatment of the ventilating system to obtain the attenuation given in line 16 of the calculation table will depend on many factors as discussed in the section on Sound Attenuation. One satisfactory treatment might employ a 4-foot package unit whose attenuation characteristics, as given in Fig. 15, are entered in the final line 17 of Table 9. The additional 2 db of attenuation required in the second and third octave bands could be obtained by lining approximately 10 ft of the 12 x 48 inch duct with 1-inch liner. However, as was pointed out above, the straight duct attenuation in line 2 is probably too conservative in these lower octave bands, hence the package unit would most likely provide a satisfactory treatment.

A calculation is now made of the acoustic power level generated by the grille itself. The face velocity of the grille is 1500 cfm/1.5 sq ft = 1000 fpm. From Fig. 8, assuming a vertical or horizontal bar deflection type grille, the average power level in the speech interference bands (600–1200, 1200–2400, 2400–4800 cps) for a face velocity of 1000 fpm is 41 dbe per square foot of grille area or 41 + 10 log 1.5 = 43 dbe for the grille in this example.

From line 10 of the calculation table the relative L_p in the speech interference bands (600–4800 cps) is −20 db. Consequently the L_p of the grille noise at 7 ft is 43 − 20 = 23 db. This is well below the 30 db criterion selected for this room and therefore, is satisfactory.

In reverberant fields all four grilles contribute so the total acoustic power level of the four grilles is 43 + 10 log 4 = 49 dbe. From line 13 of Table 9 the relative L_p is −26 db and therefore the reverberant L_p will be 49 − 26 or 23 db, which by coincidence is the same as that calculated at the 7-ft distance from one grille and, therefore, is also satisfactory.

The calculation of grille noise in this example was left to the end in order to keep a simple continuity to the principal problem of calculating the fan noise in the room. Experience shows that it is advantageous to calculate the grille noise early in the problem as indicated in the outline given in the text so that if the grille size must be changed its correct value may be used in calculating fan noise attenuation.

CROSS TRANSMISSION BETWEEN ROOMS AND THROUGH DUCT WALLS

Ducts serving more than one room permit cross talk between the rooms and should be lined with acoustical material. Where the rooms are close together and the ducts short, the ducts should be subdivided to provide ample acoustical treatment. Lagging material similar in character to acoustical board, when placed on the outside of ducts, serves to prevent noise, originating outside the ducts, being carried inside the ducts and into the air stream.

A case where outside lagging is desirable occurs when ducts originate at the fan in the equipment room and pass through this room on the way to the room being conditioned or ventilated. Unless the ducts are lagged, some of the mechanical noise from air in the equipment room may be transmitted through the wall of the duct into the air stream, and thereby carried into the room. In such cases, that portion of the duct which is exposed to the sounds in the equipment room should be lagged with material, such as cork, pipe covering, or other sound damping material, to prevent the sound from entering the duct at this point. Numerical data are not available to permit a simple and practical calculating procedure to determine thickness of covering which should be used for this purpose.

Laboratory measurements have shown that the loss through a sheet of No. 22 gage metal is 24 db. When a sheet of rock wool insulation 1 in. thick and weighing 1.4 lb per square foot is added to this, the insulation value is increased to 29 db. In general, however, adding a layer of insulation or pipe covering does not materially increase the sound insulation value unless the material is dense, or unless it is surfaced with another sound impervious layer such as metal or board. Standard reference books should be consulted for sound insulating properties of various materials. Inside lining material, used in the case previously mentioned, would serve as an absorber of the sound transmitted through the duct walls, and thus act as a means of preventing the transfer of noise into the air stream. Inside lining may also be used in ducts to absorb noise which reaches the air stream from equipment such as fans, sprays, and coils; noise due to eddying currents set up by elbows, dampers, and similar obstructions; and noise transmitted from room to room in a common duct system.

CONTROLLING VIBRATION FROM MACHINE MOUNTINGS

It is impossible to select equipment which will operate without producing some mechanical noise and, since the equipment must be mounted in a building, it is probable that a part of this noise will be transmitted to the building to such a degree as to make noisy conditions in the rooms which are to be air conditioned.

Much of this noise may be transmitted by the duct if it is rigidly connected to the fan outlet. It is common practice to make the connection between the fan and the duct with a canvas sleeve which effectively restricts noise at this point. Noise may also enter the building through the mounting of the motor and the fan. Flexible mountings should be provided in all installations, but these mountings must be carefully designed so that they will actually reduce the energy transmitted between the machinery and the supporting floor. If a flexible material is used, it is desirable to investigate the installation

so that it is not short-circuited by through bolts which are improperly insulated, and by electrical conduit which is not properly broken and is attached both to the equipment and to the building. The flexible mounting, if improperly engineered, may actually increase the energy transmitted between the equipment and the supporting floor.

In the proper isolation of vibration, which is usually in the lower range of frequencies and does not include the *airborne* vibrations known as sound, there is one basic formula which is important in the solution of the problem. It is the formula of transmissibility as governed by the equation:

$$T = \frac{1}{\left(\frac{f}{f_n}\right)^2 - 1} \qquad (27)$$

where

T = transmissibility of the support.
f = frequency of the vibratory force.
f_n = natural frequency of the machine unit on its support (damping = 0).

Equation 27 shows that the transmissibility approaches unity for disturbing frequencies considerably lower than the natural frequency of the mounting. As the disturbing frequency is increased, the transmissibility is also increased until at the resonant frequency, where $f = f_n$ the transmissibility becomes infinite. This is not true in practice because all materials have some internal damping effect. However, operating at or very close to the resonant frequency is always serious as forces and stresses may be multiplied 10 to 100 times. As the disturbing frequency becomes greater than the natural frequency, the transmissibility becomes a smaller quantity, and at the value of $f/f_n = \sqrt{2}$ it again has the value of unity. Beyond this point true isolation begins. At a ratio of 3 to 1 for f to f_n the isolation is effective enough for practical application, and experience and economical design have shown that a ratio of 5 to 1 is good. For high speeds, higher ratios for f to f_n are easily attained and give better results for effective vibration control. For the lower speeds as experienced with compressor work the higher ratios become uneconomical. At these lower speeds the unbalanced force goes down as the square of the speed ratio, so that it is quite practical to compromise and use a lower ratio of f to f_n, say 2 or 3 to 1.

For a given installation, the speed of the compressor is fixed by the specifications; therefore the value of f is fixed. That leaves only f_n to be determined, and that is accomplished by the choice of mounting material and design for the support of the machine. It is well to keep in mind that when trying to isolate vibration, no attempt should be made to isolate the driving and driven piece of equipment separately. The two should be mounted on a rigid frame, and then the entire assembly isolated according to the rules presented in this chapter.

The value of f_n can be controlled by the flexibility of the machine support, and when the deflection of the machine support is proportional to the load applied (such as with springs or nearly so with rubber-in-shear) the value of f_n can be determined by Equation 28:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{d}} \qquad (28)$$

where

g = gravitational constant = 32.2 feet per (second) (second).

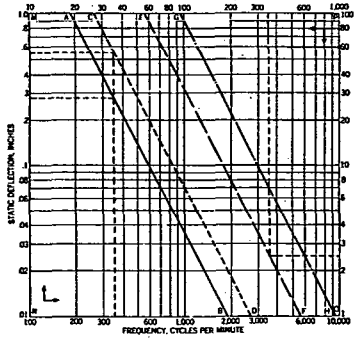


Fig. 23 .... Static Deflection for Various Frequencies

d = static deflection of supporting material, feet.
f_n = natural frequency of the machine unit on its support (damping = 0), cycles per second.

By the use of Equation 28 a set of curves may be plotted as shown in Fig. 23. The first line AB, plotted as the critical frequencies for the various static deflections, is a curve showing the worst possible conditions or resonant conditions.

Plotting another curve CD, which is $\sqrt{2}$ times curve AB, shows the area $MCDN$ in which the resilient material or mounting does more harm than good. Plotting curves EF (3 times curve AB) and GH (5 times curve AB) shows area $EGHP$ which represents efficient and economical isolation. Area $GPOH$ is excellent isolation, but for all except the highest speeds, becomes rather uneconomical because of the large deflections required.

Example 4: An electric motor driven compressor unit is to be isolated. The compressor is partially balanced and operates at a speed of 360 rpm. The speed of the motor is 1160 rpm, and it is belt-connected to the compressor. Total weight of the compressor and motor is 4500 lb.

Solution: The minimum disturbing frequency to be isolated is 360 cycles per minute. Assume that the desired ratio of forced to natural frequency is 3 as a minimum, and that 5 is desired. The desired natural frequency of the mounting is 360 ÷ 5 = 72 cycles per minute.

From Fig. 23 a deflection of 7 in. is required to attain a natural frequency of 72 cycles per minute. This value may be obtained from critical curve AB for 72 cycles, or from curve GH (5 times critical) for 360 cycles. For a ratio of 3 the deflection would be 2.5 in.

The next step is to determine the total weight to be supported by the springs. For low speed partially balanced compressors, it has been found necessary to add a foundation weighing 2 to 3 times the weight of the motor and compressor, in order to maintain the machine movement below 0.03 in.

Compressor and motor	4500 lb
Concrete foundation	9000 lb
Total	13,500 lb

Practical application dictates the number of springs to be used, which is based on the design of the machine foundation and the supporting floor structure. However, it is desirable to design for at least 8 springs and one or two spares for cases of unknown weights. As many as 50 springs have been used on one installation. The distribution of the springs must be balanced against the masses to be supported. Otherwise the

foundation design and supporting structure determine the location of the springs.

The choice of the material used in the design of the resilient mounting is also important. For the slow-speed type compressor, a common speed found in practice is 360 rpm. For speeds below this, isolation should not be attempted except under careful supervision. Referring to Fig. 23, it is found that for 360 rpm the static deflection required for a ratio of f/f_s of 3 to 1 (line *EF*) is 2.5 in., and for a ratio of 5 to 1 (line *GH*) it is 7 in. For these values of deflection the only choice of material is the coil spring. This is also true for speeds up to about 700 rpm. In consideration of the transverse spring constant (so as to maintain good ratios among the various degrees of freedom) experience has shown that the spring should be designed with a *working height* equal to 1.0 to 1.5 times the outside diameter. A long spring of small outside diameter has very low transverse rigidity, and therefore requires some additional means of preventing side drift of the unit, and on very sensitive applications this may tend to destroy the isolation efficiency. For speeds of 700 to 1200 rpm the required deflections range from 0.22 in. to 0.80 in. For speeds of 700 to 1200 deflections range from 0.22 in. to 0.8 inch. For these conditions springs or rubber-in-shear can be used. Special rubber is required if the isolators are likely to be contaminated by oil. For speeds higher than 1200 rpm rubber-in-shear, as well as cork specially made for vibration isolation, can be applied with good results. These limitations are by no means absolute, because certain liberties may be taken without impairing the result if all possible degrees of freedom have been taken into account in the design of the installation.

When a machine unit is properly isolated it will have a definite amount of movement which is determined by the ratio of the unbalanced forces to the total mass of the machine. If this resultant machine movement is too great for the necessary connections or the satisfaction of the customer, it can be reduced in two ways only without destroying the quality of the isolation; first, adding mass or dead weight to the machine (such as concrete) common in the application of low speed, partially balanced machinery; second, accurately balancing (both statically and dynamically) all moving parts so as to eliminate the vibration at the source. This latter method is the best engineering practice and is the modern trend. However, even with well balanced machinery, installed in the vicinity of quiet offices, it is usually necessary to properly isolate the equipment to prevent the transmission of vibration likely to cause complaints.

Where limitation of machine movement is desired during the starting and stopping periods, the application of friction or hydraulic damping will serve without seriously interfering with the efficiency of the isolation.

REFERENCES

[1] *American Standard Acoustical Terminology* (American Standards Association, Z24.1—1951).

[2] *Handbook of Noise Measurement* (General Radio Company, Cambridge, Massachusetts).

[3] S. S. Stevens: The measurement of loudness (*Journal Acoustical Society of America*, Vol. 27, September 1955, p. 815).

[4] *American Standard Sound Level Meters for Measurement of Noise and Other Sounds* (American Standards Association, Z24.3—1944).

[5] L. L. Beranek: Noise criteria for office spaces (*Journal Acoustical Society of America*, Vol. 28, September 1956).

[6] R. D. Madison and R. J. Wells: Fan noise (*Handbook of Noise Control*, ed. C. M. Harris, McGraw-Hill Book Co., New York, 1957).

[7] R. D. Madison (ed.): *Fan Engineering* (Buffalo Forge Company, Buffalo, New York, 1949, p. 313).

[8] B. B. Daly: *Noise Level in Fans* (*Institution of Heating and Ventilating Engineers Paper* No. B7925).

[9] L. L. Beranek, G. W. Kamperman, and C. H. Allen: Noise of centrifugal fans (*Journal Acoustical Society of America*, Vol. 27, March 1955, p. 217).

[10] C. H. Allen: Noise from air-conditioning fans (*Noise Control*, Vol. 3, January 1957, p. 28).

[11] C. F. Peistrup and J. E. Wesler: Noise of ventilating fans (*Journal Acoustical Society of America*, Vol. 25, March 1953, p. 322).

[12] J. B. Chaddock: *Ceiling Air Diffuser Noise* (Bolt Beranek and Newman, Inc., Cambridge, Massachusetts, *Technical Information Report* No. 45, August 28, 1957).

[13] T. L. Moore, H. H. Bell, and J. K. Nunnely: Noise attenuation in straight ventilation ducting (S. M. thesis, Massachusetts Institute of Technology, June 1957).

[14] D. A. Wilbur and F. R. Simons: ASHVE Research Report No. 1205—Determining sound attenuation in air conditioning systems (ASHVE Transactions, Vol. 48, 1942, p. 267).

[15] H. J. Sabine: The absorption of noise in ventilating ducts (*Journal Acoustical Society of America*, Vol. 12, 1940, p. 53).

[16] For coefficients of commercial sound absorbent materials see *Acoustical Materials Association Bulletin* (Acoustical Materials Association, New York).

[17] L. L. Beranek: Sound absorption in rectangular ducts (*Journal Acoustical Society of America*, Vol. 12, October 1940, p. 237).

[18] P. M. Morse: The transmission of sound inside pipes (*Journal Acoustical Society of America*, Vol. 11, October 1939, p. 205).

[19] R. J. Wells: Noise control problems in air conditioning equipment (*Heating, Piping and Air Conditioning*, Vol. 29, August 1957, p. 138).

[20] L. L. Beranek: *Acoustics* (McGraw-Hill Book Co., New York, 1954, Chapter 10).

CHAPTER 26

STEAM HEATING SYSTEMS

Classification of Steam Heating Systems by Types; One-pipe, Two-pipe, Subatmospheric and Orifice Systems; Sizing Piping for Steam Heating Systems; Pressure-Reducing Valves; Boiler Connections; Condensate Return Pumps; Vacuum Heating Pumps; Traps; Drips; Connections to Heating Units; Control Valves

STEAM heating systems may be classified according to any one of, or combination of, the following features: (1) piping arrangement, (2) pressure or vacuum conditions obtained in operation, (3) method of returning condensate to the boiler.

Classification by Piping Arrangement

A steam heating system is known as a *one-pipe system* when a single main serves the dual purpose of supplying steam to the heating unit and conveying condensate from it. Ordinarily, to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used.

A steam heating system is known as a *two-pipe system* when each heating unit is provided with two piping connections, and when steam and condensate flow in separate mains and branches.

Heating systems may also be described as *up-flow* or *down-flow*, depending on the direction of steam flow in the risers; and as a *dry-return* or a *wet-return*, depending on whether the condensate mains are above or below the water line of the boiler or condensate receiver.

Classification by Pressure or Vacuum Conditions

Steam heating systems may also be classified as *high-pressure*, *low-pressure*, *vapor*, and *vacuum* systems, depending on the pressure conditions under which the system is designed to operate.

A system is known as a *high-pressure system* when the operating pressures employed are above 15 psig; as a *low-pressure system* when pressures vary from 0 to 15 psig; as a *vapor system* when the system operates under both vacuum and low-pressure conditions without the use of a vacuum pump; and as a *vacuum system* when the system operates under vacuum and low-pressure conditions with the use of vacuum pump.

When automatic controls are employed to vary the pressure conditions in the system in accordance with outside weather conditions, the system may be known as a *subatmospheric*, *differential*, or *synchronized* system. These latter classifications are proprietary designations.

When orifices are employed on the inlets to the heating units the system may be known as an *orifice system*.

Classification by Method of Returning Condensate

When condensate is returned to the boiler by gravity, the system is known as a *gravity return system*. In this system all heating units must be elevated sufficiently above the water line of the boiler, so that the condensate can flow freely to the boiler. Elevation of the heating units above the water line must therefore be sufficient to overcome pressure drops due to flow, as well as pressure differences due to operation.

Referring to Fig. 1 it will be noted that the boiler and wet-return form a U-shaped container, with the boiler steam pressure on the top of the water at one end, and the steam main pressure on the top of the water at the other end. The difference between these two pressures is the *pressure drop* in the system, i.e., the friction and resistance to the flow of steam in passing from the boiler to the far end of the main, and the pressure reduction in consequence of the condensation occurring in the system. The water in the far end will rise sufficiently to overcome this difference in order to balance the pressures, and it will rise far enough to produce a flow through the return pipe and overcome the resistance of check valves, if installed.

If a one-pipe steam system is designed, for example, for a total pressure drop of ⅛ psi, and utilizes a Hartford return connection instead of a check valve on the return, the rise in the water level at the far end of the return, due to the difference in steam pressure, would be ⅛ of 28 in. (28-in. head being equal to one pound per square inch), or 3½ in. Adding 3 in. to overcome the resistance of the return main, and 6 in. as a factor of safety for heating up, gives 12½ in. as the distance the bottom of the lowest part of the steam main and all heating units must be above the boiler water line. The same system, however, installed and sized for a total pressure drop of ½ psi, and with a check valve in the return, would require ½ of 28 in., or 14 in. for the difference in steam pressure, 3 in. for the flow through the return, 4 in. to operate the check valve, and 6 in. for a factor of safety, making a total of 27 in. as the required distance. Higher pressure drops would increase the distance accordingly.

When conditions are such that condensate cannot be returned to the boiler by the action of gravity, and either traps or pumps must be employed, the system is known as a *mechanical return system*. There are three general types of

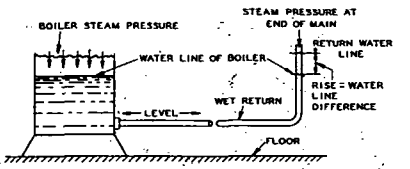


Fig. 1.... Difference in Steam Pressure on Water in Boiler and at End of Steam Main

mechanical condensate return devices in common use: (*a*) the alternating return trap, (*b*) the condensate return pump, and (*c*) the vacuum return pump.

In systems where pressure conditions in the system vary between that of a gravity return and a forced return system, a boiler return trap or alternating receiver is employed and the system may be known as an *alternating return system.*

When condensate is pumped to the boiler under pressures of the atmosphere or above, the system is known as a *condensate pump return system.*

When condensate is pumped to the boiler under vacuum conditions, the system is known as a *vacuum return system.*

In either the *condensate* or *vacuum pump systems* it is highly desirable to arrange for gravity flow to a receiver and to the pump. The pump then forces condensate into the boiler against its pressure.

ONE-PIPE SYSTEMS

One-pipe systems, as previously defined, are systems in which steam and condensate flow in the same pipe. Radiators and other heating units, in general, have only one piping connection from main to unit, although it is possible to employ two connections to the same main as indicated in Fig. 2. Unit heaters in one-pipe systems may also have separate connections to the wet-return.

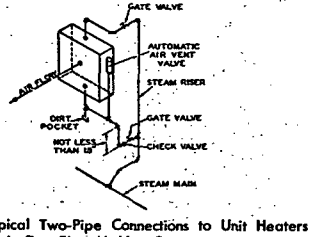


Fig. 2 .... Typical Two-Pipe Connections to Unit Heaters in One-Pipe Air-Vent Systems

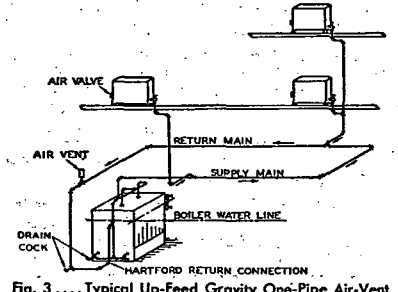


Fig. 3 .... Typical Up-Feed Gravity One-Pipe Air-Vent System

There are several variations in the piping arrangement of a one-pipe system as follows:

1. *Up-feed one-pipe systems where the radiators and other heating units are located above the supply mains.* The mains in this instance convey both steam and condensate. Therefore, it is preferable to limit this design to systems with a connected load under 2500 sq ft EDR. Such a system is illustrated in Fig. 3. Typical connections to radiator or risers are illustrated in Figs. 4 and 5 and method of changing sizes of mains in Fig. 6.

Branches from steam mains in one-pipe gravity steam systems should use the *preferred* connection shown in Fig. 7, but where radiator condensate does not flow back into the main, the *acceptable* method shown in the same figure may be used. This acceptable method has the advantage of giving a perfect swing joint when connected to the vertical riser or radiator connection, whereas the preferred connection does not give this swing without distorting the angle of the pipe. Runouts are usually made about 5 ft long to provide flexibility for movement in the main.

2. *Up-feed one-pipe system with a connected load in excess of 2500 sq ft EDR, where radiators and other heating units are located above the mains, as illustrated in Fig. 8.* It will be noted that the mains are dripped at each riser connection to a wet-return, as indicated in Fig. 5, so that the mains carry a minimum of the condensation. This feature is always desirable in the design of any one-pipe steam system. Radiators immediately above the mains may be connected as shown in Fig. 4. Up-feed systems are not recommended for systems higher than four or five stories.

3. *Down-feed one-pipe systems, where the radiators and other heating units are located below the supply main.* In this

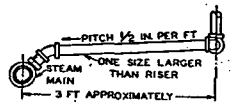


Fig. 4 .... Typical Steam Runout where Risers Are Not Dripped

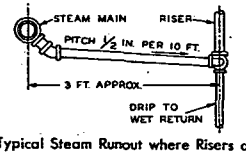


Fig. 5 .... Typical Steam Runout where Risers are Dripped



Fig. 6 .... Method of Changing Size of Steam Main when Runouts are Taken from Top



Fig. 7 .... Methods of Taking Branch from Main

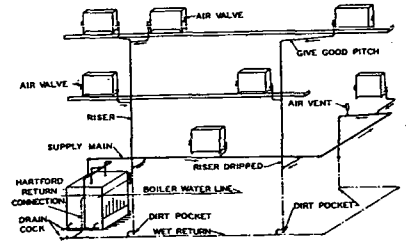


Fig. 8 .... Preferred Up-Feed Gravity One-Pipe Air-Vent System

arrangement only risers and connections to heating units convey both steam and condensate, and both are flowing in the same direction. The steam main is kept relatively free of condensate by dripping through the drop risers. Drain valves should be installed on all wet-return mains and on the return header at the boiler.

Each radiator or heating unit in a one-pipe system must be supplied with a thermostatic air valve which functions to relieve air from the heating unit under pressure, and to close when steam itself heats the thermostatic element of the valve.

To improve steam circulation in one-pipe systems quick-vent air valves should be provided at the ends and at intermediate points where the steam main is brought to a higher elevation, or where dropped below the water line. It is desirable to install the air-vent valves about a foot ahead of the drips, as indicated in Fig. 8, to prevent possible damage to their mechanisms by water.

Air valves are of two general types, the pressure and the vacuum types. The pressure type permits the inflow of atmospheric air to the system when the steam pressure in the system falls below atmospheric pressure. The vacuum type, which contains a small check valve, prevents the air from flowing back to the system and thereby maintains vacuum conditions in the system, and a consequent evaporation or generation of steam or vapor at subatmospheric pressures, and at consequent lower temperatures. Systems which use vacuum valves are known as *vapor* or *vacuum*

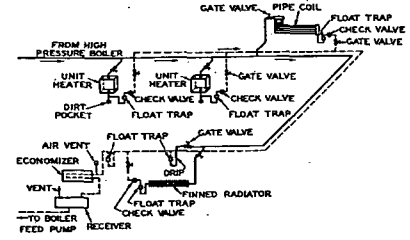


Fig. 9 .... Typical High-Pressure Heating System

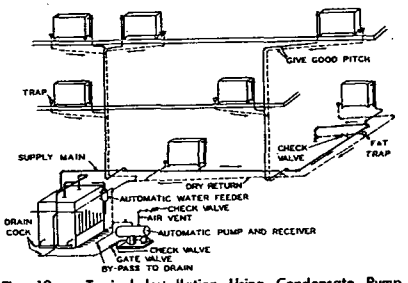


Fig. 10 .... Typical Installation Using Condensate Pump

one-pipe systems. The vapor or vacuum systems will maintain a more uniform temperature condition than the pressure *systems.*

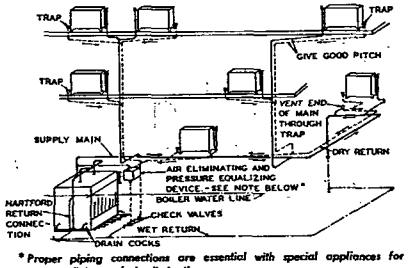
Each heating unit in a one-pipe system may also be provided with a valve on the connection to the unit, although this is not essential except to shut the unit off when it is not desired for heating. Valves on one-pipe systems must be either fully opened or fully closed. No throttling or modulating position can be maintained since, if a valve is partially closed, condensate will not drain from the unit. This condition is dangerous because it may create a low water condition in the boiler with consequent burning or cracking of the boiler, or create a hazard due to the freezing of the waterlogged heating unit itself.

TWO-PIPE SYSTEMS

Two-pipe systems, as previously defined, are systems in which steam and condensate flow in separate pipes. Two-pipe systems may operate under high-pressure, low-pressure, vapor, or vacuum conditions. Either the up-flow or the down-flow arrangement of mains may be employed.

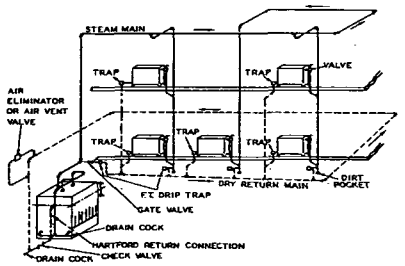
Two-Pipe High-Pressure Systems

Two-pipe high-pressure systems operate at pressures above 15 psig, usually from 30 to 150 psig. They are usually used in large industrial buildings, which are equipped



* *Proper piping connections are essential with special appliances for pressure equalizing and air elimination.*

Fig. 11 .... Typical Up-Feed Two-Pipe System with Automatic Return Trap*



Note: When an air eliminator or vent valve is used on a vapor system, the boiler pressure must not exceed the static head necessary for the return by gravity of the condensate.

Fig. 12 .... Typical Down-Feed Two-Pipe System

with unit heaters or large built-up fan units, or in which high-pressure steam is required for process work.

Fig. 9 illustrates a typical high-pressure system. Because of the high pressures and the great differential between steam and return mains, it is possible to locate returns above the heating units and lift the condensate to these returns.

The condensate can be flashed into steam in low-pressure mains if any are available, or passed through an economizer heater before being discharged to a vented receiver. It is, of course, necessary to provide for the elimination of air from high-pressure systems, just as in low-pressure systems.

Return traps used on high-pressure systems are usually of the bucket, inverted-bucket, float, or impulse type. They are described in a later section Steam Traps.

Two-Pipe Low-Pressure Systems

Low-pressure systems operate at pressures of 0 to 15 psig. The piping arrangement of both up-feed and down-feed low-pressure systems is identical with those of two-pipe vapor systems described in the following section. The only difference between the two systems is in the type of air valve used. The air valves used in low-pressure systems usually do not contain the check discs and hence, the system cannot operate under a vacuum. The low-pressure systems are not as popular as the vapor systems, because they

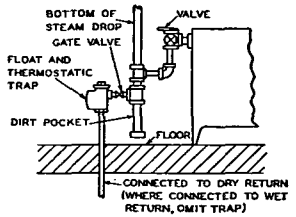


Fig. 13 .... Detail of Drip Connections at Bottom of Down-Feed Steam Drop

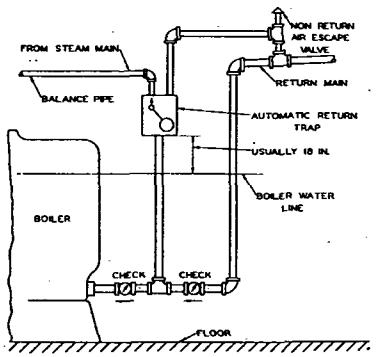


Fig. 14 .... Typical Connections for Automatic Return Trap

have the disadvantage of not holding heat when the rate of steam generation is diminishing. They also have the disadvantage of corroding to a greater extent than vapor systems, due to the continued presence of new air in the system.

Low-pressure systems have the advantage, however, of returning condensate to the boiler readily and not retaining it in the piping, as may be possible in vapor systems when the system pressure exceeds the static head provided for the gravity return at the boiler. Fig. 10 illustrates the conversion of a conventional gravity return system to a typical low-pressure system with a condensation pump. A check valve and a gate valve must always be installed between the pump discharge and the boiler. A check valve on the receiver vent will prevent the backflow of air when the system is cooling down.

Two-Pipe Vapor Systems

Two-pipe vapor systems operate at pressures varying from 20-in. vacuum or more (depending upon the tightness with which the system is assembled) to 15 psig without the use of a vacuum pump. A typical two-pipe up-feed vapor system is shown in Fig. 11, and a typical two-pipe down-feed system is illustrated in Fig. 12. The method of dripping drop

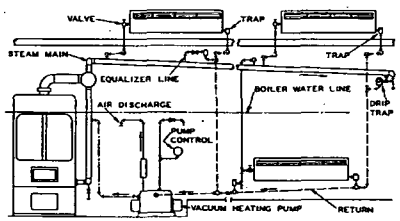


Fig. 15 .... Typical Two-Pipe Vacuum Pump System

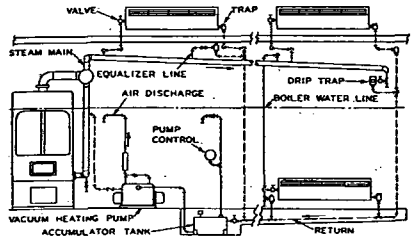


Fig. 16 .... Accumulator Tank Installation in Typical Two-Pipe Vacuum Pump System

risers in a down-feed system is illustrated in Fig. 13. Radiators discharge their condensate and air through thermostatic traps to the dry-return main. Air is eliminated, when the system is under pressure, at the ends of the supply and return mains just before they drop to the wet return. The vent valves are of the float rather than float and thermostatic type since there should be no steam in the return main as long as the thermostatic traps on the radiators or convectors are operating properly. The float element of the valve closes the valve when, due to pressure differences, water rises to the point of overflow in the main. The vent valves are also provided with a small check disc which closes to prevent the inflow of air to the system when the pressure drops below atmospheric pressure. This enables the system to operate under vacuum conditions at lower steam temperatures for a period of four to eight hours, depending on the tightness of the system.

Vapor systems may also be provided with an automatic return trap or alternating receiver which automatically returns condensate to the boiler when the boiler is steaming under pressure conditions which would prevent the return of condensate by gravity. The typical connections for an automatic return trap are illustrated in Fig. 14. The two check valves are necessary for its successful operation.

Each heating unit in a vapor system, as in all two-pipe systems, is provided with a graduated or modulating valve which permits the control of heat in the radiator by varying the opening of the valve.

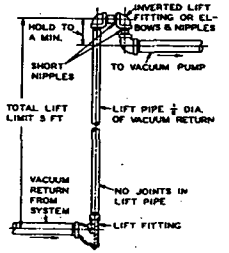
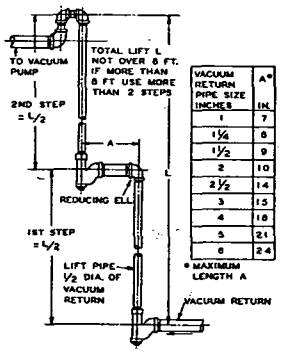


Fig. 17 .... Method of Constructing a Lift in a Return Line

Two-Pipe Vacuum Systems

Vacuum systems operate under conditions of both low pressure and vacuum, but employ the vacuum pump to insure maintenance of subatmospheric pressures in the return piping for all operating conditions. The pump thus assists the supply steam in overcoming the resistance of the system piping and insures rapid circulation and even heating. The system may operate transiently with subatmospheric pressure in the supply piping during the time the rate of steam generation is equivalent to or less than the total connected load.

A typical two-pipe vacuum system is illustrated in Fig. 15.



VACUUM RETURN PIPE SIZE INCHES	A* IN.
1	7
1¼	8
1½	9
2	10
2½	14
3	15
4	18
5	21
6	24

Fig. 18 .... Construction of a Step Lift

The return risers are connected in the basement into a common return main which slopes downward toward the vacuum pump. The vacuum pump withdraws the air and water from the system, separates the air from the water and expels it to atmosphere, and pumps the water back to the boiler or other receiver, which may be a feed-water heater or hot well. It is essential that no connection be made from the supply side to the return side at any point except through a trap. The return system should provide unrestricted gravity flow of the condensate and air to the pump receiver so that the vacuum differential may be more uniformly applied throughout the return system. In some instances, load conditions make it necessary to drop the return below the level of the vacuum pump inlet before this pump can be reached. This can be taken care of when using a vacuum pump by installing a float controlled auxiliary accumulator tank, pitted if necessary, just ahead of the vacuum pump inlet as shown by Fig. 16. The auxiliary tank as shown still permits an uninterrupted gravity flow from all return lines, but it does reduce the vacuum on the returns by 1-in. Hg for every foot of lift. The lift should be limited to 5 ft.

Lifts should be avoided, but on individual mains near the end of the system a lift fitting may be employed. Fig. 17 shows a single-step lift, which may be used up to 5 ft of lift. When the lift is greater but does not exceed 8 ft, it should be divided into two lifts shown in Fig. 18. It is preferable

to use commercial stock sizes of lift fittings, but if these are not available in the sizes required a satisfactory lift connection can be made from standard pipe fittings as shown by Fig. 19. When a lift assembly requires a union, it should be installed in a horizontal run. The use of a lift in any part of the system results in slower heat-up as well as interrupted return of the condensate from the section ahead of the lift. Lift fittings should not be used in place of an auxiliary tank in the return main at the inlet to the vacuum pump receiver.

A condensate pump may be used as a mechanical lift where severe outlying lift conditions cannot be avoided. This may be piped into the system as shown by Fig. 20 in such a manner as not to interfere with the normal functioning of the vacuum pump and without loss of vacuum on the returns preceding the condensate pump.

TWO-PIPE SUBATMOSPHERIC SYSTEMS

Subatmospheric systems are similar to vacuum systems but, in contrast, provide control of building temperature by variation of the heat output from the radiators. The radiator heat emission is controlled by varying the pressure, temperature, and specific volume of steam in circulation. These systems differ from the ordinary vacuum system in that they maintain a controllable partial vacuum on both the supply and return sides of the system, instead of only on

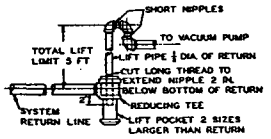


Fig. 19 .... How to Build a Lift Fitting

the return side. In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the subatmospheric system, atmospheric pressure or higher exists in the steam supply piping and radiators only during severe weather. Under average winter temperature the steam is under partial vacuum which in mild weather may reach as high as 25-in. Hg, after which further reduction in heat output is obtained by restricting the quantity of steam.

The rate of steam supply is controlled by a valve in the steam main or by thermostatically controlling the rate of steam production in the boiler. The control valve may be of the automatic *modulating* or *floating* type governed thermostatically from selected control points in the building, or it may be a special pressure reducing valve which will maintain the desired subatmospheric pressures by continuous flow into the heating main. In some systems radiator supply valves include adjustable orifices, or are equipped with regulating orifice plates. The sizes of orifices used are larger than for other types of orifice systems because, for equal radiator sizes, the volume flowing is larger. Orifices are omitted on some systems. Radiator traps and drips are designed to operate at any pressure from 15 psig to 26-in. Hg.

Steam for heating domestic hot water should be taken from the boiler header back of the control valve so that pressures sufficiently high for heating the water may be maintained on the heater. The subatmospheric method of

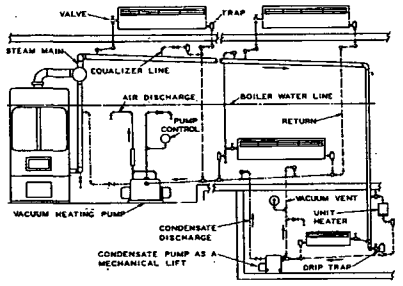


Fig. 20 .... Use of Condensate Pump as a Mechanical Lift

heating can be used for the heating coils of ventilating and air-conditioning systems. The flexible control of heat output secured by this method materially reduces the required size of bypass around the heaters. Some applications of subatmospheric systems are proprietary.

TWO-PIPE ORIFICE SYSTEMS

Orifice steam heating systems may have piping arrangements identical with vacuum systems. Some of these omit the radiator thermostatic traps, but use thermostatic or combination float and thermostatic traps on all drip points. A return condensate pump with receiver vented to atmosphere, a return-line vacuum pump, or a return trap, is generally used to return the condensate to the boiler or place of similar disposition, such as a feed-water heater or hot well. The heat emission from the radiators is controlled by varying the pressure differential maintained.

The principle on which these systems operate is that the steam flow through an orifice will vary with square root of pressure drop when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 percent. If the absolute pressure on the outlet side is less than 58 percent of the absolute pressure on the inlet side, no further increase in flow will be obtained as a result of the increased pressure difference. If an orifice is so designed in size as to exactly fill a radiator with 2 psig steam on one side and ¼ psig on the other, the absolute pressure relation is

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 0.90 \text{ or } 90 \text{ percent.}$$

If the steam pressure were dropped to ¼ psig on the supply pipe, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be apparent that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, reducing this steam main pressure will permit filling various desired portions of the radiator down to the point where the main pressure equals the back pressure in the radiator, provided the supply pipe pressures are closely controlled. If orifices are designed on a similar basis for a given system and proportioned to the heating capacity of the radiators they serve, all radiators will heat proportionately to the steam pressure. The range of pressure variation is limited by the permissible noise level of the steam flowing under the pressure difference required for

maximum heat output. The control of the steam supply is obtained by a valve placed in the steam main to maintain a determined pressure, and by varying the vacuum in the return lines. The valves are frequently set manually from a remote location, guided by temperature indicating stations in the building; or thermostatically controlled from a thermostat on the roof, which automatically measures the differential of outdoor and indoor temperatures. Since the range through which the pressures may be varied is usually from 0 to 4 psig, the control should be capable of maintaining close regulation of the desired space temperatures, particularly in mild weather.

A recommended orifice schedule is shown in Table 1. Some systems use orifices not only in radiator inlets, but also at different points in the steam supply piping for the purpose of balancing the system to a greater extent. In this manner the difference between the initial and terminal pressure in the steam main may be compensated to a great extent. For example, if the initial pressure is 3 psig and the pressure at the end of the main is 2 psig, an orifice could be used in each branch for the purpose of obtaining a more uniform pressure throughout the system. Such a provision may be particularly useful in this system for branches close to the boiler where the drop in the main has not yet been produced. Some orifice systems are proprietary.

SIZING PIPING FOR STEAM HEATING SYSTEMS

The functions of the piping system are the distribution of the steam, the return of the condensate, and, in systems where no local air vents are provided, the removal of the air. The distribution of the steam should be rapid, uniform, and without noise, and the release of air should be facilitated as much as possible, because an air bound system will not heat readily nor properly. In designing the piping arrangement, it is desirable to maintain equivalent resistances in the supply and return piping to and from a radiator. Arrangement of the piping so that the total distance from the boiler to the radiator is the same as the return piping distance from the heating unit back to the boiler, tends to obtain such a result. The condensate which collects in steam piping as well as in radiators must be drained to prevent interference with the ready flow of the steam and air. The effect of back pressure in the returns and excessive revaporization, such as occurs where condensate is released from pressures considerably higher than the vacuum or pressure in the return, must be avoided.

It is important that steam piping systems distribute steam not only at full *design* load, but during excess and partial loads. Usually the average winter steam demand is *less than half* of the demand at the *design* outdoor temperature. Moreover, in rapidly warming up a system even in moderate weather, the load on the steam main and returns may exceed the maximum operating load for severe weather, due to the necessity of raising the temperature of the metal in the system to the steam temperature, and the building to the *design* indoor temperature. Investigations of the return of condensate have revealed that as high as 143 percent of the design condensation rate may exist under conditions of actual operation. Expressed in gallons per minute per 1000 sq ft equivalent direct radiation the theoretical condensing rate of the system at the design indoor temperature is 0.5. During the warming-up period this value will approach 0.72 gpm.

The piping design of a heating system is greatly influenced by its operating characteristics. Heating systems do

Table 1 .... Orifice Capacities for Low-Pressure Steam Systems—Pounds per Hour

This table is based on data from actual tests[a]

Orifice Diameter 64ths of an Inch	6 in. Hg Differential	5 in. Hg Differential	4 in. Hg Differential
7	4.5–5.8	4.0–5.3	3.8–4.8
8	5.8–7.3	5.3–6.8	4.8–6.3
9	7.3–9.0	6.8–8.3	6.3–7.5
10	9.0–11.0	8.3–10.0	7.5–9.3
11	11.0–13.0	10.0–12.0	9.3–11.0
12	13.0–15.5	12.0–14.3	11.0–12.8
13	15.5–18.0	14.3–16.5	12.8–14.8
14	18.0–20.8	16.5–19.0	14.8–16.8
15	20.8–23.5	19.0–21.5	16.8–19.0
16	23.5–26.5	21.5–24.3	19.0–21.5
17	26.5–29.8	24.3–27.3	21.5–24.3
18	29.8–33.3	27.3–30.5	24.3–27.0
19	33.3–37.0	30.5–33.8	27.0–30.0
20	37.0–40.8	33.8–37.3	30.0–33.3
21	40.8–44.8	37.3–41.0	33.3–36.3

Orifice Diameter 64ths of an Inch	2 in. Hg Differential	1 in. Hg Differential	Orifice Area sq in.
7	2.5–3.3		0.00940
8	3.3–4.3	2.0–2.8	0.01227
9	4.3–5.3	2.8–3.5	0.01553
10	5.3–6.5	3.5–4.3	0.01917
11	6.5–7.8	4.3–5.0	0.02320
12	7.8–9.3	5.0–6.0	0.02761
13	9.3–10.8	6.0–7.0	0.03241
14	10.8–12.3	7.0–8.0	0.03758
15	12.3–14.0	8.0–9.3	0.04314
16	14.0–16.0	9.3–10.5	0.04909
17	16.0–18.0	10.5–11.8	0.05542
18	18.0–20.0	11.8–13.0	0.06213
19	20.0–22.0	13.0–14.5	0.06922
20	22.0–24.5	14.5–16.0	0.07670
21	24.5–26.8	16.0–17.8	0.08456

Note.—The radiator orifice plates recommended in this table are made of brass stampings 0.023 in. thick cup-shaped to be inserted in radiator valve unions.

[a] S. S. Sanford and C. B. Sprenger: Flow of steam through orifices into radiators (ASHVE Transactions, Vol. 37, 1931, p. 371).

not operate under constant conditions, because conditions change continually, due to variation in load. As the system is being filled with steam, the pressures existing in various locations may be different from those which exist for appreciable periods at other locations, although at equilibrium conditions the pressures are approximately the same. In designing piping it is of particular importance to arrange the system to preclude trouble caused by such pressure differences. The systems which readily release the air, permit uniform pressures to be attained in much shorter time intervals than those which are sluggish. Results are given in Fig. 21, from investigations[1] to determine the rate of condensate and air return from a two-pipe gravity heating system. Variations in the steam pressure during the warming-up period, when the rate of air elimination and condensation is high, are clearly indicated in these curves.

It is evident that the condensate flow during the initial warming-up period reaches a peak, which is greater than the constant condensing rate eventually reached when the pressure becomes uniform. Moreover, the peak condensing

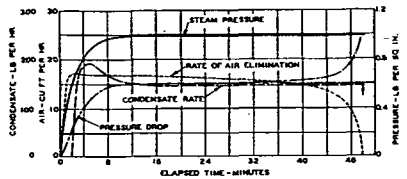


Fig. 21 .... Relation Between Elapsed Time, Steam Pressure, Condensate and Air Elimination Rates

rate is obtained when the system steam pressure is lower than that existing during a period of constant condensing rate. It will also be noted that the peak rate of air elimination does not coincide with the higher condensing rate.

Steam Flow

The rate of flow of dry steam, or steam with a small amount of water flowing in the same direction, is in accordance with the general laws of gas flow, and is a function of the length and diameter of the pipe, the density of the steam, and the pressure drop through the pipe. This relationship, developed by Unwin and later by Babcock, has been used for years as a means of determining steam flow through pipes. The new charts for weight-flow rate, pressure drop, and velocity presented in this chapter take into account the Reynolds number and its effect on friction loss, two items which were not considered when using the Unwin or Babcock formulas.

The data presented in Figs. 22, 23, 24, 25, 26 and Table 5, in later sections of this chapter, are based on the Moody Friction Factor where the absolute roughness of the internal pipe surface is that of new commercial steel or wrought iron. The Reynolds number, as expressed by Equation 9, Chapter 4, is

$$N_{Re} = \frac{dV\rho}{\mu} \qquad (1)$$

where

V = velocity of the steam, feet per second.
d = internal diameter of the pipe, feet.
ρ = density of saturated steam at a specified saturation pressure, pounds per cubic foot.
μ = absolute viscosity, pounds per foot-second.

Table 2 .... Pressure Drops in Common Use for Sizing Steam Pipe[a]

(For Corresponding Initial Steam Pressures)

Initial Steam Pressure, Psig	Pressure Drop Per 100 Ft, Psi	Total Pressure Drop in Steam Supply Piping, Psi
Subatmos. or vacuum return	2-4 oz	1-2 psi
0	½ oz	1 oz
1	2 oz	1-4 oz
2	2 oz	8 oz
5	4 oz	1½ psi
10	8 oz	3 psi
15	1 psi	4 psi
30	2 psi	5-10 psi
50	2-5 psi	10-15 psi
100	2-5 psi	15-25 psi
150	2-10 psi	25-30 psi

[a] Equipment, control valves, etc., must be selected on the basis of delivered pressures.

The velocity is determined by the equation

$$V = \frac{W}{3600\rho A} \qquad (2)$$

where

W = given weight-flow rate, pounds per hour.
A = internal pipe area, square feet.

The values used for the absolute viscosity of saturated steam are those of Lieb, published in *Combustion*, December, 1940. The relative roughness of the internal pipe surface is obtained by dividing the absolute surface roughness by the internal pipe diameter. The values for absolute roughness are those presented by L. F. Moody in *Mechanical Engineering*, Vol. 69, 1947, p. 1005, and appear in Table 1, Chapter 4. From the Reynolds number and relative roughness, the friction factor f is obtained from the Moody Friction Factor Chart, Fig. 4 of Chapter 4.

The head loss in feet is given by the equation

$$h = \frac{fLV^2}{2gd} \qquad (3)$$

where

f = friction factor.
L = length of pipe (100 ft).
g = acceleration due to gravity, 32.174 feet per (second) (second).

Table 3 .... Comparative Capacity of Steam Lines at Various Pitches for Steam and Condensate Flowing in Opposite Directions[a]

(Pitch of Pipe in Inches per 10 Ft. Velocity in Ft per Sec)

Pitch of Pipe..	¼ in.		½ in.		1 in.		1½ in.		2 in.		3 in.		4 in.		5 in.	
Pipe Size Inches	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.
	Capacity Expressed in Pounds per Hour															
¾	6.3	12	7.6	14	9.3	18	10.1	19	10.6	20	11.5	21	11.9	22	12.3	23
1	11.5	12	13.2	15	15.8	17	17.5	20	18.8	22	20.8	23	22.0	25	22.6	26
1¼	26.2	18	29.3	20	33.3	23	36.1	25	38.5	27	41.3	28	43.2	29	44.6	31
1½	35.7	18	39.8	21	45.3	23	49.1	25	52.3	27	56.0	28	58.7	30	60.7	31
2	59.0	19	65.9	20	74.9	23	81.4	25	86.6	27	92.4	28	97.1	29	100.3	30

[a] From The American Society of Heating and Air-Conditioning Engineers Research Laboratory.

The pressure drop in psi per 100 ft of pipe is

$$\Delta P = \frac{\rho \Delta h}{144} \qquad (4)$$

Pipe Sizes

The determination of pipe sizes for a given load in steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and at the end of the return system.
2. The maximum velocity of steam allowable for quiet and dependable operation of the system, taking into consideration the direction of condensate flow.
3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.
4. The direction of flow of the condensate, whether against or with the steam.

Initial Pressure and Pressure Drop

Theoretically, there are several factors to be considered such as initial pressure and pressure required at the end of the line, but it is most important that: (1) the total pressure drop does not exceed the initial gage pressure of the system, and in actual practice it should never exceed one-half of the initial gage pressure; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems specially designed for varying initial pressures, such as the subatmospheric, which normally operate under controlled partial vacuums, and orifice and vapor systems, which at times operate under such partial vacuums as may be obtained due to the condition of the fire; and (4) the rise in water due to pressure drop does not exceed the difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, or the dry-return, and the boiler water line.

Table 2 lists pressure drops in common use with corresponding initial steam pressures for sizing steam piping. It is common practice to limit the total drop in the supply piping to approximately ⅓ of the initial pressure. The allowable pressure per 100 ft is thereby determined by the equivalent length. Designers may utilize total pressure drops up to ½ the initial pressure when steam velocities and operating pressure requirements of the selected equipment will permit. For initial pressures of 30 psig and higher many designers prefer to size steam piping on the velocity method.

The total pressure drop should never exceed one-half of the initial gage pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. ASHAE Research Laboratory experiments limit this to the capacities given in Table 3 for horizontal pipes at various grades.

Maximum Velocity

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensate present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically; (2) the pitch of the pipe if it runs horizontally; (3) the quantity of condensate flowing against the steam; and (4) freedom of the piping from water pockets which under certain conditions act as a restriction in pipe size.

Table 4 .... Length in Feet of Pipe to be Added to Actual Length of Run—Owing to Fittings—To Obtain Equivalent Length

Size of Pipe Inches	Length in Feet to be Added to Run				
	Standard Elbow	Side Outlet Tee[b]	Gate Valve[a]	Globe Valve[a]	Angle Valve[a]
½	1.3	3	0.3	14	7
¾	1.8	4	0.4	18	10
1	2.2	5	0.5	23	12
1¼	3.0	6	0.6	29	15
1½	3.5	7	0.8	34	18
2	4.3	8	1.0	46	22
2½	5.0	11	1.1	54	27
3	6.5	13	1.4	66	34
3½	8	15	1.6	80	40
4	9	18	1.9	92	45
5	11	22	2.2	112	56
6	13	27	2.8	136	67
8	17	35	3.7	180	92
10	21	45	4.6	230	112
12	27	53	5.5	270	132
14	30	63	6.4	310	152

[a] Valve in full open position.
[b] Values given apply only to a tee used to divert the flow in the main to the last riser.

Example of length in feet of pipe to be added to actual length of run.



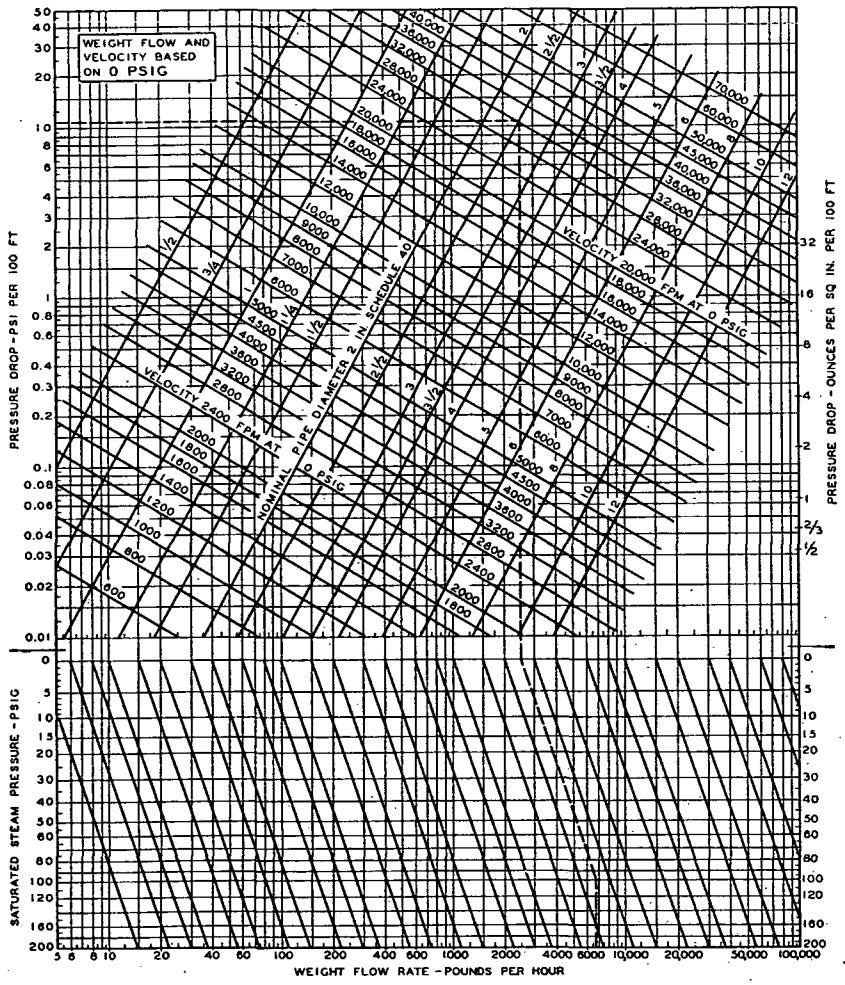
Measured Length = 132.0 ft
4 in. Gate Valve = 1.9 ft
4-4 in. Elbows = 36.0 ft
2-4 in. Tees = 36.0 ft
Equivalent = 205.9 ft

Reaming Important

It is extremely important that the ends of all pipe be reamed or filed. This insures full pipe area and minimizes disturbance of the steam or condensate stream.

Equivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe, as well as for the additional resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to the stated number of feet of straight run of the same size of pipe. Table 4 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter the *length of run* refers to the *equivalent length of run* as distinguished from the *actual length* of pipe



Based on Moody Friction Factor where flow of condensate does not inhibit the flow of steam.

Fig. 22 .... Basic Chart for Weight-Flow Rate and Velocity of Steam in Schedule 40 Pipe Based on Saturation Pressure of 0 Psig

(With Multiplier Charts for Obtaining Weight-Flow Rates and Velocities at All Saturation Pressures Between 0 and 200 Psig)

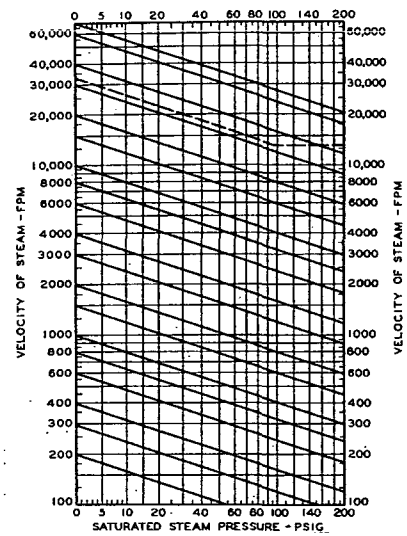


Fig. 22 (*Continued*).... Velocity Multiplier Chart

Example of Use of Basic and Velocity Multiplier Charts.

Given:

a. Weight-Flow Rate = 6700 lb per hr.

b. Initial Steam Pressure = 100 psig.

c. Pressure Drop = 11 psi per 100 ft.

Find:

a. Size of Schedule 40 pipe required.

b. Velocity of steam in pipe.

Solution: The following steps are illustrated by the broken line on Fig. 22:

Step 1. Enter Fig. 22 at a weight-flow rate of 6700 lb per hr and move vertically to the horizontal line at 100 psig.

Step 2. Follow along inclined multiplier line (upward and to the left) to horizontal 0 psig line. The equivalent weight flow at 0 psig is about 2500 lb per hr.

Step 3. Follow the 2500 lb per hr line vertically until it intersects the horizontal line at 11 psi per 100 ft pressure drop. The nominal pipe size is 2½ in. The equivalent steam velocity at 0 psig is about 32,700 fpm.

Step 4. To find the steam velocity at 100 psig, locate the value of 32,700 fpm on the ordinate of the velocity multiplier chart at 0 psig.

Step 5. Move along the inclined multiplier line (downward and to the right) until it intersects the vertical 100 psig pressure line. The velocity as read from the right (or left) scale is about 13,000 fpm.

Note: The preceding Steps 1 to 5 would be rearranged or reversed if different data were given.

Table 5 .... Weight Flow Rate of Steam in Schedule 40 Pipe[a] at Initial Saturation Pressures of 3.5 and 12 Psig.[b, c]

(Weight Flow Rate Expressed in Pounds per Hour)

Nom. Pipe Size Inches	Pressure Drop—Psi Per 100 Ft in Length													
	⅛ Psi (1 oz)		¼ Psi (2 oz)		½ Psi (4 oz)		¾ Psi (8 oz)		1 Psi (12 oz)		1 Psi		2 Psi	
	Sat. press. psig		Sat. press. psig		Sat. press psig.		Sat. press. psig		Sat. press. psig.		Sat. press. psig		Sat. press. psig	
	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12
¾	9	11	14	16	20	24	29	35	36	43	42	50	60	73
1	17	21	26	31	37	46	54	66	68	82	81	95	114	137
1¼	36	45	53	66	78	96	111	138	140	170	162	200	232	280
1½	56	70	84	100	120	147	174	210	218	260	246	304	360	430
2	108	134	162	194	234	285	336	410	420	510	480	590	710	850
2½	174	215	258	310	378	460	540	660	680	820	780	950	1150	1370
3	318	380	465	550	660	810	960	1160	1190	1420	1380	1670	1950	2400
3½	462	550	670	800	990	1218	1410	1700	1740	2100	2000	2420	2950	3450
4	726	800	950	1160	1410	1690	1980	2400	2450	3000	2880	3460	4200	4900
5	1200	1430	1680	2100	2440	3000	3570	4250	4380	5250	5100	6100	7500	8600
6	1920	2300	2820	3350	3960	4850	5700	7000	7200	8600	8400	10,000	11,900	14,200
8	3900	4800	5570	7000	8100	10,000	11,400	14,300	14,500	17,700	16,500	20,500	24,000	29,500
10	7200	8800	10,200	12,600	15,000	18,200	21,000	26,000	26,200	32,000	30,000	37,000	42,700	52,000
12	11,400	13,700	16,500	19,500	23,400	28,400	33,000	40,000	41,000	49,500	48,000	57,500	67,800	81,000

[a] Based on Moody Friction Factor, where flow of condensate does not inhibit the flow of steam.

[b] The weight-flow rates at 3.5 psig can be used to cover sat. press. from 1 to 5 psig, and the rates at 12 psig can be used to cover sat. press. from 8 to 16 psig with an error not exceeding 8 percent.

[c] The steam velocities corresponding to the weight-flow rates given in this table can be found from the basic chart and velocity multiplier chart, Fig. 22.

in feet. The length of run is not usually known at the outset; hence, it may be necessary to assume some pipe size at the start. Such an assumption frequently is considerably in error, and a more common and practical method is to assume the length of run and to check this assumption after the pipes are sized. For this purpose the length of run usually is taken as double the actual length of pipe.

BASIC CHART FOR STEAM PIPE SIZING

Fig. 22 is the basic chart for determining the weight-flow rate and velocity of steam in Schedule 40 pipe for various values of pressure drop per 100 ft, based on zero psig saturated steam. By use of the multiplier charts it may be used at all saturation pressures between 0 and 200 psig.

TABLES FOR PIPE SIZING FOR LOW-PRESSURE

The basic chart, Fig. 22, can be used for sizing pipe on low-pressure systems. The values in Table 5 which are taken from the basic chart provide a more rapid means of selecting pipe sizes for the various pressure drops listed and for systems operated at 3.5 and 12 psig. The weight-flow rates shown for 3.5 psig can be used for saturated pressures from 1 to 6 psig, and those shown for 12 psig can be used for saturated pressures from 8 to 16 psig with an error not exceeding 8 percent.

Both Fig. 22 and Table 5 are for use where the flow of condensate does not inhibit the flow of steam. Columns *B* and *C* of Table 6 are for cases where steam and condensate flow in opposite directions as in risers or runouts that are not dripped. Columns *D*, *E*, and *F* are for one-pipe systems and include risers, radiator valves and vertical connections, and radiator and riser runout sizes, all of which are based on the critical velocities of the steam to permit the counterflow of condensate without noise.

Return piping may be sized with the aid of Table 7, where pipe capacities for wet, dry, and vacuum return lines are shown for several values of pressure drop per 100 ft of equivalent length. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

Example 2: What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft, and the initial pressure is not to be over 2 psig?

Solution: It will be assumed, if the measured length of the longest run is 500 ft, that when the allowance for fittings is added, the equivalent length of run will not exceed 1000 ft. Then, with the pressure drop not over one-half of the initial pressure, the drop could be 1 psi or less. With a pressure drop of 1 psi and a length of run of 1000 ft, the drop per 100 ft would be 0.1 psi, while if the total drop were 0.5 psi, the drop per 100 ft would be 0.05 psi. In both cases the pipe could be sized for a desired capacity according to the 0.1 and 0.05 pressure drop lines in Fig. 22. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is all right; if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used, and the lines resized. Ordinarily, resizing will be unnecessary.

Table 6 .... Steam Pipe Capacities for Low-Pressure Systems

(For Use on One-Pipe Systems or Two-Pipe Systems in which Condensate Flows Against the Steam Flow)

This table is based on data developed through research investigations of the American Society of Heating and Air-Conditioning Engineers

Nominal Pipe Size, Inches	Capacity in Pounds per Hour				
	Two-Pipe Systems		One-Pipe Systems		
	Condensate flowing against steam		Supply risers upfeed	Radiator valves and vertical connections	Radiator and riser runouts
	Vertical	Horizontal			
A	B[a]	C[a]	D[b]	E	F[a]
¾	8	—	6	—	7
1	14	9	11	7	7
1¼	31	19	20	16	16
1½	48	27	38	23	16
2	97	49	72	42	23
2½	159	99	116	—	42
3	282	175	200	—	65
3½	387	288	286	—	119
4	511	425	380	—	186
5	1,050	788	—	—	278
6	1,800	1,400	—	—	545
8	3,750	3,000	—	—	—
10	7,000	5,700	—	—	—
12	11,500	9,500	—	—	—
16	22,000	19,000	—	—	—

Note: Steam at an average pressure of 1 psig is used as a basis of calculating capacities.

[a] Do not use Column B for pressure drops of less than 1/16 psi per 100 ft of equivalent run. Use Fig. 22 or Table 5 instead.

[b] Do not use Column D for pressure drops of less than 1/64 psi per 100 ft of equivalent run except on sizes 3 in. and over. Use Fig. 22 or Table 5 instead.

[c] Pitch of horizontal runouts to risers and radiators should be not less than ½ in. per ft. Where this pitch cannot be obtained, runouts over 8 ft in length should be one pipe size larger than called for in this table.

CHARTS FOR SIZING PIPE FOR HIGH-PRESSURE

Many installations of heating systems for large industrial-type buildings have been designed for the use of high-pressure steam, that is, without the use of pressure-reducing valves. Such systems usually involve the use of unit heaters or large built-up fan units with blast heating coils. Pressures on these systems vary from 30 to 150 psi. Temperatures are controlled by a modulating- or throttling-type thermostatic valve controlled by the air temperature in the room, fan inlet, or fan outlet.

Figs. 23, 24, 25, and 26 provide charts for sizing steam piping for systems of 30, 50, 100, and 150 psig at various pressure drops. These charts are based on the Moody Friction Factor, which takes into account the Reynolds number and the roughness of the internal pipe surfaces, and contain the same information as the basic chart of Fig. 22 but in a more convenient form.

Return pipe capacities given in Tables 8 and 9 are based on the following assumed pressures in the return piping:

Pressure Drop Lb/100 Ft	Pressure in Return Line (Psig)	
	30 psi System	150 psi System
⅛	½	1½
¼	1	2½
½	2	5
¾	3	7½
1	4	10
2	....	20

The pressure loss and line loss as given were substituted in the Babcock steam flow formula, and the resulting capacity was multiplied by 12 for 30-psig systems and by 16 for 150-psig systems to obtain the values given in Tables 8 and 9. The return-line pressures used were found by observation of operating systems. The Babcock formula was used in this case since information on condensate flow in return lines based on the Moody Friction Factor is not readily available.

Table 7 .... Return Main and Riser Capacities for Low-Pressure Systems—Pounds per Hour

(Reference to this table will be made by column letter G through Y)

This table is based on pipe size data developed through the research investigations of The American Society of Heating and Air-Conditioning Engineers

	Pipe Size Inches	1/32 Psi or ½ Oz Drop per 100 Ft			1/64 Psi or ¾ Oz Drop per 100 Ft			1/16 Psi or 1 Oz Drop per 100 Ft			1/8 Psi or 2 Oz Drop per 100 Ft			¼ Psi or 4 Oz Drop per 100 Ft			½ Psi or 8 Oz Drop per 100 Ft		
		Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y
Mains	¾	....	....	....	....	....	42	....	....	100	....	....	142	....	....	200	....	....	283
	1	125	62	....	145	71	143	175	80	175	250	103	249	350	115	350	....	....	494
	1¼	213	130	....	248	149	244	300	168	300	425	217	426	600	241	600	....	....	848
	1½	338	206	....	393	236	388	475	265	475	675	340	674	950	378	950	....	....	1,340
	2	700	470	....	810	535	815	1,000	575	1,000	1,400	740	1,420	2,000	825	2,000	....	....	2,830
	2½	1,180	760	....	1,580	868	1,360	1,680	950	1,680	2,350	1,230	2,380	3,350	1,360	3,350	....	....	4,730
	3	1,880	1,460	....	2,130	1,560	2,180	2,680	1,750	2,680	3,750	2,250	3,800	5,350	2,500	5,350	....	....	7,560
	3½	2,750	1,970	....	3,300	2,200	3,250	4,000	2,500	4,000	5,500	3,230	5,680	8,000	3,580	8,000	....	....	11,300
	4	3,880	2,930	....	4,580	3,350	4,500	5,500	3,750	5,500	7,750	4,830	7,810	11,000	5,380	11,000	....	....	15,500
	5	....	....	....	....	....	7,880	....	....	9,680	....	....	13,700	....	....	19,400	....	....	27,300
	6	....	....	....	....	....	12,600	....	....	15,500	....	....	22,000	....	....	31,000	....	....	43,800
Risers	¾	....	48	....	....	48	143	....	48	175	....	48	249	....	48	350	....	....	494
	1	....	113	....	....	113	244	....	113	300	....	113	426	....	113	600	....	....	848
	1¼	....	248	....	....	248	388	....	248	475	....	248	674	....	248	950	....	....	1,340
	1½	....	375	....	....	375	815	....	375	1,000	....	375	1,420	....	375	2,000	....	....	2,830
	2	....	750	....	....	750	1,360	....	750	1,680	....	750	2,380	....	750	3,350	....	....	4,730
	2½	....	....	....	....	....	2,180	....	....	2,680	....	....	3,800	....	....	5,350	....	....	7,560
	3	....	....	....	....	....	3,250	....	....	4,000	....	....	5,680	....	....	8,000	....	....	11,300
	3½	....	....	....	....	....	4,480	....	....	5,500	....	....	7,810	....	....	11,000	....	....	15,500
	4	....	....	....	....	....	7,880	....	....	9,680	....	....	13,700	....	....	19,400	....	....	27,300
	5	....	....	....	....	....	12,600	....	....	15,500	....	....	22,000	....	....	31,000	....	....	43,800

Table 8 .... Return Pipe Capacities for 30 Psig Steam Systems[a]

Capacity Expressed in Pounds per Hour

Pipe Size Inches	Drop in Pressure—Pounds per 100 Ft in Length				
	⅛	¼	⅜	¾	1
¾	115	170	245	308	365
1	230	340	490	615	730
1¼	485	710	1,025	1,290	1,530
1½	790	1,160	1,670	2,100	2,500
2	1,580	2,360	3,400	4,300	5,050
2½	2,650	3,900	5,600	7,100	8,400
3	4,850	7,100	10,300	12,900	15,300
3½	7,200	10,600	15,300	19,200	22,800
4	10,200	15,000	21,600	27,000	32,300
5	19,000	27,800	40,300	55,500	60,000
6	31,000	45,500	65,500	83,000	98,000

[a] *Note:* The above table is based on steam at pressures of 0 to 4 psig.

Table 9 .... Return Pipe Capacities for 150 Psig Steam Systems[a]

Capacity Expressed in Pounds per Hour

Pipe Size Inches	Drop in Pressure—Psi per 100 Ft in Length					
	⅛	¼	⅜	¾	1	2
¾	156	232	360	465	560	890
1	313	462	690	910	1,120	1,780
1¼	650	960	1,500	1,950	2,330	3,700
1½	1,070	1,580	2,460	3,160	3,800	6,100
2	2,160	3,300	4,950	6,400	7,700	12,300
2½	3,600	5,350	8,200	10,700	12,800	20,400
3	6,500	9,600	15,000	19,500	23,300	37,200
3½	9,600	14,400	22,300	28,700	34,500	55,000
4	13,700	20,500	31,600	40,500	49,200	78,500
5	25,600	38,100	58,500	76,000	91,500	146,000
6	42,000	62,500	96,000	125,000	150,000	238,000

[a] *Note:* The above table is based on steam at pressures of 1 to 20 psig.

SIZING PIPING FOR ONE-PIPE GRAVITY SYSTEMS

Gravity one-pipe air-vent systems, in which the equivalent length of run does not exceed 200 ft, should be sized by means of Tables 5, 6, and 7 and Fig. 22 as follows:

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use 1/16 psi drop (Table 5 or Fig. 22).

2. *Where the riser runouts are not dripped* and the steam and condensate flow in opposite directions, and *also for the radiator runouts* where the same condition occurs, use Table 6, Column *F*.

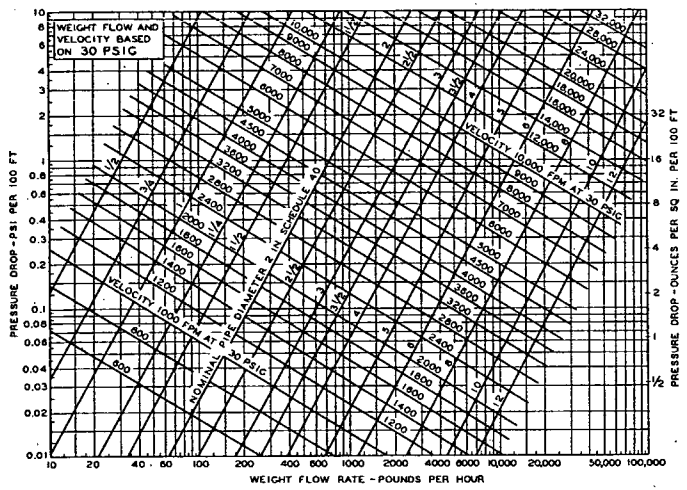
3. *For up-feed steam risers* carrying condensate back from the radiators, use Table 6, Column *D*.

4. *For down-feed systems, the main risers* of which do not carry any radiator condensate, use Table 6, Column *B*.

5. *For the radiator valve size and the stub connection*, use Table 6, Column *E*.

6. *For the dry-return main*, use Table 7, Column *O*.

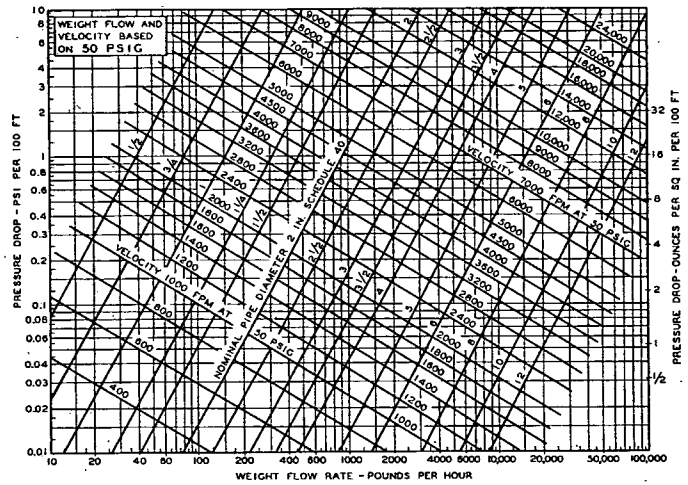
7. *For the wet-return main*, use Table 7, Column *N*.



Based on Moody Friction Factor where flow of condensate does not inhibit the flow of steam.

Fig. 23 .... Chart for Weight-Flow Rate and Velocity of Steam in Schedule 40 Pipe Based on Saturation Pressure of 30 Psig

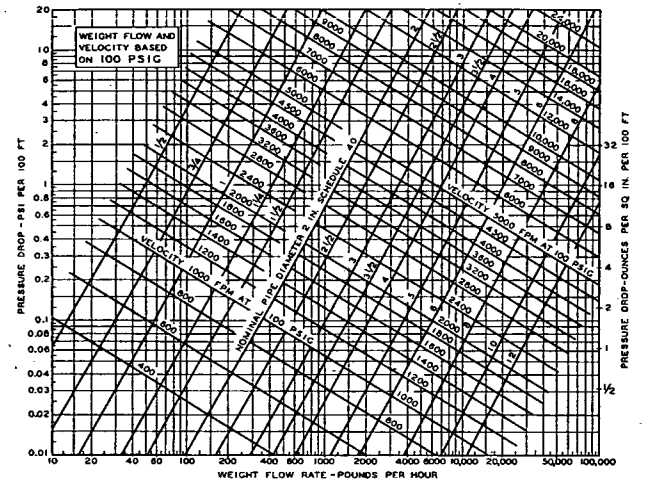
(May be Used for Steam Pressures from 23 to 37 Psig with an Error Not Exceeding 8 Percent)



Based on Moody Friction Factor where flow of condensate does not inhibit the flow of steam.

Fig. 24 .... Chart for Weight-Flow Rate and Velocity of Steam in Schedule 40 Pipe Based on Saturation Pressure of 50 Psig

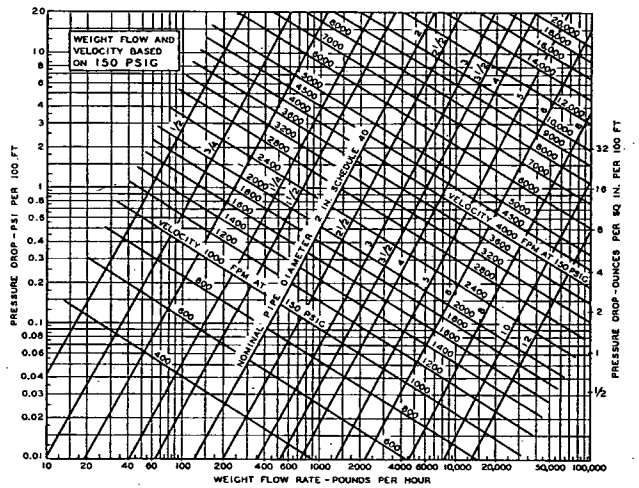
(May Be Used for Steam Pressures from 40 to 60 Psig With an Error Not Exceeding 8 Percent)



Based on Moody Friction Factor where flow of condensate does not inhibit the flow of steam.

Fig. 25 .... Chart for Weight-Flow Rate and Velocity of Steam in Schedule 40 Pipe Based on Saturation Pressure of 100 Psig

(May Be Used for Steam Pressures from 85 to 125 Psig With an Error Not Exceeding 8 Percent)



Based on Moody Friction Factor where flow of condensate does not inhibit the flow of steam.

Fig. 26 .... Chart for Weight-Flow Rate and Velocity of Steam in Schedule 40 Pipe Based on Saturation Pressure of 150 Psig

(May Be Used for Steam Pressures from 127 to 180 Psig With an Error Not Exceeding 8 Percent)

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over ¼ psi. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where $\frac{1}{64}$ psi drop is being used, the steam main and dripped runouts would be sized from Fig. 22; radiator runouts and undripped riser runouts from Table 6, Column *F*; up-feed risers from Column *D*; the main riser on a down-feed system from Fig. 22 (it will be noted that if Column *E* is used the drop would exceed the limit of $\frac{1}{64}$ psi); the dry-return from Table 7, Column *L*; and the wet-return from Column *K*.

With a $\frac{1}{32}$ psi drop the sizing would be the same for as $\frac{1}{64}$ psi, except that the steam main and dripped runouts would be sized from Fig. 22, the main riser on a down-feed system from Fig. 22, the dry-return from Column *I*, and the wet-return from Column *H*.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should not be less than ⅛ in. in 10 ft.
2. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. per foot. Where this pitch cannot be obtained, runouts over 8 ft in length should be one size larger than called for in the table.
3. In general, it is not desirable to have a main less than 2 in. The diameter of the far end of the supply main should not be less than half its diameter at its largest part.
4. Supply mains, runouts to risers, or risers, should be dripped where necessary.
5. Where supply mains are decreased in size they should be dripped, or be provided with eccentric couplings, flush on bottom.

Example 3: Size the one-pipe gravity steam system shown in Fig. 27 assuming that this is all there is to the system, or that the riser and main shown involve the longest run on the system.

Solution: The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of ¼ psi the drop per 100 ft will be slightly less than $\frac{1}{16}$ psi. It would be well in this case to use $\frac{1}{64}$ psi, and this would result in the theoretical sizes

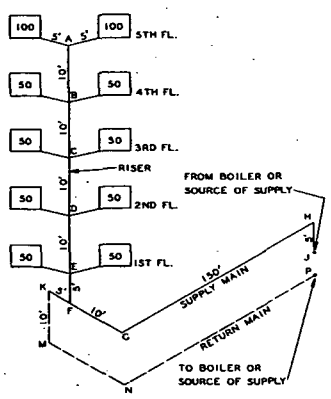


Fig. 27 .... Riser, Supply Main and Return Main of One-Pipe System

Table 10 .... Pipe Sizes for One-Pipe Up-Feed System Shown in Fig. 27

Part of System	Section of Pipe	Radiation Supplied EDR Sq Ft	Theoretical Pipe Size (Inches)	Practical Pipe size (Inches)
Branches to radiators	.....	100	2	2
Branches to radiators	.....	50	1¼	1¼
Riser	*A* to *B*	200	2	2
Riser	*B* to *C*	300	2½	2½
Riser	*C* to *D*	400	2½	2½
Riser	*D* to *E*	500	3	3
Riser	*E* to *F*	600	3	3
Runout to riser	*F* to *G*	600	3½	3½
Supply main	*G* to *H*	600	3	3
Branch to supply main	*H* to *J*	600	2½	3
Dry return main	*F* to *K*	600	1¼	2
Wet return main	*K* to *M*	600	1	2
Wet return main	*M* to *N*	600	1	2
Wet return main	*N* to *P*	600	1	2

indicated in Table 10. These theoretical sizes, however, should be modified by using a wet-return not less than 2 in., while the main supply, *G-H*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *K-M* should be made 2 in. if the wet-return is made 2 in.

SIZING PIPING FOR ONE-PIPE VAPOR SYSTEMS

Piping for one-pipe vapor systems is sized so as to permit only a few ounces pressure drop in the system. Otherwise, the method follows that outlined for sizing one-pipe gravity systems.

SIZING PIPING FOR TWO-PIPE HIGH-PRESSURE SYSTEMS

Steam supply piping for two-pipe high pressure can be sized for greater pressure drops than that of the return piping. For a system using steam at 30 psig, the total pressure drop can be 5 to 10 psi, and for 150 psig systems, 25 to 30 psi.

It has been observed that the maximum total pressure in the returns of a 30 psig system is about 5 psig, and that of a 150 psig system is about 20 psig. The pressure in the return mains is caused by the discharge of traps and flashing of condensate into steam because the return-line pressure is below that corresponding to the saturation temperature of the condensate. The usual practice in the sizing of high-pressure returns has been to size on the basis of ½ psi per 100 ft of pipe for 30 psig systems, and 1 psi per 100 ft for 150 psig systems. This is an average figure which corresponds generally to several of the previously published tables for the design of high-pressure return piping.

Notes on Two-Pipe High-Pressure Systems

Pitch of mains should not be less than ¼ in. in 10 ft.

Pitch of horizontal runouts to risers and heating units should not be less than ½ in. per ft.

SIZING PIPING FOR TWO-PIPE LOW-PRESSURE SYSTEMS

Piping for two-pipe low-pressure systems is sized in the same manner as for two-pipe vapor systems, except that the pressure drop throughout the system can be based on ½ psi to 1 psi drop.

SIZING PIPING FOR TWO-PIPE VAPOR SYSTEMS

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensate return to the boiler by gravity; (2) to obtain a more uniform distribution of steam throughout the system, especially when it is desirable to carry a moderate or low fire; and (3) to prevent large variations in pressure which would nullify the value of graduated valves on radiators.

For small vapor systems when the equivalent length of run does not exceed 200 ft, it is recommended that the main, and any runouts to risers that may be dripped, should be sized for about $\frac{1}{16}$ psi drop per 100 ft, using Fig. 22 or Table 5, while riser runouts not dripped and radiator runouts should be obtained from Column *C*, Table 6. The up-feed steam risers should be taken from Column *B*. For returns use Table 7, Column *O*, the upper portion for mains and the lower portion for risers. It should again be noted that the pressure drop in the steam side of the system is kept the same as on the return side, except where the flow in the riser is concerned.

On a down-feed system, the main vertical riser should be sized from Table 6, Column *B*, but the down-feed risers can be taken from Fig. 22 or Table 5 using a $\frac{1}{16}$ psi drop.

For vapor systems over 200 ft of equivalent length the drop should not exceed ⅛ to ¼ psi if possible. Thus, for a 400-ft equivalent run the drop should be not over $\frac{1}{32}$ psi per 100 ft. In this case the steam mains would be sized from Fig. 22, the radiator and undripped riser runouts from Table 6, Column *C*; and the risers from Fig. 22 because Column *B* gives a drop in excess of $\frac{1}{32}$ psi. On a down-feed system, Fig. 22 would have to be used for both the main riser and the smaller riser feeding the radiators in order not to increase the drop one $\frac{1}{32}$ psi. The return risers would be sized from the upper portion of the same column, while any wet-returns used would be sized from Column *I*. The same pressure drop is applied on both the steam and return sides of the system.

Notes on Vapor Systems

1. Pitch of mains should not be less than ⅛ in. in 10 ft.
2. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. per ft. Where this pitch cannot be obtained, runouts over 8 ft in length should be one size larger than called for in the table.
3. It is not desirable to have a supply main smaller than 2 in.
4. When necessary, supply main, supply risers, or runouts to supply risers should be dripped separately into a wet-return, or may be connected into the dry-return through a thermostatic drip trap.

SIZING PIPING FOR TWO-PIPE VACUUM SYSTEMS

Vacuum, atmospheric, subatmospheric, and orifice systems are usually employed in large installations and have total drops varying from ¼ to 2 psi. Systems in which the maximum equivalent length does not exceed 200 ft preferably employ the smaller pressure drop, while systems over 200-ft equivalent length of run, more frequently are designed for the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 2400-ft longest equivalent length of run could be designed for a pressure drop per 100 ft of 1 psi divided by 12, or $\frac{1}{64}$ psi. In this case the steam main would be sized from Fig. 22, and the risers also from Fig. 22 (Column *B*, Table 6, could be used as far as critical velocity is concerned, but the drop would exceed the limit of $\frac{1}{64}$ psi). Riser runouts, if dripped, would also be obtained from Fig. 22 but if undripped, from Column *C*, Table 6. Pipe sizes for other parts would be obtained as follows: radiator runouts from Column *C*; return risers from lower part of Column *S*, Table 7; and return runouts to radiators—one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. Pitch of mains should not be less than ¼ in. in 10 ft.
2. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. per ft. Where this pitch cannot be obtained, runouts over 8 ft in length should be one size larger than called for in the table.
3. In general, it is not considered desirable to have a supply main smaller than 2 in.
4. When necessary, the supply main, supply riser, or runout to a supply riser should be dripped separately through a trap into the vacuum return. A connection should not be made between the steam and return sides of a vacuum system without interposing a trap to prevent the steam from entering the return line.
5. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in this chapter.
6. No lifts can be used in orifice and atmospheric systems.

SIZING PIPING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized according to the quantity of steam condensed by each unit. The condensate per unit depends upon the entering temperature and the air velocity, and may be obtained from manufacturers' rating tables. Where two or more units are placed in series, the entering air temperature for any unit will be the leaving temperature for the preceding unit.

When the amount of condensate has been obtained for each unit, the pipe sizes should be based on the length of run and the pressure drop desired, as in the case of radiators. It is generally desirable to place the indirect heating units on a separate piping system rather than to connect them to the piping which supplies direct radiation. For type of connections see section on Connections to Heating Units.

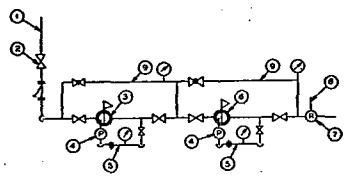
PRESSURE-REDUCING VALVES

While the illustrations of steam systems in this chapter indicate that they are supplied by separate boiler plants, it is also possible to have steam supplied by a remote boiler plant or by a district heating system at pressures higher than desired for the equipment served. District heating is discussed in Chapter 27.

There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from district steam mains. Some of these are peculiar to district heating and some, although they may apply to any building, have been demonstrated to be particularly important when district steam is used. District heating companies have certain regulations regarding the consumer's installation to safeguard the mutual interest of supplier and consumer and to insure satisfactory and economical service. When an installation is designed, local regulations and code requirements should be observed.

Where steam is supplied from a boiler delivering steam at pressures higher than required for the heating system, one or more pressure-reducing valves (pressure regulators) are required. These are used in two classes of service, one

where the steam must be shut off tight to prevent build-up of pressure on the low-pressure side at time of no load, and the other where the low-pressure lines will condense enough steam to prevent build-up of pressure due to normal leakage through the valve. In the first case single-seated valves, either direct-operated or pilot-controlled, should be used and in the latter case double-seated valves, either direct-operated or pilot-controlled, may be used. Valves available are direct-operated, spring, weight, air-loaded, or pilot-controlled, using either the flowing steam or auxiliary air or water as the operating medium. The direct-operated, double-seated valve is less affected by varying inlet steam pressure than the direct-operated, single-seated valve. Pilot-controlled valves, either single- or double-seated, tend to eliminate the effect of variable inlet pressures.



1. Service pipe from district steam main
2. Service valve
3. Pressure-reducing valve
4. Pilot
5. Balance line
6. Pressure-regulating valve
7. Safety relief valve
8. Relief valve discharge to outside
9. Bypass, one size smaller than pressure-reducing valve

Note: All fittings should be American Standard Class 250 cast iron and/or 300 lb steel.

(Used Where High-Pressure Steam is Supplied for Low-Pressure Requirements)

Fig. 28 .... Two-Stage Pressure-Reducing and Pressure-Regulating Equipment

Valves that shut off all steam are called *dead end* type. They are single-seated, and some of them have pilot operation or constant air loaders that provide close control of the reduced pressure. If a thermostatically controlled valve is installed after, and near, a reducing valve in such a manner as to cut off the passage of steam, the dead end type of reducing valve should be used and a suitable trap should be installed to provide for drainage of condensate formed between the reducing valve and the control valve. If a one-stage reduction is desired, it is necessary to use either a pilot-controlled or an air-loaded direct-operated pressure-reducing valve where low pressures are to be maintained closely.

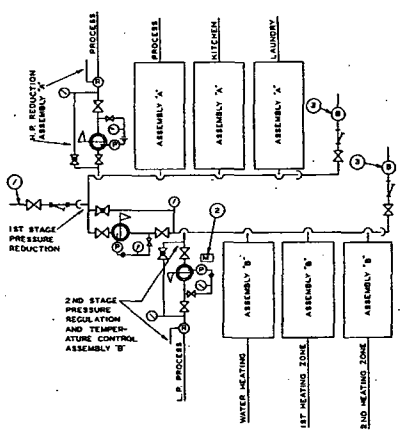
Installation

The installation of pressure-reducing valves in pipe lines requires detailed planning. They should be installed to give ease of access for inspection and repair. There should be a bypass around each reducing valve of area equal to the area of the reducing-valve seat ring. The globe valve in a bypass line should be of a good type of construction, and must shut off absolutely tight. A steam pressure gage, graduated up to the initial pressure, should be installed on the low-pressure side. This should be ahead of the shut-off valve because the reducing valve can be more accurately and conveniently adjusted with the shut-off valve closed. A similar gage should be installed downstream from the shut-off valve for use during manual operation.

Strainers should be installed on the inlet of the primary pressure-reducing valve and are also desirable before the second-stage reduction if there is considerable piping between the two stages. If a two-stage reduction is made, it is well to install a pressure gage immediately before the reducing valve of the second-stage reduction for use in setting and checking the operation of the first valve. In all cases it is advisable to install a drip trap between the two reducing valves.

Safety valves located on the low-pressure side should be set at least 5 psi higher than the reduced pressure where the reduced pressure is under 35 psig and at least 10 psi higher than the reduced pressure if the reduced pressure is above 35 psig or is the first-stage reduction of a double reduction. A safety valve or valves, installed on the low-pressure side of a reducing valve or valves, should be of sufficient capacity and should be set to relieve the entire output of the reducing valves in the event pressure on the equipment exceeds the low-pressure setting by the values indicated above. The outlet from relief valves should be piped to a location where the discharge to atmosphere cannot jeopardize persons or property.[8] (See Figs. 28 and 29.)

Typical service connections are shown in Fig. 30 for low-



1. Service pipe from district steam main
2. Motorized pilot on pressure-regulating valve
3. Bucket trap on header drip line

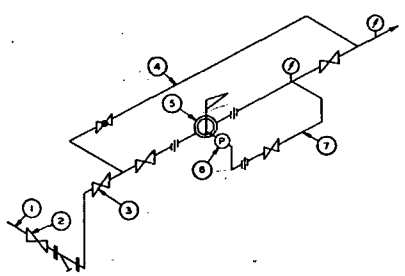
Note: Where process equipment is individually controlled, temperature control on header may be omitted. All fittings should be American Standard Class 250 cast iron or 300 lb steel in both reduction and regulation assemblies.

Fig. 29 .... Steam Supply Installation with Header for Process and Zone Connections

pressure service and Fig. 31 for high-pressure service. The district heating industry refers to valves according to their functional use. The pressure-reducing valve reduces pressure to a safe point, and the pressure-regulating valve regulates pressure to that required by equipment served. Fig. 31 illustrates the use of a pressure-regulating valve with a bypass. This bypass permits the operation of the system in case of failure in the pressure-regulating valve. In the smaller sizes, the bypass is sometimes omitted. When servicing is required, the pressure-regulating valve can be removed, a filler installed, and the inlet stop valves used for manual regulation of the pressure until repairs are made. Fig. 29 shows a typical service installation with a separate line to the various heating zones and process equipment. If the initial pressure is below 50 psig the first stage pressure-reducing valve may be omitted. In Assembly *A*, Fig. 29, the single-stage pressure-reducing valve is also the pressure-regulating valve.

If the radiation served is of cast iron and the inlet pressure exceeds 50 psig, the ASME Code[9] requires the installation of pressure-relief valves where the inlet pressure exceeds the safe working pressure of the steam-using equipment, except where two reducing valves are installed in series and *both* are set at or below the safe working pressure of the equipment serviced. This, however, is not two-stage reduction. See Fig. 31.

Two reducing valves, installed in series and both set at or below the safe working of the equipment serviced, introduce a source of safety while taking advantage of the more stable and satisfactory control provided by a single-step reduction. As can be seen from Fig. 31, the second valve which is set for a slightly higher reduced pressure is normally wide open and only comes into play to throttle the flow of high-pressure steam resulting from any failure of the primary valve. The increase of the system pressure to the higher pressure of the second valve is a signal that the primary valve requires maintenance. With bypasses as illustrated,



1. Service pipe from district steam main
2. Service valve
3. Operating valve
4. Manual bypass
5. Pressure-regulating valve
6. Pilot
7. Balance or impulse line. Follow manufacturer's recommendation.

Fig. 30 .... Pressure-Regulating Valve with Bypass for Low Pressure Service

the primary valve can be serviced without shutting down the system. Since the valves are not both operating at the same time the distance between the valves is not critical.

When making a two-stage reduction, e.g., 150 to 50 psig and then 50 to 2 psig, allowance for expansion of steam on the low-pressure side of each reducing valve should be made by increasing the pipe area to approximately double the area of a pipe of the size of the reducing valve. This also allows steam flow to be at a more nearly uniform velocity. Separating the valves by a distance of at least 20 ft is recommended to reduce excessive hunting action of the first valve.

Fig. 28 shows a typical double-reduction installation where the pressure in the district steam main is higher than can safely be applied to building heating systems. The first or

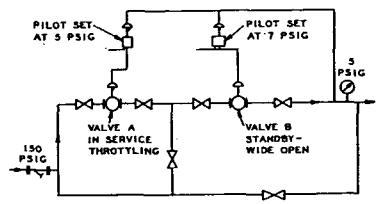


Fig. 31 .... Single-Stage Pressure Reduction—Series Installation

pressure-reducing valve effects the initial pressure reduction. The pressure-regulating valve regulates the steam to the desired final pressure.

Pilot-controlled or air-loaded direct-operated reducing valves may be used without limitation for all reduced pressure settings for all heating, process, laundry, hot water, etc. services. Spring-loaded direct-operated valves may be used for reduced pressures up to 50 psig providing they can pass the required steam flow without excessive deviation in reduced pressure.

Weight-loaded valves are used for reduced pressures below 15 psig and for moderate steam flows.

Pressure-equalizing or impulse lines must be properly connected to serve the type of valve selected. For direct-operated diaphragm valves with rubber-like diaphragms, the impulse line should be connected into the bottom of the reduced-pressure steam line to allow maximum condensate to exist on the diaphragm and in the impulse or equalizing line or, if connected to the top of the steam line, a condensate accumulator should be used to reduce variations in the head of condensate on the diaphragm. Equalizing or impulse lines for pilot-controlled and direct-operated reducing valves employing metal diaphragms should be connected into the expanded outlet piping approximately 2 to 4 ft from the reducing valve and pitched away from the reducing valve to prevent accumulation of condensate. Pressure impulse lines for externally pilot-controlled reducing valves employing compressed air or fresh water should be installed in accordance with manufacturer's recommendations.

Selection of Valve Size

Pressure-regulating valves should be sized to supply adequately the maximum steam requirements of the heating

Table 11.... Suggested Capacity of Pressure-Regulating Valves and Traps

Type of Control	Pressure-Regulating Valve Capacity Lb/(Hr)(Sq Ft EDR[a])
Continuous or modulating operation, intermittent with *on* periods over 3 hr in duration	⅓
Intermittent operation with *on* periods of ½ to 3 hr in duration	½
Intermittent operation with *on* periods of ½ hr or less	½[b]

[a] One square foot EDR = 240 Btu per hr.

[b] Except for 1-pipe systems or 2-pipe systems not orificed which should be 1 lb per sq ft EDR.

system or equipment. Consideration should be given to rangeability, speed of load changes, and accuracy of regulation required to properly handle system needs. This is especially important with temperature control systems using intermittent flow of steam to heat the building.

Certain types of pilot-operated pressure-regulating valves can be used as a temperature control valve in some temperature control applications, by actuating the pilot by means of a damper operator or small automatic control valve.[4]

Pressure-regulating valve capacities recommended for use with various types of temperature control are shown in Table 11.

Care should be exercised in selecting the size of a reducing valve. The safest method is to consult the manufacturer. It is essential that sizes of piping to and from the reducing valve be such that they will pass the desired amount of steam with the maximum velocity desired. A common error is to make the size of the reducing valve the same size as that of the service, or outlet pipe size. Generally, this will make the reducing valve oversized and cause wiredrawing of valve and seat due to the resultant small lift of the valve.

On installations where the steam requirements are relatively large and variable, wiredrawing and cycling control may occur in mild weather or during reduced-demand periods. To overcome this condition, two reducing valves are installed in parallel, with the sizes selected on a 70 and 30 percent proportion of maximum flow. For example, if 10,000 lb of steam per hour are required, the size of one valve is on the basis of 7000 lb, and the other on the basis of 3000 lb. During mild weather (spring and fall) the larger valve is set for slightly lower reduced pressure than the smaller one and hence remains closed as long as the smaller can supply the demand. During the remainder of the heating season the valve settings are reversed to keep the smaller one closed except when the larger is unable to supply the demand.

BOILER CONNECTIONS

Steam

Cast-iron, sectional heating boilers usually have several outlets in the top. Two or more outlets should be used whenever possible to reduce the velocity of the steam in the vertical uptakes from the boiler, and thus to prevent carrying of water into the steam main. Piping connections to the steam header are shown in Fig. 32.

Return

Cast-iron boilers are generally provided with return tappings on both sides, while steel boilers are generally equipped with only one return tapping. Where two tappings are provided, both should be used to effect proper circulation through the boiler. Recommended piping connections including the Hartford return for a battery of steam boilers having the *condensate returned by a pump*, are shown in Fig. 32. Connections for a single boiler are shown in Fig. 33. The 90-deg elbow, short horizontal nipple, and tee at the top of the Hartford connection may be replaced by an inverted Y fitting if the condensate flows to the boiler by *gravity*. The point at which the condensate enters the equalizer pipe should be 4 in. below the normal water line when a pump is used and usually 2 to 4 in. for gravity return.

The inverted Y fitting should *not be used* if condensate is *returned by a pump* because the fitting may direct the condensate upward into the equalizer pipe where it will cause noise due to rapid condensation of some steam. Proper selection of a pump capacity and discharge pressure is required in any case because a pump may deliver water at such a high rate that steam and cold condensate are brought into contact and cause noise.

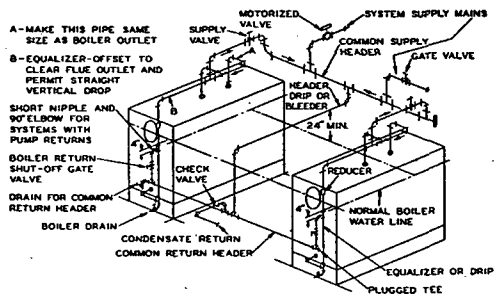


Fig. 32.... Piping Including Hartford Return Connection for Boilers in Battery

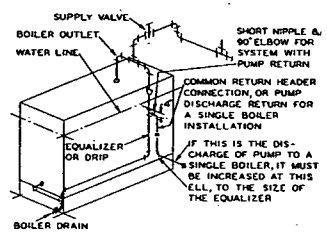


Fig. 33.... Boiler Piping Including Hartford Return Connections

Sizing Boiler Connections

Little information is available on the sizing of boiler runouts and steam headers. Although some engineers prefer an enlarged steam header to serve as additional steam storage space, there ordinarily is no sudden demand for steam in a steam heating system, except during the heating-up period, at which time a large steam header is a disadvantage rather than an advantage. The boiler header may be sized by first computing the maximum load that must be carried by any portion of the header under any conceivable method of operation, and then applying the same schedule of pipe sizing to the header as is used on the steam mains for the building. The horizontal runouts from the boiler, or boilers, may be sized by calculating the heaviest load that will be placed on the boiler at any time, and sizing the runouts on the same basis as the building mains. The difference in size between the vertical uptakes from the boiler, which should be of same size as the boiler outlet tapping, and the horizontal main or runout, is compensated for by the use of reducing ells.

Return connections to boilers in gravity systems are made the same size as the return main itself. Where the return is split and connected to two tappings on the same boiler, both connections are made the full size of the return line. Where two or more boilers are in use, the return to each may be sized to carry the full amount of return for the maximum load which that boiler will be required to carry. Where two boilers are used, one of them being a spare, the full size of the return main would be carried to each boiler, but if three boilers are installed, with one spare, the return line to each boiler would require only half of the capacity of the entire system, or, if the boiler capacity were more than one-half the entire system load, the return would be sized on the basis of the maximum boiler capacity. As the return piping around the boiler is usually small and short, it should not be sized to the minimum.

With returns pumped from a vacuum or receiver return pump, the size of the line may be calculated from the water rate on the pump discharge when it is operated, and the line sized for a very small pressure drop. The relative boiler loads should be considered, as in the case of gravity return connections. Boiler header and piping sizes should be based on the total load.

CONDENSATE RETURN PUMPS

Condensate return pumps are used for gravity systems when the local conditions do not permit the condensate to return to the boiler under the existing static head. The return of the condensate permits the water to pass repeatedly through the cycle of vaporization, with subsequent condensation and return to the boiler. During such repeated cycles any incrustants or other substances in solution are precipitated and the water de-activated to a considerable extent, so that corrosion of a serious nature is seldom ever encountered where the condensate is repeatedly used. Serious corrosion is more frequently found in systems in which the condensate is wasted, and fresh make-up water is continually being introduced.

A generally accepted condensate pump unit for low-pressure heating systems consists of a motor-driven centrifugal pump with receiver and automatic float control. Other types in use include rotary, screw, turbine and reciprocating pumps with steam turbine or motor drive, and direct-acting steam reciprocating pumps.

The receiver capacities of these automatic units should be sized so as not to cause too great a fluctuation of the boiler water line if fed directly to the boiler, and at the same time not so small as to cause too frequent operation of the unit. The usual unit provides storage capacity between stops in the receiver of approximately 1.5 times the amount of condensate returned per minute, and the pump generally has a delivery rate of 2 to 3 times the normal flow. This relation of receiver and pump size to heating system condensing capacity takes account of the peak condensation rate.

A typical installation of a motor-driven automatic condensate unit is illustrated in Fig. 10.

VACUUM HEATING PUMPS

On vacuum systems, where the returns are under a vacuum, and subatmospheric systems, where the supply piping, radiation, and the returns are under a vacuum, it is necessary to use a vacuum pump to discharge the air and noncondensable gases to the atmosphere and to dispose of the condensate. Direct-acting steam-driven reciprocating vacuum pumps are sometimes used where high-pressure steam is available, or where the exhaust steam from the pump can be utilized. In general, however, these have been replaced by the automatic motor-driven return-line heating pump especially developed for this service. Steam-turbine drive is also frequently used where steam at suitable pressures is available, the steam being used afterward for building heating. The usual vacuum pump unit consists of a compact assembly of exhausting unit for withdrawing the air-vapor mixture and discharging the air to atmosphere, and a water removal unit which discharges the condensate to the boiler. They are furnished complete with receiver, separating tank, and automatic controls mounted as an integrated unit on one base. There are also special steam-turbine driven units which are operated by passing the steam to be used in heating the building through the turbine with only a 2 to 3 psi drop across the turbine required for its operation. Under special conditions such as installations where it is necessary to return the condensate to a high-pressure boiler, auxiliary water pumps may be supplied. In some instances separate air and water pumps may be used.

For rating purposes[4] vacuum pumps are classified as *low*

vacuum and *high vacuum*. Low-vacuum pumps are those rated for maintaining 5½ in. Hg vacuum on the system, and high-vacuum pumps are those rated to maintain vacuums above 5½ in. Hg.

The required air and condensate removal capacity of vacuum pumps depends upon the size and type of heating system to which they are applied.

The required condensate capacity is closely related to the amount of radiation installed together with the range of the heating requirement. For average design conditions, the theoretical condensing rate of a heating system is approximately 0.5 gpm per 1000 sq ft equivalent direct radiation (EDR) served. Vacuum-pump water-delivery rates generally should be the same as for condensate pumps, which range from 2 to 3 times the design condensing rate. This provides sufficient capacity for peak loads, and for intermittent automatic operation.

The wide varying operating range and characteristics of individual heating systems have a greater effect on the quantity of air to be removed than on the condensate to be handled. The air-handling requirement cannot be found by applying a common factor to the equivalent direct radiation design load although this would be possible in determining the condensate. For low-vacuum systems, where the returns are controlled at an average vacuum of 5.5-in. Hg vacuum and the temperature averages 160 F, the various manufacturers of vacuum pumps provide pump capacities from 0.3 to 1.0 cfm of air per 1000 sq ft equivalent direct radiation served. For high-vacuum systems, the vacuum-pump air capacity requirements are generally greater. A capacity of 2.0 cfm of air, at 20-in. Hg vacuum per 1000 sq ft equivalent direct radiation served, is a typical capacity *furnished on high-vacuum systems*. These higher air capacities on subatmospheric controlled systems are required basically because of the increase in vacuum which expands the volume of air and vapor that is to be removed to maintain circulation at the higher operating vacuum.

It is particularly important on high-vacuum installations to see that the entire system is tight in order to reduce the amount of inward air leakage and, furthermore, to assure that relatively higher temperature steam is prevented from entering the vacuum return lines through leaky traps, high-pressure drips, etc. The hotter the returns, the lower will be the possible vacuum for a given air leakage into the system. It is for this reason that the condensate from equipment using steam at high pressure should not be connected directly to a vacuum return line, but should drain to a flash tank or flash leg through a high-pressure trap. The receiver should have an equalizing connection to a low-pressure steam main and drain through a low-pressure trap to the vacuum return main, as indicated later in this chapter in section on Drips.

Vacuum-Pump Controls

*In the ordinary vacuum system, the vacuum pump is con*trolled by a vacuum regulator which cuts in when the vacuum drops to the lowest point desired, and cuts out when it has been increased to the highest point, these points being varied to suit the particular system or operating conditions. In addition to this vacuum control, a float control is included which will start the pump whenever sufficient condensate accumulates in the receiver, regardless of the vacuum on the system. A selector switch is usually provided to allow operation at night as a condensate pump only, also to give manual or continuous operation when desired.

There are several variations in the control of the vacuum maintained on the system by the pump. In some subatmospheric systems where orifices are used, the vacuum-pump control maintains a pressure difference between the supply and the return piping, which is held within relatively close limits. There are other subatmospheric systems which utilize special temperature-pressure actuated controls for maintaining the desired conditions in the return lines. Where various zones are connected to the same return main, the return vacuum must be controlled to meet the requirements of the zone operating at the lowest steam supply pressure.

Piston-Displacement Vacuum Pumps

Piston-displacement return vacuum heating pumps may be either electric or steam driven. Their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. They are usually supplied with an air separating tank, open to the atmosphere, placed on the discharge side of the pump, and at an elevation sufficiently high to allow gravity flow of the condensate to the boiler. If the boiler pressure is too high for such gravity feed, then an additional steam pump for feeding the boiler is desirable. The extra pump is sometimes avoided by using a closed separating tank with a float-controlled vent. In both arrangements, the air taken from the system must be discharged against the full discharge pressure of the vacuum pump. In the case of high- or medium-pressure boilers, it is better to use the atmospheric separator and the second pump.

In figuring the required displacement for such pumps, a value of from 6 to 10 times the volumetric flow of condensate is used for average vacuums and systems.

STEAM TRAPS

Steam traps, as the name implies, are automatic devices used to trap or hold steam in an apparatus or piping system until it has given up its latent heat, and to allow condensate and air to pass as soon as it accumulates. In general, traps consist of a vessel in which to accumulate the condensate, an orifice through which the condensate is discharged, a valve to close the orifice port, mechanisms to operate the valve, and inlet and outlet openings for the entrance and discharge of the condensate from the trap vessel.

Steam traps are classified according to the type of operating device by which they function. The traps which are available on the market today may be classified as (1) float, (2) thermostatic, (3) float and thermostatic, (4) upright bucket, (5) inverted bucket, (6) flash, (7) impulse, (8) lifting, and (9) boiler return trap or alternating receiver.

Float Traps. Float traps operate by the rise and fall of a float due to a change of condensate level in the trap. When the trap is empty, the float is in its lowest position and the discharge valve is closed. As condensate accumulates in the trap chamber, the float rises and gradually opens the valve, and the pressure of the steam pushes the condensate out of the valve. The discharge from a float trap is generally continuous, since the opening of the valve is proportioned to the flow of condensate through the trap. A gage glass may be used to indicate the height of condensate in the trap chamber.

Unless float traps are well made and proportioned, there is danger of considerable steam leakage through the discharge valve due to unequal expansion of the valve and seat, and the sticking of moving parts. Float traps are made in sizes from ½ to 3 in., and for pressures varying from vacuum conditions to 200 psig. Float traps are used for draining condensate from steam separators, steam headers, blast coils, heating systems, steam-water heaters, laundry equipment, sterilizers, and other equipment. When used for draining low-pressure systems, float traps should be equipped with a thermostatic air vent (see Float and Thermostatic Traps). Fig. 34 illustrates a single-port float trap.

Thermostatic Traps. Thermostatic traps function by means of elements which expand and contract under the influence of heat and cold. In early types of thermostatic traps, carbon posts and bimetallic elements were used for expansion. In general, the modern type of thermostatic trap consists of thin corrugated metal bellows or discs enclosing a hollow chamber. The chamber is either filled with a liquid, or a small amount of a volatile liquid, such as alcohol, is introduced. The liquid expands or becomes a gas when steam comes into contact with the expansive element. The pressure created in either case expands the element and closes the valve of the trap against the escape of steam. When condensate or air comes in contact with the element, it cools and contracts, opening the valve and allowing them to escape.

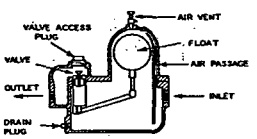


Fig. 34 .... Single Port Float Trap

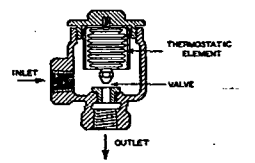


Fig. 35 .... Thermostatic Trap Bellows Type

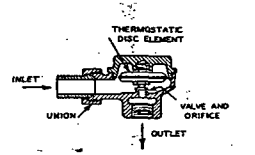


Fig. 36 .... Thermostic Trap Disc Typeat

The discharge from this type of trap is intermittent. Thermostatic traps find their use generally for the draining of condensate from radiators, convectors, pipe coils, drips, unit heaters, water heaters, cooking kettles, and other equipment. Except for radiators and convectors, it is recommended that a strainer be installed on the inlet connection to the trap to prevent dirt, pipe scale, and other foreign substance from entering the trap. A cooling leg of a length of pipe should also be provided ahead of the trap on unit heaters and similar apparatus to cool the condensate in order to help in the trap action. Thermostatic traps are made in sizes from ½ to 2 in., and for pressures ranging from vacuum conditions to 200 psig. Figs. 35 and 36 illustrate types of thermostatic traps which are available at the present time.

Float and Thermostatic Traps. This type of trap is a combination of the float trap and the thermostatic trap, and finds its use in the draining of condensate from unit heaters, blast heaters, and coil heaters for water, oil, or other liquids where there is apt to be a large volume of condensate which would not permit successful operation of thermostatic traps alone. The function of the float element of this trap is to handle the condensate, and of the thermostatic element to permit the flow of air and to prevent the flow of steam around the float valve. Float and thermostatic traps are made in sizes from ½ to 2 in. and usually operate under pressures varying from vacuum conditions to 40 psig, although some are made to operate at a maximum pressure of 200 psig. Fig. 37 illustrates a typical float and thermostatic trap.

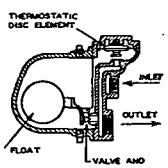


Fig. 37 .... Typical Float and Thermostatic Trap

Upright Bucket Traps. In this type of trap, the condensate enters the trap chamber and fills the space between the bucket and the walls of the trap. This causes the bucket to float, and *forces the valve against its seat, the valve and its stem usually* being fastened to the bucket. When the condensate in the chamber rises above the edges of the bucket, it overflows into it and causes the bucket to sink, thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of condensate in the bucket to force the water to the discharge opening. When the bucket is emptied, it rises and closes the valve and another cycle begins. The discharge from this type of trap is intermittent, and it requires a definite differential pressure (usually 1 psi at least) between the inlet and outlet of the trap in order to lift the condensate out of the bucket to the return opening.

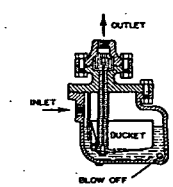


Fig. 38 .... Upright Bucket Trap

Upright bucket traps are used for draining condensate and air from blast coils, unit heaters, steam mains, laundry equipment, sterilizers, water and oil heaters and other equipment. This type of trap is particularly suited for use where there are pulsating pressures, such as occur in draining steam lines and separators at reciprocating pumps or engines. It is not influenced by pulsations or wide fluctuations of pressure. Upright bucket traps are obtainable in sizes varying from ½ to 2½ in., and for pressures varying from vacuum to 1200 psig. Fig. 38 illustrates an upright bucket trap.

Inverted Bucket Traps. In this type of trap, steam, condensate, and air enter the trap under the bell or inverted bucket. Steam floats the inverted bucket and closes the valve. Condensate entering the trap enables the inverted bucket to fall, opening the valve. The condensate then discharges through the open valve until steam again enters and displaces the water contained in the bucket, thus restoring its buoyancy. The steam pressure entering through the open valve discharges the trap. Air is eliminated automatically by passing through the small vent hole located in the top of the inverted bucket.

Inverted bucket traps for use on low-pressure systems, particularly with blast coils or unit heaters, are usually furnished with a large capacity opening equipped with a bimetallic thermostatic element which closes when heated by steam, and opens when cooled by air and condensate, allowing air to escape from the inverted bucket to the trap outlet. Inverted

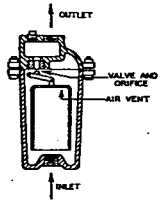


Fig. 39 .... Inverted Bucket Trap

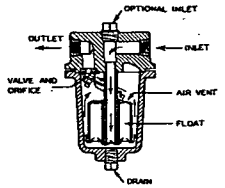


Fig. 40 .... Inverted Bucket Trap with Central Guide

bucket traps are used for draining condensate and air from blast coils, unit heaters, steam drips, laundry equipment, sterilizers, steam water heaters, and other equipment. They are particularly suited for draining condensate from steam lines or equipment where abnormal amounts of air must be discharged, and where there is also foreign matter such as dirt, sludge, and oil draining to the trap.

The discharge from inverted bucket traps, like that of the upright bucket traps, is intermittent and requires a definite differential pressure between the inlet and the outlet of the trap in order to lift the condensate from the bottom of the trap to the outlet of same. Inverted bucket traps are made in sizes from ½ to 3 in., and for pressures varying from vacuum to 2400 psig. Figs. 39 and 40 illustrate some of the types of inverted bucket traps which are available on the market at the present time.

Flash Traps. These traps depend on the property of condensate at a high pressure and temperature to flash into steam at a lower pressure. Condensate flows freely through the orifice of the trap due to the pressure difference from inlet to outlet of trap until steam enters the inlet chamber and mixes with the remaining condensate, heating the condensate and causing it to flash, thereby choking the flow through the orifice and allowing more condensate to accumulate in the trap. The discharge from flash-type traps is intermittent. There are no moving parts in this type of trap. The orifice, however, is adjustable for the pressure differential required. A gage glass or float indicates whether the trap is operating.

These traps can be used for draining condensate from steam, water, and oil heaters; blast heaters; unit heaters; dryers; vulcanizers; kitchen equipment; laundry equipment; evaporators; steam lines; and other equipment, where the pressure differential between steam supply and condensate return does not drop below 5 psi. Flash-type traps are made in sizes from ½ to 3 in., and for pressures varying from vacuum to 450 psig. Fig. 41 illustrates a trap of the flash type.

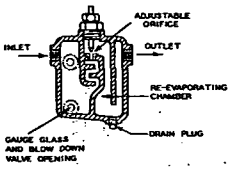


Fig. 41 .... Flash Trap

Impulse Traps. These traps are a modification of the flash trap, and depend on the same principle of flash for their operation. In the impulse trap the flashing action is utilized to govern the movement of a valve by causing changes in pressure in a control chamber above the valve. A small portion of condensate, called control flow, bypasses continuously through the control chamber. At low and medium temperatures, the discharge through the center orifice reduces control chamber pressure, and the valve opens for free discharge of air and condensate.

When condensate reaches near-steam temperature, part of the control flow flashes into vapor, due to reduced pressure in control chamber. The increased volume of the condensate-vapor mixture restricts the discharge through the center orifice, and therefore the reduced pressure in control chamber builds

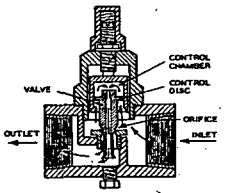


Fig. 42 .... Impulse Trap

up, closes the valve, and shuts off all discharge of hot condensate, except the small amount flowing through center orifice.

Under normal condensate loads, the valve opens and closes at short intervals. Under heavy loads, the valve opens wide and the discharge is heavy and continuous. Impulse traps can be used for draining condensate from steam mains, unit heaters, laundry equipment, kitchen equipment, oil and water heaters, sterilizers, and other equipment where the pressure at the trap outlet is not greater than 25 percent of the inlet pressure. Impulse traps are made in sizes from ½ to 2 in., and for pressures ranging from 1 to 600 psig. Fig. 42 illustrates a trap of the impulse type.

Lifting Traps. This type of trap is an adaptation of the up-

right bucket trap. It has the added feature of an auxiliary pressure inlet through which steam is introduced at a pressure higher than that of the trap inlet pressure. This high-pressure steam forces the condensate to a point above the trap, and against a back pressure higher than that which is possible with normal steam pressure. Lifting traps are made in sizes from 1 to 3 in., and for pressures ranging from vacuum to 150 psig. Fig. 43 illustrates a trap of the lifting type.

Boiler Return Trap or Alternating Receiver. This device is not actually a steam trap in that it is not used to trap or hold steam, but is an adaptation of the lifting trap. It is used for returning condensate to a low-pressure boiler, when due to excess pressure, the condensate cannot flow to the boiler by gravity without flooding the return mains, and endangering the boiler by permitting it to go dry. The boiler return trap is a vessel (vented to the atmosphere) in which condensate collects until a float-operated mechanism equalizes the boiler and

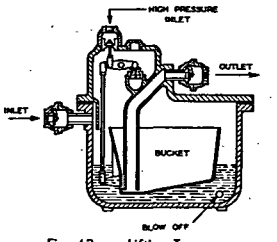


Fig. 43 .... Lifting Trap

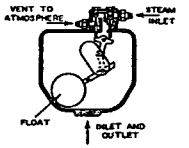


Fig. 44 .... Boiler Return Trap or Alternating Receiver

trap pressure to enable the water in the trap to flow with the boiler by gravity.

These traps are available in sizes from 1½ to 2½ in., and for pressures varying from 0 to 100 psig. A typical boiler return trap is shown in Fig. 44, and a typical connection to a low-pressure heating system in Fig. 14.

Steam Trap Installations

The following general rules should govern the installation of traps of all types:

1. A vertical drip, as long as possible, and a strainer should be installed between the trap and the apparatus it drains. Exceptions to this rule are the installation of thermostatic traps in radiators, convectors, and pipe coils. These, in general, are attached directly to the units without strainers.

2. Whenever it is necessary to maintain, in continuous service, apparatus which is to be drained, it is advisable to install a gate valve on each side of the trap, and a valved bypass around the trap, so that the trap may be removed and repaired and condensate drained through the throttled bypass valve.

3. Whenever it is necessary to install traps for lift service, as when the condensate must be discharged to a main located above the trap, or where the trap must discharge against a definite back pressure, a check valve and a gate valve should be installed on the discharge side of the trap, the check valve to prevent continuous pressure on the discharge side of the valve, and the gate valve to shut off pressure in case the trap is removed for service or repair.

DRIPS

A steam main in any type of steam heating system may be dropped to a lower level without dripping if the pitch is downward with the direction of steam flow. The steam main in any heating system can be elevated if dripped. Fig. 45 shows a connection where the steam main is raised and is drained to a wet-return. If the elevation of the low point

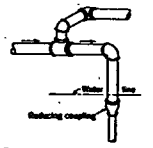


Fig. 45 .... Dripping Main Where It Rises to Higher Level

is above a dry-return, it may be drained through a trap to the dry-return in two-pipe vapor, vacuum and subatmospheric systems. Horizontal steam pipes may also be run over obstructions without a change in level, if a small pipe is carried below the obstruction to care for the condensate (Fig. 46). Horizontal return pipes may be carried past doorways and other obstructions by using the scheme illustrated in Fig. 47. It will be noted that the large pipe, in this case, runs below the obstruction, and the smaller one over it.

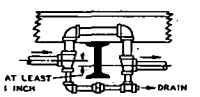


Fig. 46 .... Looping Main Around Beam

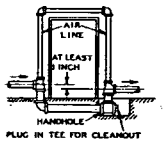


Fig. 47 .... Looping Dry Return Main Around Opening

Offsets in steam and return piping should preferably be made with 90-deg ells, but occasionally fittings of other angles are used, and in such cases the length of the diagonal offset will be found as shown in Fig. 48.

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and sludge which will interfere with their operation. Dirt pockets are usually made 8 in. to 12 in. deep, and serve as receivers for foreign matter which otherwise would be carried into the trap.

On vapor systems where the end of the steam main is

dripped into the wet-return, the air venting at the end of the main is accomplished by an air vent passing through a thermostatic trap into the dry-return line as shown in Fig. 49. On low-pressure or vacuum systems, the ends of the

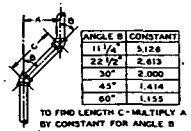


Fig. 48 .... Constants for Determining Length of Offset Pipe

steam mains are dripped and vented into the return through drip traps opening into the return line. A float and thermostatic type trap is recommended for dripping steam mains and risers as indicated in Figs. 50 and 51.

The dripping of high-pressure mains, or of equipment using high-pressure steam into low-pressure or vacuum re-

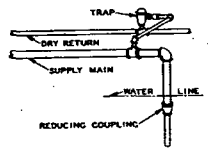


Fig. 49 .... Dripping End of Main into Wet Return

turns, is generally accomplished by the use of a flash tank or flash leg into which the high-pressure trap is arranged to discharge. This tank provides the required space for the flashing from high-temperature condensate to low-pressure steam to take place. The low-pressure steam therein generated is passed directly to the low-pressure steam mains, and the condensate is discharged through a second trap to the low-pressure or vacuum return. A typical arrangement of a flash leg, with sizes required for varying capacities, is given in Fig. 52.

CONNECTIONS TO HEATING UNITS

Riser, radiator, and convector connections must not only be properly pitched at the time they are installed, but must

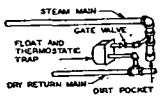


Fig. 50 .... Dripping End of Steam Main into Dry Return

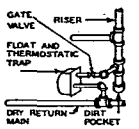


Fig. 51 .... Dripping Heel of Riser into Dry Return

be arranged so that the pitch will be maintained under the strains of expansion and contraction. These connections may be made by swing joints which permit the expansion or contraction to occur under heating and cooling without bending of pipes. To take care of expansion in long risers, either expansion joints of commercial construction or pipe swing joints are used. Anchoring of pipes between expansion joints is desirable.

Two satisfactory methods of making runouts for one-pipe systems for either the up-feed or the down-feed type are

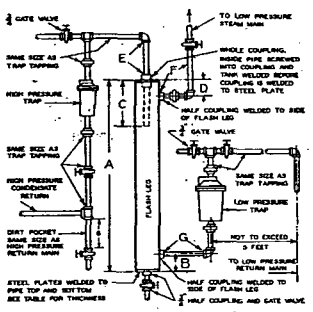
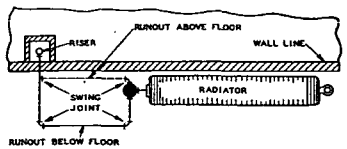


Table 12 .... Dimensions Applying to Fig. 52

Condensate per Hr, Lb at 70 Psig	Traps						Flash Leg, ips	A Ft	B In.	C In.	D In.	E In.	F In.	G In.	Plate Thickness, In.
	HP Side			LP Side											
	Pipe size	Orifice	Pres. range, psi	Pipe size	Orifice	Pres. range, psi									
200	3/4	1/8	21–80	3/4	3/16	0–20	4	3	6	8	3	3/4	1 1/4	3/4	3/8
300	3/4	5/32	31–70	3/4	5/16	0–15	4	3	6	8	3	3/4	1 1/4	3/4	3/8
700	3/4	3/16	31–70	3/4	3/8	0–15	4	3	6	8	3	3/4	1 1/4	3/4	3/8
1500	3/4	9/32	61–80	1	1/2	0–15	5	4	6	10	4	3/4	1 1/2	1	3/8
2500	1	9/32	61–80	1 1/4	3/4	0–15	8	5	6	10	4	1	2	1 1/4	1/2
4000	1 1/4	11/32	61–80	2	1 1/16	0–15	10	6	6	12	5	1 1/4	2 1/2	1 1/4	5/8
8000	2	3/16	61–80	2 1/2	1 1/16	0–15	12	6	6	12	5	2	3	2	3/4
15000	2 1/2	3/16	61–80	2 1/2	1 1/16	0–15	16	6	8	14	6	3	3	3	7/8

Fig. 52 .... Flash Leg Installation for 80 Psi Maximum Steam Working Pressure



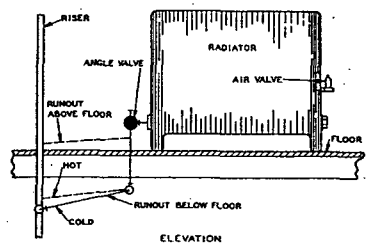


Fig. 53 .... One-Pipe Radiator Connections

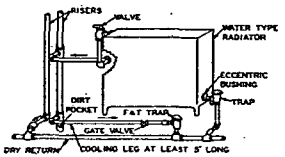


Fig. 54 .... Two-Pipe Top and Bottom Opposite End Radiator Connections

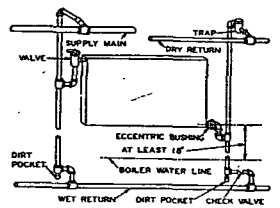


Fig. 55 .... Two-Pipe Connections to Radiator Hung on Wall

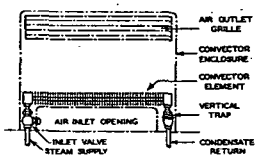


Fig. 56 .... Typical Convector Connections



Fig. 57 .... Typical Connections to Finned Pipe Convector

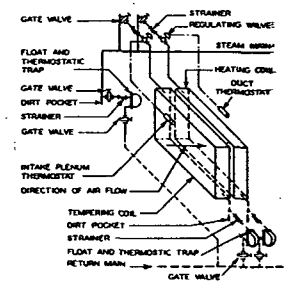


Fig. 58 .... Typical Connections to Finned-Tube Blast Heating Coils Arranged for Series Flow of Air

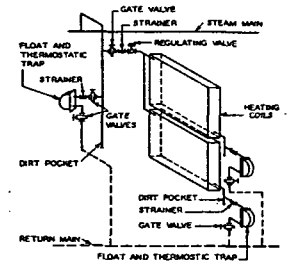


Fig. 59 .... Typical Connections to Finned-Tube Blast Heating Coils Arranged for Parallel Flow of Air

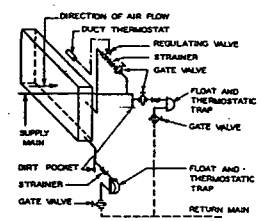


Fig. 60 .... Typical Connections to Finned-Tube Blast Heating Coils of the Steam Distributing or Nonfreeze Type

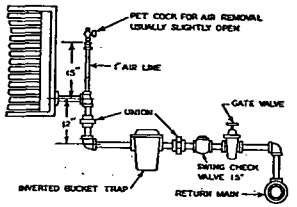


Fig. 61.... Typical Return Connections to Finned-Tube Blast Heaters with High-Pressure Steam

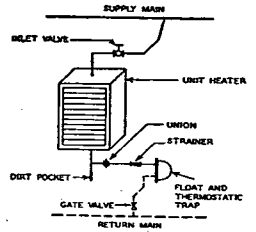


Fig. 62.... Typical Unit Heater Connections for Two-Pipe System

shown in Fig. 53. Where the vertical distance is limited and the runouts must run above the floor, the radiator may be set on pedestals or raised by means of high legs. A method of connecting a unit heater to a one-pipe steam heating system is illustrated in Fig. 2.

Typical two-pipe radiator connections are shown in Figs. 54 and 55. While these show top inlet supply connections which are preferred, it is also possible to connect the supply to the bottom of the radiator. Short radiators may be connected with top supply and bottom return on the same end.

A typical method of connecting convectors is shown in Fig. 56. Sometimes the supply valve is omitted on convector connections, and a damper is supplied in the outlet grille for heat control.

A typical connection for finned pipe convectors is shown in Fig. 57.

Typical connections to blast heaters are shown in Figs. 58, 59, and 60. Fig. 61 shows a typical return and connection for blast heaters connected to high-pressure systems.

A typical two-pipe connection to a unit heater is indicated in Fig. 62.

CONTROL VALVES

Gate valves are recommended in all cases where service demands that the valve be either entirely open or entirely closed, but they should never be used for throttling. Angle globe valves and straight globe valves should be used for throttling in such cases as bypasses around pressure-reducing valves or on bypasses around traps.

REFERENCES

[1] F. C. Houghten and J. L. Blackshaw: ASHVE Research Report No. 954—Condensate and air return in steam heating systems (ASHVE Transactions, Vol. 39, 1933, p. 199).

[2] Reducing and relief valves on consumers' premises (*American Standard Code for Pressure Piping*, ASA B31.1-1955, p. 66).

[3] *American Standard Code for Pressure Piping* (ASA B31.1-1955, p. 66).

[4] N. H. Davidson: Economies effected by combination pressure and temperature control valve (*National District Heating Association Proceedings*, 1956).

[5] *ASHVE Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps* (ASHVE Transactions, Vol. 40, 1934, p. 33).

CHAPTER 27

DISTRICT HEATING

Steam Requirements, Boiler Plants, Steam Distribution, Design Considerations, Distribution Pressures, Pipe Sizes, Conduits for Piping, Manholes and Tunnels, Accessory Equipment, Insulation, Return of Condensate, Building Piping and Equipment, Metering, Hot Water Distribution

THE term *district heating* refers to the supplying of heat from a central plant to a group of buildings in a city, institution, housing development, or industrial or commercial area. The heat may be used for any applicable purpose such as space heating, air conditioning, or processing. It is usually preferable that groups of commercial, industrial, or institutional buildings be supplied with heat from a central plant rather than from individual plants in order to permit design for better combustion efficiency with less expensive fuel, and reduction of labor per unit of output. The central plant justifies use of more competent personnel, and often decreases the investment required.

Those phases of district heating which frequently fall within the province of the heating engineer are treated here, and information is given for solving incidental problems. Some data are included with reference to special requirements for steam piping in buildings served with district steam. The data are confined principally to the use of steam as the heating medium, but the use of hot water has gained some recognition in recent years.[1]

In the design of a district heating system to provide public utility service in a city, it is advantageous to make a thorough study of the entire problem with competent men having experience in both design and operation of such systems.

STEAM REQUIREMENTS

The first step in the design of a district heating system is to determine the maximum hourly and the annual steam requirements of each of the buildings to be supplied. Methods of determining the maximum *hourly* requirements will be found for space heating in Chapter 12, for building service-water heating in Chapter 56, and for various process applications in Table 1.[2] Measured demands of several types of buildings are given in Table 9, Chapter 37.

Methods of estimating *annual* steam requirements for heating various types of buildings are also given in Chapter 37. Table 7 in Chapter 37 lists the average annual steam consumption per degree-day for buildings located in all sections of the United States. Annual steam requirements for building service-water heating are given in Table 2 of this chapter. Additional data on annual steam requirements, including those for process uses and air conditioning, of various types of buildings in a number of cities may be found in the *District Heating Handbook*, Third Edition.[3]

BOILER PLANTS

In the design of a plant for district heating, consideration should be given to the following factors:

1. *Annual load factor.* This factor is the ratio of the average hourly plant send-out (defined as rate of flow to the distribution system) to the maximum hourly send-out, which is a direct measure of the extent to which investment in facilities is used. It is often employed to determine savings by equipment offering higher efficiency. The load factor for space heating is usually between 15 and 30 percent. For space heating plus additional steam requirements such as water heating, cooking, or laundry, it is between 15 and 40 percent. Where process requirements predominate, as in factories, load factors are sometimes 50 percent or more. Load factors for various types of buildings are given in Table 9, Chapter 37.

2. *Duration of plant output.* The cumulative number of hours for the various plant send-outs and the predicted peak boiler demand on the coldest day are used in conjunction to determine the number and sizes of boilers, in order to effect most efficient loading of individual units and the plant as a whole. The cumulative number of hours a typical district heating plant will send out steam at various plant outputs, with outputs expressed in percentage of maximum hourly rate, is given in Table 3. It will be noted that the duration of plant output for the higher send-outs is very low.

3. *Feed-water treatment.* Good feed-water treatment is an absolute necessity in a plant in order to be able to operate the boilers at maximum outputs. Dirty tubes will limit boiler capacity. Proper feed-water treatment will prevent corrosion and scale formation and keep boiler outage to a minimum.

4. *Smoke abatement.* This is a modern requirement. The installation of suitable fuel-burning and dust-collecting equipment to limit the emission of smoke and fly ash is usually necessary.

5. *Plant location.* A boiler plant location is determined by an economic study of fuel handling, water supply, system piping, land costs, and other associated factors. Such a study usually reveals that the economical location of the plant will be near the distribution center, in order to simplify and shorten the distribution piping. Plant location and the local air pollution ordinances will have a direct bearing on the type of firing equipment selected and the dust collection apparatus required.

STEAM DISTRIBUTION

Distribution piping in district heating systems must be designed in accordance with the same basic principles applying to any other steam piping. The piping may be run through buildings, through the air on poles or other structures, or underground in conduits or tunnels. Local conditions and investment will be governing factors in establishing the route of the pipes.

Design Considerations

Important points in laying out distribution piping are:

1. Piping within buildings or basements is normally least expensive to install and to maintain.

2. Underground installations involve excavation which represents a substantial portion of the total cost. The depth of such excavations should be kept to a minimum after due consideration has been given to the possible damaging effect of the operating-line heat on lawns and shrubbery and the possible damage to the underground structure due to heavy traffic, etc.

3. Walking tunnels usually are not provided for steam mains

Table 1 .... Unit Steam Consumption of Process Equipment[a]

Kitchen Appliances (exclusive of water heating)	Consumption,[b] Lb per Hr	Pressure, psig
Stock and vegetable kettles (per 5 gal)	20.0	7–20
Vegetable steamers (per compartment)	40.0	7–20
Steam tables (per sq ft)	1.7	7–20
Bain-Maries (per sq ft)	3.4	7–20
Coffee urns (per gal)	3.4	7–20
Water urns (per gal)	5.0	7–20
Soup warmers (30 x 30 x 28 in.)	100.0	7–20
Egg boilers (3 compartments)	18.0	7–20
Clam, lobster, and potato steamers (per compartment)	40.0	7–20
Oyster pots	18.0	7–20
Plate and cup warmers (per 20 cu ft)	35.0	7–20
Food-warming ovens (per 20 cu ft)	35.0	7–20
Silver burnisher and washers	69.0	7–20
Dishwashers (per tray)	60.0	7–20
Laundry Equipment		
Tumblers		
40 x 94 in.	360	100
30 x 36 in.	225	100
Washers, (per gal water heated)	1	15–100
Flatwork ironers,		
100 to 120 in. chest type (per roll)	60	100
50 to 100 in. cylinder type (per machine)	60–120	100
Standard presses	105	100
Large shirt body presses	190	100
Cuff and neckband presses	15	100
Sleevers	12	100
Steam-electric irons	8	100
Woolen pressing machines		
Large—electric vacuum	25	65
Small—electric vacuum	12	65
Large—steam vacuum	35	65
Small—steam vacuum	20	65
Feather cleaning and sterilizing 15 pillows per hr	175	100

Hospital Equipment	Consumption,[a] Lb per Hr	Pressure, psig
Sterilizers, for bottles or pasteurization (per bottle)	1	40
Sterilizers, for water, (per gal)	12	40
Sterilizers, for instruments and utensils		
8 x 9 x 18 in., water depth 3½ in.	27	40
9 x 10 x 20 in., water depth 3½ in.	30	40
10 x 12 x 22 in., water depth 4 in.	39	40
12 x 16 x 24 in., water depth 4 in.	60	40
10 x 12 x 36 in., water depth 4 in.	66	40
16 x 15 x 20 in., water depth 10 in.	92	40
20 x 20 x 24 in., water depth 24 in.	144	40
Sterilizers, for instruments @ 240–250 F		
12 x 20 in.	48	40
14 x 22 in.	60	40
16 x 24 in.	72	40
Sterilizers, for surgical supplies		
10 x 20 in.	10	40
12 x 20 in.	22	40
14 x 22 in.	28	40
16 x 24 in.	38	40
16 x 36 in.	54	40
20 x 28 in.	60	40
20 x 36 in.	78	40
20 x 48 in.	98	40
20 x 60 in.	124	40
Sterilizers, (autoclave) (240–250 F)		
15.5 x 24 in.	24	40
17.5 x 26 in.	32	40
21.5 x 30 in.	40	40
24 x 36 in.	42	40
Disinfector, mattress		
30 x 42 x 84 in.	42	35–60
60 x 66 x 108 in.	318	35–60
Blanket warmers		
18 x 24 x 72 in.	4	35–60

[a] From *District Heating Handbook* (National District Heating Association, Third Edition, 1951).
[b] These values represent approximate requirements after warm-up period.

in the district heating industry because of their relatively high first cost as compared with smaller conduits unless they are required to accommodate miscellaneous other services or to provide underground passage between buildings.

4. Overhead piping on poles or structures is generally less expensive than underground lines, especially in some institutional and industrial areas and may be a means of eliminating buried conduit in areas where underground installations would be expensive or undesirable. Thin corrugated aluminum sheets or asphalt-saturated asbestos felt are available for covering the insulation on lines exposed to the weather. Such lines may introduce freeze-up problems, higher heat losses and, in many locations, would be considered unsightly.

5. High-pressure transmission feeders may be employed where the plant is remote from the distribution area, or where it is desirable to augment the capacity of an existing system Design of such feeders and of distribution systems for various load conditions is covered fully in *District Heating Handbook*, 3rd Edition.[3]

6. Piping routed in loops with suitable valves to provide alternate feeds for major loads, allows service to be maintained despite trouble in any one feeder.[4]

7. It is desirable to estimate as far as possible the probable growth and change in load centers. Since character of the district served is almost certain to change with time the distribution system must be planned so that it can be adapted to accommodate changes. It is helpful to provide a master distribution plan which can be revised from time to time to suit conditions as the system grows.

Steam Distribution Pressure

The pressure at which the steam is to be distributed will depend upon that available at the plant and pressure requirements of apparatus to be served.[4]

The advantages of low-pressure distribution (2 to 50 psig) compared with high pressure are: (1) lower heat loss per square foot of pipe surface; (2) lower losses from leakage, trap discharge, and venting; (3) less trouble with traps and valves; (4) less trouble from flash steam at drainage points; (5) simplified problems in pressure reduction at the buildings; (6) suitability of standard fittings; and (7) general reduction in maintenance costs.

Some heating companies find that the overall losses in relation to steam delivery are higher with high-pressure distribution. With distribution pressures not exceeding 50 psig, there is less danger if the full distribution pressure should build up in the radiators or heaters through the faulty operation of pressure-regulating valves. With pressures higher than 50 psig, or the safe working pressure of heating or process equipment, a second pressure-reducing valve or some form of emergency relief is required to prevent excessive pressures in the radiators or other equipment.[4]

Pressure reducing valves are more fully described in Chapter 26 (Steam Heating Systems). The installation of reducing valves on district steam installations, however, may present specific problems that are not normally encountered in building steam heating applications. Utility companies in some instances must assume greater responsibilities in their operation and maintenance and are often under regulations that are different for various cities.

The advantages of high-pressure distribution are: (1) smaller pipe sizes; (2) availability of the steam for various other operations requiring higher pressures than the building heating system; and (3) wider flexibility in allowance for maximum pressure drop. Where distribution pressures are not as high as desired, higher pressures can be obtained by use of a steam compressor.[4]

It is sometimes desirable to install a separate high-pressure line from the boiler plant for steam supply to laundries or other high-pressure equipment rather than to operate the entire distribution system at higher pressures.

Pipe Sizes

In district heating work it is common practice to design the piping system on the basis of pressure drop. The initial pressure and the minimum permissible terminal pressure are specified, and the pipe sizes are so chosen that the required amount of steam, with suitable allowances for future increases, will be transmitted without exceeding this pressure drop. The steam velocity is therefore almost disregarded and may be very high. Velocities of 35,000 fpm are not considered high for pipes in conduits. Where pipes pass through buildings, however, consideration should be given to the possibility of objectionable noise due to high velocity. By the use of permissible pressure drops with high velocities, after allowance has been made for future increase in load, the pipe sizes may be kept to a minimum with consequent savings in investment. Some lines in a loop system may need to be increased in size in order to handle reverse-direction flow in emergencies.[4]

The steam flowing through any section of the piping can be computed from a study of the hourly steam requirements of the several buildings served. In this connection, allowance should be made for the diversity of the load, that is, the ratio of the coincident maximum demand of all of the group of buildings served by the main, to the sum of the individual maximum demands of each of the buildings served. This ratio is known as the Coincidence Factor and varies from 0.50 in some institutional groups to 0.90 in areas where loads are principally for heating and heat is turned on at about the same time each morning. Consideration should also be given to the net total load on a line, based on a study of the loads in the various buildings, in order to determine the relative merits of providing one or more separate supply lines to serve all-year loads. Such lines will permit turning off the mains that supply strictly seasonal space-heating loads during the summer months.

When lengths of pipe, steam quantities, and initial and terminal pressures have been chosen, the pipe sizes can be calculated by means of Figs. 22 to 26 of Chapter 26. The district-heating industry, however, uses *Unwin's* formula for *determining pipe sizes because pressure drops* can be determined simply and directly from a chart. This formula has been checked by tests, and found correct within ranges of velocity used in district heating. A convenient chart developed from this formula is available from the National District Heating Association.[7]

Conduits for Piping

Conduits for underground steam pipes are used to protect the pipe and insulation from external damage and to provide a space in which the pipe is free to expand or contract while held in proper alignment. The conduit should be constructed of non-corrosive materials, or, if of metallic content, should be suitably protected against corrosion. It must be of adequate strength to withstand the earth and traffic forces to which it will be subjected. The conduit should be reasonably waterproof and in many cases provision should be made to drain seepage from its inside. With any form of conduit unless it is actually below water level, provision should be made to completely exclude water or to carry away ground water and prevent it from collecting around the conduit. This is often accomplished by placing a layer of coarse gravel or stone underneath and partly surrounding the conduit with one or two lines of tile having open joints which serve to carry the water to a sewer or other suitable

Table 2 .... Annual Steam Consumption for Water Heating for Various Types of Buildings, per Cu Ft of Volume[a,b]

Type of Building	Consumption, Lb/(Yr) (Cu Ft)	Consumption Lb/(Yr) (Cu Ft) (100 F deg Temp. Rise of Water)
Hotel	3.20	2.66
Residence club	2.56	2.11
Apartment house	2.41	1.99
Department store	0.95	0.79
Office	0.85	0.71
Theater	0.57	0.48
Parking garage	0.46	0.38

[a] Based on analysis by fourteen district heating companies from metered consumptions. Building volume is approximate gross volume.
[b] From *District Heating Handbook* (National District Heating Association, Third Edition, 1951, p. 347).

Table 3 .... Load Duration for Typical District-Heating System

(Cumulative Hours of Plant Send-out per Year at Various Loads Expressed in Percent of Peak Hourly Load[a])

Plant Output % of Peak	Hours per Year	Plant Output % of Peak	Hours per Year
100	5	55	1800
95	40	50	2200
90	90	40	3100
85	150	30	4100
80	300	20	5150
75	500	15	5700
70	750	10	6400
65	1050	6	8760
60	1400		

[a] Based on curve on page 89, *District Heating Handbook*, Third Edition, 1951.
Note: A plant having a peak send-out of 100,000 lb per hr will send out 80,000 (80 percent of 100,000) lb per hr *or more* for 300 hr per yr.

outlet. Proper foundations should be provided to prevent settlement and to insure proper pitch of piping at all times.

Manholes may be required at intervals to provide access to valves, traps, and expansion joints. They should be drained. Some safe means of disposing of condensate from piping should be provided.

There are many types of conduits. They fall into six following general classifications: Box-type, Solid-pour, Prefabricated, Tile, Monolithic, Granular Fused, and Waterproof. Cross-sections of a few of the typical types are shown in Fig. 1.

Box-type conduits are typical of those built with common materials and have many variations. Conduit A, Fig. 1, shows a commonly used type in which a corrugated metal form is placed as an inverted U over the insulated pipe line installed on a concrete slab. The concrete may be reinforced or not, depending upon the load on the conduit. Drainage is provided by the drainage pockets filled with crushed stone or gravel and may be installed on either or both sides of the conduit. They conduct water from the top of the conduit to the drain below which is connected to the sewer or some other disposal facility. Variations of the concrete-box type have side walls constructed of hollow vitrified tile, concrete blocks, or other suitable materials, and covered with a reinforced concrete slab. The top and sides are sometimes covered with a waterproofing membrane of asphalt or hot pitch and roofing paper.

The solid-pour construction, Conduit B, Fig. 1, is also usually made with common materials and consists of poured structural concrete which is vibrated and tamped around a conventionally insulated steam line. If the insulation has a high compressive strength it can support the pipe, otherwise, the pipe should be supported independently. Reinforcing is sometimes used to prevent settlement. An eccentric space may be left around the pipe by using insulation larger than the outside diameter of the pipe to permit upward movement of the pipe during the warm-up periods. During construction, pipe is suspended by wires. Precautions should be taken to prevent the pipe from floating. The top of the conduit should be coated with asphalt emulsion or tar to help seal shrinkage cracks. Anchor stresses are higher with this type of construction. Variations of this type include enclosure of the pipe and insulation in tile, concrete, steel, or asbestos pipe and then pouring of concrete around outer pipe. This method provides space for pipe supports, if needed. External drainage can be provided by methods similar to the concrete box type construction.

Conduits constructed of common materials are often preferred by the district heating industry because of the conditions imposed in congested street beds, heavy traffic loads, and drainage conditions or because of the ease of opening and closing the conduit, for branch pipe connections.

Prefabricated conduits, C and D, Fig. 1, are commonly constructed in standard lengths. In some, as in C, the insulated pipe or pipes are placed within a corrosion-resistant helically-corrugated metal jacket which is protected with a heavy asphaltic coating. Similar casings are made of steel protected with an anti-corrosive compound. Both are of welded construction and can be tested under air pressure. In conduit D the insulated pipe is placed concentrically within the corrosion-resistant metal jacket and the intervening space between jacket and insulation poured full of high melting-point asphalt or left as an air space. In these conduits, the pipes are supported by precast rings or guides

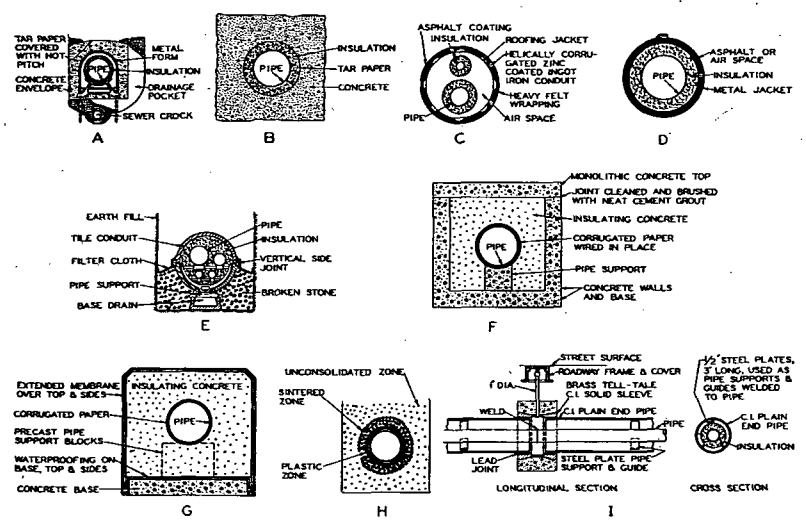


Fig. 1 . . . . A Few Typical Conduits for District Heating Use

to maintain alignment and to prevent pressure on insulation. Fittings are available for joints, bends, etc. Some of these conduits are also available with cast-iron casings.

Tile conduit E, Fig. 1, is constructed of specially-shaped glazed vitrified bell-and-spigot tile. In the conduit shown, the base drain is made in separate units and is designed to support the main conduit and pipe lines, and to drain the conduit. All joints are cemented. There are several types of tile conduits.

Monolithic conduits F and G, Fig. 1, are similar to the solid-pour type except that the insulation, referred to as insulating concrete, serves also as a conduit and is poured directly around the steam main. The insulation consists of expanded-mica, shredded-rubber or other cellular materials mixed with Portland cement or asphalt; or of a special foaming material mixed with Portland cement. The latter process creates a multitude of minute air cells in the concrete to obtain its insulating qualities. The poured structure is often protected by a water-proofing membrane and strengthened by a concrete duct.

Granular-fused type conduits such as H, Fig. 1, consist of a specially sized and selected granular bitumen which is poured in the trench around the pipes. The manufacturers advise that the conduit is formed when heat passing through the pipes forms three distinct concentric zones, all resistant to water penetration. A plastic area surrounds the pipe, outside of which is a sintered zone providing thermal insulation, and finally, an outer layer of unconsolidated particles providing further insulation value. This unconsolidated zone is also the load-bearing portion of the structure.

Waterproof conduits are intended for use where permanency and thoroughly tight joints are required. Some of the prefabricated conduits, C and D, Fig. 1, can be supplied with water-tight cast-iron casings and fittings. The casing shown in I, Fig. 1, consists of cast-iron pipe with lead-caulked joints enclosing the insulated steam main. It is water tight and intended for special applications, where, for instance the duct may be submerged, subject to flooding, or laid in extremely wet soil.

All of the preceding conduits have their application and the choice depends upon local conditions. Any underground conduit is only as good as, among other things, its foundations to prevent settlement, its external and internal drainage, its ability to stand up under the corrosive action of certain ground conditions (see Buried Pipe Lines, Chapter 55), its strength to carry the load imposed on it, its field installation, and its insulating properties.

Manholes

When conduits are used for underground piping, manholes may be required to house valves and other accessory equipment. They are built of a variety of materials such as reinforced concrete block. They should have adequate volume to permit men to work on equipment. They should be as nearly waterproof as possible and precaution should be taken to prevent entrance of surface water around the covers. Provision should be made for lifting necessary equipment from the pipe line, for drainage, for ventilation, and for automatic water-removal equipment if a sewer is not accessible.

Details of typical manhole construction housing a single expansion joint are shown in the *NDHA District Heating Handbook*, Third Edition, p. 164. If more equipment is installed, the size should be increased accordingly.

Where water removal equipment is required, sumps are installed under the manholes. Water can be removed periodically with portable pumps, or, where water conditions are bad or periodic inspection is not feasible, by installed sump pumps or water ejectors.

The motors for electric pumps should be enclosed in a separate compartment which is ventilated to protect it from heat and vapors. Precautions should be taken against corrosion as mentioned in the Section Piping and Accessory Equipment.

Piping Tunnels

Because of their relatively high first cost as compared with conduits, *walking tunnels* are usually not provided for steam mains unless, as in some institutions, they are required to accommodate miscellaneous other utility services or to provide underground passage between buildings.

Tunnels are usually built of reinforced concrete, brick, concrete blocks or other suitable structural materials, and are waterproofed on the outside. Provision for drainage and ventilation is essential.

Since tunnels and conduits are apt to be hot and humid, special precautions should be taken to reduce temperatures and protect equipment. Heat loss from pipes and supports can be reduced by insulating return mains and using more than normal insulation on steam mains. A metal jacket over the pipe covering can permit insulation to carry the weight of the pipe.[8]

The precautions on sizing, design, installation, and protection against corrosion, for accessory equipment, as discussed in the sections Piping and Accessory Equipment, and Conduits for Piping, should be taken for tunnels. Special precautions should be taken to prevent rapid deterioration of enclosed electrical conduit. All electrical boxes, junctions and outlets should be vapor proof. Ventilation of the tunnel can be obtained by installation of suitable fans or by natural means.

Piping and Accessory Equipment

A district heating distribution system is usually buried in the ground where it is not readily accessible for repairs or replacement. This makes it imperative that the system be carefully designed and installed. The *American Standard Code for Pressure Piping* ASA B-31.1—1955[9] has sections, covering district heating piping systems and fabrication details, from which general specifications and minimum requirements for piping and accessory equipment can be obtained.

Suggestions for selection and installation of materials based on long operating experience of the district heating industry are given in following paragraphs. More detailed information is available in the *District Heating Handbook*, Third Edition.[10]

Pipe. Either steel or wrought-iron pipe is satisfactory. Welded construction is most desirable.

Valves. Flanged cast-iron valves for pressures under 125 psig are satisfactory. For pressures from 125 to 300 psig flanged-steel and welded-end valves are used. Outside screw and yoke valves are preferred.

Fittings. Forged or rolled steel is used for pressures in excess of 125 psig. Where pipe-line apparatus is to be removable, flanged fittings are preferred.

Expansion Joints. Pipe bends to absorb expansion are preferable wherever feasible in order to eliminate the inspections, maintenance, and housing required by slip joints. Slip joints should be guided either externally or internally, or both. Corrugated-type joints should have stainless steel

or copper corrugations depending upon operating conditions. Spare expansion capacity should be allowed for contingencies such as higher pressures, superheat, and installation tolerances.

Anchors. Anchors can be of structural steel, special fittings, or U-shaped steel straps which partially encircle the pipes. Anchors should be firmly bolted or welded to a short length of cast or stainless steel set in concrete. Anchors are often fabricated by welding structural steel to pipe and embedding the ends in manhole walls or concrete poured in connection with conduit. Anchors are subject to humid conditions in conduits and manholes, and can deteriorate rapidly. They should be designed for adequate strength, intended life, and the steam pressure to which they can be subjected, with liberal factors of safety and protection from corrosion. Anchors should be attached to both sides of the pipe. Lugs should be welded to pipe on both sides of loop-type anchors to prevent slippage. The cantilever type of anchor should be avoided. Cast steel is preferred to cast iron for anchor material.

Alignment Guides. To assure pipe alignment and proper operation of expansion joints, pipes should be adequately guided. Special precautions should be taken if the pipe deviates from a straight line.

Pressure Reducing Valves. Suitable single-seated valves are preferable. The valves should be selected from the manufacturer's rating regardless of pipe size. Pressure gages are necessary at valve and control points. Valves require some auxiliary source of power for valve positioning, if they are to be controlled from a remote point to regulate terminal pressure. (See Chapter 26 for detailed recommendations regarding pressure reducing valves.)

Traps. To assure adequate drainage of piping, extra trap capacity and large drip pockets should be provided. Extra heavy or corrosion-resistant trap discharge piping is desirable. On steep grades or in high-velocity steam mains, special deflector scoops or baffles should be placed in the line to direct condensate into traps.[11]

Supports. Due to humid conditions in conduit and manholes, it is desirable to use stainless steel supports.

Instruments. The instruments required will be the same as for other steam systems as described in Chapter 43, Automatic Controls, or Chapter 44, Instruments and Measurements, with modifications depending on the records desired with regard to pressures and temperatures maintained or the records of response of the system to varying load demand.[12]

Branch Connections. Branch connections should be made at or near anchor points; if not, ample clearance must be provided around the branch to take care of pipe movement.

Insulation

Insulating materials constitute a primary element of district steam distribution structures and must be suitable for underground service. Insulation should be noncombustible, vermin-proof, moisture-resistant, durable, non-corrosive to steel pipe especially when wet, and able to retain its position in relation to the pipe line. It should be of a type which, if flooded, will dry out and regain its original physical, chemical and insulating properties. It should be relatively unaffected by such chemicals as are usually found in ground or other contaminated waters. It should have the lowest practical heat conductivity consistent with the given requirements.

Insulating properties vary with soil conditions, weather variations, types of conduits used, and, if in tunnels, the amount of ventilation. The selection of insulation to provide for the various conditions is covered in *District Heating Handbook*—Third Edition. Where underground insulation is applied directly to the pipe, copper wire is normally used to band the covering. Insulation may be poured as a monolithic structure in a suitable duct. Where insulation in an existing conduit has deteriorated the line may be reinsulated with a minimum amount of excavating by use of some of the materials used in monolithic conduits.[13] Insulation is discussed further in Chapter 32, and studies of heat loss from bare and insulated buried pipe are given in *District Heating Handbook*[14] and in *Piping Handbook*.[15]

RETURN OF CONDENSATE

If the condensate can be returned to the plant to be used for boiler feed water at an overall cost less than that required to process raw water, the installation of return lines parallel to the steam supply lines should be given consideration. A typical analysis of comparative costs of returning condensate and processing feed water is given in *District Heating Handbook*.[16]

Return of condensate by gravity is desirable, but usually because of terrain conditions or user's return facilities, the condensate is pumped back under pressure.

Pipes carrying condensate are more subject to corrosion than other parts of the system. Care must be taken to give proper pitch to the pipes and to provide proper venting of non-condensable gases. (See Chapter 55, Corrosion.)

BUILDING PIPING AND EQUIPMENT

There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from district steam mains. Some of these are unique to district heating and some, although they may apply to any building have been demonstrated to be particularly important when district steam is used. District heating companies have certain regulations regarding the consumer's installation to safeguard the mutual interest of supplier and consumer and to insure satisfactory and economical service. When an installation is designed, local regulations and code requirements[17] should be observed.

Zoning. Where the hours of occupancy or heating loads differ in various sections of the building, it is good practice to install separate supply lines to the different sections. For example, in an office building with first floor stores or restaurants open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the building in the late afternoon. The division of the building into zones, each with a separately-controlled heat supply, is sometimes desirable, as it permits the heat to be adjusted according to variations in sunshine and wind. It is desirable, also, to provide separate lines for water heating and for various cooking or process appliances. In order that zones may operate at the optimum pressure, it is often advisable to install a separate pressure-regulating valve for each. See Fig. 29 of Chapter 26 for typical service installation with separate lines to heating zones and process equipment.

The heat supply should be graduated according to variations in the outdoor temperature. Because of the continuous availability and constant pressure of district steam, the most economical operation and satisfactory heating can only be obtained by the use of automatic temperature control which, except in unusual cases, is well justified. The application of control equipment in various types of build-

Table 4 .... Suggested Capacity of Traps

Type of Control	Trap Capacity, Lb/(Hr) (Sq Ft of Pipe and Radiation[a])
Continuous or modulating operation, intermittent with *on* periods over 3 hr in duration	⅓
Intermittent operation with *on* periods of ½ to 3 hr in duration	⅓
Intermittent operation with *on* periods of ½ hr or less	⅓[b]

[a] Square Feet EDR.
[b] Except for drip traps on mains which should be 1 lb per sq ft EDR.

ings is fully discussed in the book, *Principles of Economical Heating*.[18] Various types of control equipment are discussed in Chapter 43.

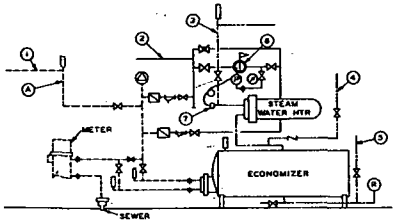
Rapid draining and venting of the heating system should be obtained by means of suitable traps and air vents in order to obtain rapid circulation and quick heating. This is particularly important with intermittent-type temperature-control equipment. The selection of traps is discussed in Chapter 26. Suggested trap capacities are given in Table 4 of this Chapter. To obtain efficient and economical operation of any system it is imperative that all traps be properly maintained. In a system supplied with district steam any leakage through traps may prevent air removal from the system and otherwise impair its operation. Furthermore, such steam must be condensed and its condensate cooled before being discharged to sewer. This represents a direct waste of steam.

Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed. A considerable amount of heat can be saved by shutting off steam at night and other times when optimum building temperatures are not required.[19] Equipment is available which will automatically shut heat off at night for varying lengths of time, depending upon outdoor temperature.[20]

Residual heat in the condensate should be salvaged. This heat may be salvaged by means of an air-cooling coil, or as is more frequently done, by a water-heating economizer, shown in Fig. 2, which preheats the hot water supply to the building.[21]

The condensate from the heating system, after leaving the traps, passes through the economizer. The water supply to the heater passes through the economizer, absorbing heat from the condensate. If the hot-water system in the building is of the recirculating type, the recirculating line should be connected *between* the economizer and the water heater proper, not at the economizer inlet, because the recirculated hot water is itself at a high temperature.

Since the heating-system load and the hot-water demand are not coincident a greater amount of heat can be extracted from the condensate if storage capacity is provided for the preheated water. Frequently, therefore, the economizer coils are submerged in a storage tank. Where condensate is pumped, the pump should discharge into a receiving tank equipped with a thermostatic air vent and the condensate should be permitted to flow by gravity through the economizer and meter.



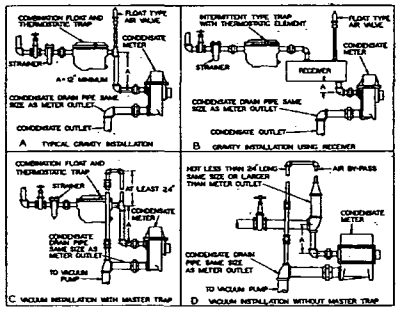
1. Main Return Line from gravity heating system with all returns properly trapped or pump discharge from vacuum system. With vacuum system a receiving tank with thermostatic vent should be provided at Point A.
2. Steam Supply to Water Heater
3. Hot Water Supply to Building
4. Recirculating Line
5. City Water Line
6. Pressure-Regulator Temperature Control Assembly With Bypass
7. Thermostat Bulb and Tubing

Fig. 2 .... Method of Installing a Water Heater and Economizer in a Gravity Heating System

METERING

The *Condensate Meter* was developed for the district heating industry and is an approved method of measuring condensate where all of it can be brought to a common point for metering purposes. Its simplicity of design, ease in testing, accuracy at all loads, low cost, and adaptability to either high- or low-pressure distribution have made it standard equipment with most heating companies.

Condensate meters must not be operated under pressure;



Meter Capacity Lb Per Hr	250	500	750	1500	3000	6500	12000
	Dimension A—Inches						
Diagram B.....	4	4	8	8	8	12	12
Diagram C.....	12	12	15	15	15	18	18
Diagram D.....	12	12	15	15	15	18	18

Fig. 3 .... Condensate Meter Installation Details

they are made for either gravity or vacuum installations. Typical meter installations are shown in Fig. 3. In diagram A a continuous-flow type trap is shown installed ahead of the meter, while in diagram B an intermittent type trap is used. In the latter case, a receiving tank must be placed between the trap and meter to prevent intermittent overloading of the meter. When measuring the discharge from a vacuum pump, a vented receiver should always be installed ahead of the meter.

Installations of meters in a vacuum return line are shown in diagrams C and D. In diagram C a master, continuous-flow trap is installed immediately ahead of the meter to prevent steam from entering it in case of leaking radiator or fixture traps. Where individual traps are reliable, connections may be made as shown in diagram D.

Flow meters are generally used for measuring sendout from plants, high-pressure requirements in buildings, and in installations where all of the condensate cannot be returned to a central point. Further information on flow meters is available from publications of the Fluid Meter committee of the American Society of Mechanical Engineers and from the *District Heating Handbook*,[28] and are described in Chapter 4.

HOT WATER DISTRIBUTION

District heating in the United States is confined almost entirely to the use of steam as the heating medium. Hot water has been used and is being used today, where economic conditions are favorable and good engineering practice dictates.

The use of low-pressure hot water distribution below 200 F has been practically discontinued for district heating in the United States because of practical and economic difficulties. The use of high-pressure, high-temperature water distribution at 275 to 400 F has found favor in Europe, where fuel is scarce and is finding some application in the United States.[29]

High-pressure water distribution has some practical and economic advantages over steam (see Chapter 29 High Temperature Water Systems). Higher thermal efficiencies, better daily and annual system load factors, lower water treatment costs, smaller pipe sizes, fewer piping accessories, and feasibility of installing pipe to follow contour of ground are given as a few. On the other hand hot water requires a two-pipe distribution system which increases the capital investment in some cases. System maintenance is complicated by the necessity to drain at least a portion of the system when connections are made or leaks repaired.

The advantages of hot water have exceeded the disadvantages in some industrial applications and in Air Force bases in this country. As the cost of fuel increases, additional applications will probably become practicable, especially in new installations serving buildings designed for hot water heating.[1]

REFERENCES

[1] P. L. Geiringer: Recent developments in high temperature water for area heat distribution in the United States (*National District Heating Association Proceedings*, 1956).

[2] Steam requirements for processes, maximum demand and load factor (*District Heating Handbook*, National District Heating Association, 3rd ed., p. 346).

[3] Influence of pressure (*District Heating Handbook*, National District Heating Association, 3rd ed., p. 147).

[4] Effect of looping (*District Heating Handbook*, National District Heating Association, 3rd ed., p. 155).

[5] *American Standard Code for Pressure Piping* (American Standards Association, B 31.1, 1955, p. 66).

[6] Steam pressure boosters (*District Heating Handbook*, National District Heating Association, 3rd ed., p. 329).

[7] *Graphical Solution of Unwin's Formula* (chart published by the National District Heating Association). For explanation of chart see *District Heating Handbook* (National District Heating Association, 3rd ed., p. 194).

[8] Gordon Carlson: Improved method of supporting steam pipes in tunnels (*National District Heating Association Proceedings*, 1955, p. 125).

[9] District heating piping systems, and Fabrication details (*American Standard Code for Pressure Piping*, American Standards Association, B 31.1, 1955, Sections 4 and 6).

[10] Piping, valves, fittings and accessories (*District Heating Handbook*, National District Heating Association, 3rd ed., p. 169).

[11] Steam traps and drain pockets (*District Heating Handbook*, National District Heating Association, 3rd ed., p. 175).

[12] Control of steam flow (*District Heating Handbook*, National District Heating Association, 3rd ed., p. 211).

[13] The reinsulation of underground steam mains (*National District Heating Association Proceedings*, 1945, p. 21).

[14] Underground insulation specifications, Thermal conductivity, Steam pipe insulation, Soil temperature surrounding a buried steam line, and Heat loss from insulated buried steam line (*District Heating Handbook*, National District Heating Association, 3rd ed., p. 162, p. 180, p. 184, p. 188, and p. 189, respectively).

[15] Sabin Crocker: Heat loss (*Piping Handbook*, McGraw-Hill Book Co., 1955, 4th ed., p. 1016).

[16] Return of condensate (*District Heating Handbook*, National District Heating Association, 3rd ed., p. 158).

[17] Reducing and relief valves on consumers' premises (*American Standard Code for Pressure Piping*, American Standards Association, B 31.1, 1955, p. 66).

[18] Temperature control (*Principles of Economical Heating*, National District Heating Association, 5th ed., p. 25).

[19] Hours of heating (*Principles of Economical Heating*, National District Heating Association, 5th ed., p. 22).

[20] H. T. Kucera: New developments in programming building heating (*National District Heating Association Proceedings*, 1956).

[21] Utilization of heat in condensate (*Principles of Economical Heating*, National District Heating Association, 5th ed., p. 40).

[22] Metering (*District Heating Handbook*, National District Heating Association, 3rd ed., p. 221).

[23] Hot water for district heating (*District Heating Handbook*, National District Heating Association, 3rd ed., p. 423).

CHAPTER 28

HOT WATER HEATING SYSTEMS

Features and Design of Low Temperature Hot Water Systems; Two-Pipe, One-Pipe, and Series-Loop Systems; Piping Principles; Circulating Pumps; Expansion Tanks; Boiler Room Piping; Use of Steam; Temperature Control; System Adjustment; Cleaning Systems

A HOT WATER heating system is one in which water is used to convey heat by flowing through pipes connecting a boiler or water heater with radiators, convectors, or other suitable heat distributing means. The systems discussed in this chapter have supply water temperatures less than 250 F and are classified as *Low Temperature Systems*. Systems having supply water temperatures above 250 F are classified as *High Temperature Systems* and are discussed in Chapter 29. Classified according to the means of generating flow, hot water heating systems are of two types: the *Gravity System* in which circulation of the water is due to the difference in weight between the supply and return water columns of any circuit or system, and the *Forced System* in which a pump, usually driven by an electric motor, maintains the necessary flow.

Since the available head or force producing circulation in a gravity system is limited, the piping must be of such size that the friction loss at the desired flow is not in excess of the available head.

Gravity circulation heads which can be obtained with various supply and return temperatures are shown in Fig. 1. The basic principles which determine design in gravity and forced systems are the same but since the former has less total head available to produce flow, greater precautions are necessary in design to secure even distribution of heat with gravity systems. Gravity systems are no longer of commercial importance. Complete details on the design of gravity type systems are contained in editions of THE GUIDE issued before 1957. Fig. 1 is also useful in determining effects due to temperature difference in forced systems when large elevation or temperature differentials occur.

This chapter is devoted to the design of forced hot water systems using supply water temperatures below 250 F.

The selection of type of distribution system for heating a particular building should be based upon an objective analysis of the characteristics of the building, the comfort and temperature conditions required, the system operating and maintenance characteristics, the space available for piping and equipment, and the economics involved.

FEATURES OF HOT WATER SYSTEMS

Advantageous features of forced low temperature hot water systems are:

1. The piping need not be run at a definite level or pitch but may change up and down as required by architectural and structural elements of the building. High points should be vented and low points be provided with drain connections.
2. Heat distribution when warming up is uniform.
3. Hot water systems are readily adaptable to simple approximate control of capacity and indoor temperature by instruments sensitive to outdoor temperature changes.
4. The system may generally be designed with a minimum of mechanical specialties thereby favoring economical maintenance.
5. Due to practical elimination of air, corrosion within the system is minimized.
6. Where extended glass exposures occur, the heating elements, which are relatively large because they operate at low temperature, may be well distributed under such exposures to counteract downdrafts.
7. Quiet operation may be expected in a properly designed and installed system.

In the design and operation of forced circulation systems, it is necessary to take precautions against any condition that would permit the water temperature to fall below freezing, particularly where coils are used to heat incoming outdoor air or where the heating plant may be shut down for more than 24 hours in cold weather.

If the static pressure in the system is high, it may be necessary to use boilers built for operating pressures above 30 psig unless a steam boiler with heat exchanger is used to heat the water. See discussion in this chapter in section Expansion Tanks and System Pressure Control.

DESIGN OF LOW TEMPERATURE HEATING SYSTEMS

The significant factors affecting the design of low temperature hot water heating systems are discussed in the sections that follow.

Heat Loss Calculations

The adequacy as well as the proper distribution of the heat in a building served by a hot water system depends primarily on the proper calculation of the heat loss for the individual spaces and the accurate sizing of the heating units to offset those heat losses. Heat loss should be determined in accordance with the principles discussed in Chapter 12, Heating Load.

Selection and Location of Heating Units

Information on the types of room heating units available for hot water systems will be found in: Chapter 14, Radiators and Convectors; Chapter 30, Panel Heating; and Chapter 15, Unit Ventilators and Unit Heaters. While the style or type of unit selected for a given space may depend on personal preferences of the purchaser, the designer should make certain that the heat output at the water temperature available is equivalent to the calculated heat loss from the space.

The heating units within each separately controlled circuit should be of similar types, for example: either all cast-iron or all fin-tube convector-type units. Fan units should not be included in a controlled direct radiation zone. The

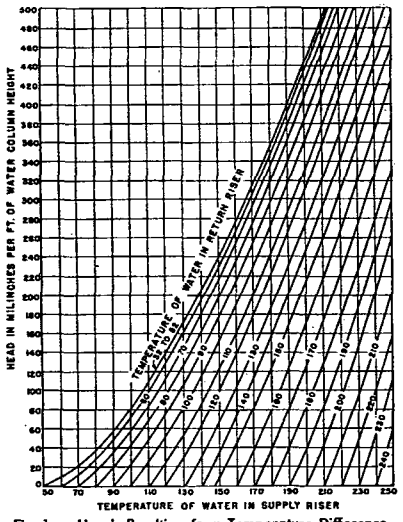
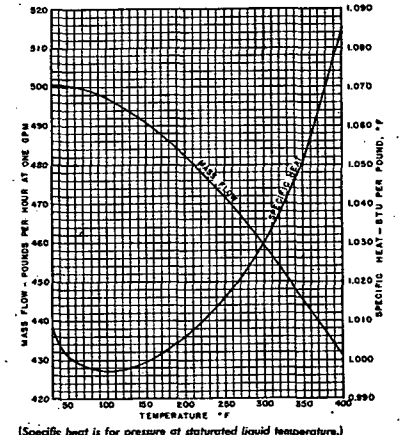


Fig. 1.... *Heads Resulting from Temperature Difference (Gravity Systems)*



(Specific heat is for pressure at saturated liquid temperature.)
(Basic data for calculation and construction of curves from Keenan and Keyes.)[1]

Fig. 2.... Mass Flow and Specific Heat of Water

radiation should be located at points of maximum heat loss or along outside walls, unless this is impracticable.

Determination of Water Flow Rate

The rate of water flow in a system is determined by the heat carrying capacity of the water. This capacity is the product of the rate of flow, the specific heat, and the temperature drop of the water as it passes through the heating units. The equation generally used to describe this relationship is:

$$H = Wc(t_1 - t_2) \qquad (1)$$

where

H = heat carrying capacity, Btu per hour.
W = mass flow or weight of water, pounds per hour.
c = specific heat of water.
t_1 = temperature of water entering heating unit, Fahrenheit.
t_2 = temperature of water leaving heating unit, Fahrenheit.

The equation may be rewritten as follows for convenient use with Fig. 2.

$$H = Gwc(t_1 - t_2) \qquad (2)$$

where

G = rate of water flow, gallons per minute.
w = rate of water flow per gallon per minute, pounds per hour.

The value of w should be obtained from Fig. 2 for the temperature at which the rate of flow is determined, and the value of c should be determined at the average temperature. For example: If water enters a heating unit at 210 F and leaves at 190 F, and a water flow of 1 gpm is measured at 190 F, the heat carrying capacity, found by substituting values from Fig. 2 in Equation 2, is

$H = (1)(484)(1.005)(210\text{-}190) = 9728.4$ Btu per hr or 9.73 MBh

At 60 F the mass flow rate is 500 lb per hr at 1 gpm and the specific heat is very close to 1 Btu per (lb) (F deg). Accordingly, at a 20-deg temperature drop, 1 gpm will release 10 MBh.

This value, 10 MBh per gpm of water at a 20-deg temperature drop, is extensively used throughout the hot water heating industry in the calculation of data for low temperature systems. It is considered well within the limits of accuracy of heating system design. If 1 gpm will release 10 MBh at a 20-deg drop, it will release 15 MBh at a 30-deg drop, and 20 MBh at a 40-deg drop.

Most low temperature systems installed today are designed on the basis of a 20-deg temperature drop. The temperature drop selected in the design of a system is used to determine the water flow necessary to supply the heat required under conditions of maximum heat loss with supply water temperature at a maximum value. If the supply water temperature is increased at any time, the capacity of a heating unit and the temperature drop of water through the unit is increased accordingly. When the supply water temperature is modulated according to outdoor temperature, the temperature drop of the water in the system is considerably less than the design value during the greater part of the heating season.

The use of a design temperature drop greater than 20 deg should be considered where large quantities of required ventilation air are heated by extended-surface coils, where the required pumping capacity is high, where pumping costs are a critical item, and where extended systems make a saving by the reduction in pipe size possible.

Factors considered in selection of a proper temperature drop for a system are (1) the effect of the design average water temperature and water velocity on the capacity and cost of the heating units, (2) the size and cost of the piping, and (3) the initial and operating cost of the pump.

Design Water Temperature

The *design supply water temperature* is the maximum temperature of the water supplied to the system for the purpose of developing the selected rated capacity of the heating units.

The *design average water temperature* of a hot water heating system is the design supply water temperature less ½ the design temperature drop. Capacities of heating units are generally determined on the basis of the design *average* water temperature for cast-iron radiation and for baseboard or finned pipe radiation. Where a greater degree of precision is required to establish the capacity of heating units, for example, for contract and competitive bidding work, it may be desirable to determine heating unit capacities on the basis of design supply water temperature and temperature drop through the unit. This is particularly true where the secondary heating surface in the heating element of the unit is a large percentage of the total heating surface, as in convectors. Design supply water temperatures for radiation systems vary from 180 to 250 F, a common temperature being 210 F.

Factors considered in the selection of a design supply water temperature are (1) the effect on the capacity of the heating unit, (2) the effect on occupants who may be near the unit, and (3) the greater possibility of boiling as the pressure varies.

Pipe Sizing Methods

The theoretical basis of calculations of head loss due to friction of fluids in pipe is the Darcy or Fanning equation, discussed in some detail in Chapter 4, Fluid Flow.

Pipe sizes for hot water heating systems are determined most conveniently from charts such as Figs. 3 to 6 and Table 1 showing pressure loss in relation to rate of flow and size of pipe. Allowance for fittings is generally expressed in elbow equivalents which is the number of 90-deg elbows that would have equivalent friction loss at the same rate of flow. Fig. 7 and Table 2 are useful in determining the number of elbow equivalents which then may be converted to equivalent length of pipe by means of Table 3. Figs. 3 to 7 and Tables 1 to 3 are based chiefly on empirical equations developed by F. E. Giesecke in cooperative research with the Society at the University of Texas and the Engineering Experiment Station of the Agricultural and Mechanical College of Texas.[8-12]

The friction loss selected for sizing pipe for a system depends upon the type of pump used and the economics of small pipe size and high pump head versus large pipe and low pump head. Noise is also an important factor. Velocities are usually limited to a maximum of 4 fps for pipe sizes less than 2-in. to reduce the possibility of noise. Velocities up to 10 fps are frequently used in piping larger than 6-in. Average friction drops of 1.0 to 4.0 ft of water per 100 ft of pipe (120 to 480 milinches per ft of pipe) are commonly used in sizing piping systems. The actual friction loss for a

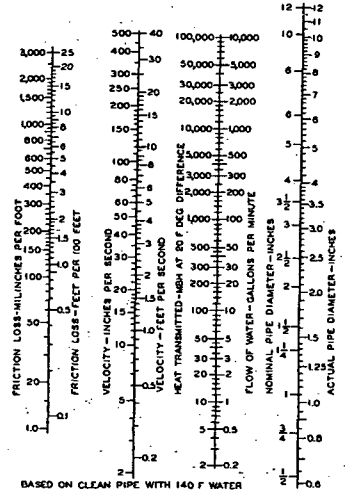


Fig. 3.... Friction Loss Due to Flow of Water in Iron Pipe[2,3]

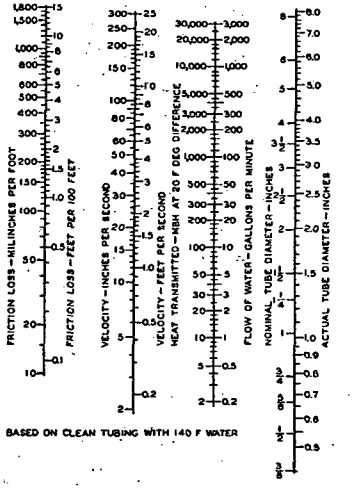


Fig. 4.... Friction Loss Due to Flow of Water in Type L Copper Tube[4]

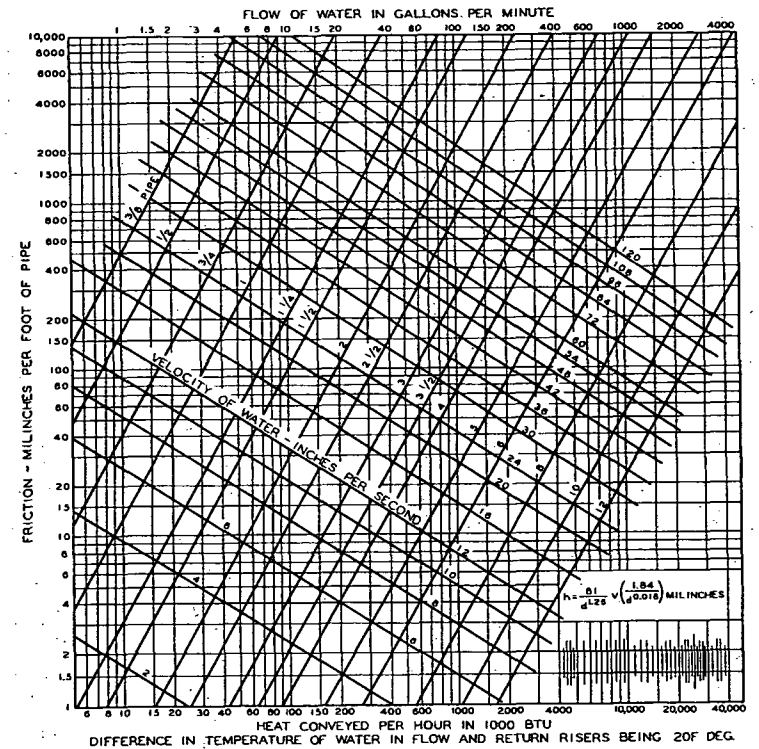
particular section of pipe may vary considerably from the average loss for the system due to the limitations of commercial pipe sizes. For example, if the average friction loss is 2.0 ft of water per 100 ft of pipe (240 milinches per ft), the actual friction loss in particular sections of pipe may vary from 1.0 to 3.0 ft per 100 ft (120 to 360 milinches per ft).

One method of sizing pipe is based on preliminary selection of a pump. The head which the pump can produce is divided by the equivalent length of the longest circuit to obtain an average design friction loss.

The equivalent length of a system or circuit is the length of pipe in series flow in the circuit, plus the length of pipe equivalent in resistance to the fittings and also the equipment in series flow in the circuit. For preliminary calculation the equivalent length is sometimes assumed to be 1.5 times the measured length.

Table 1 can be used for pipe sizing by this method. It provides the designer with a means for quickly assessing the relative effects, economic and otherwise, of various pumps on the system in terms of pipe size.

After sizing the piping on the basis of a preliminary design friction loss the actual friction loss at design rate of flow



Lower scale of chart is based on 20-deg temperature difference between flow and return risers. To find friction when temperature drop is other than 20 deg, multiply the actual heat conveyed by (20 ÷ actual temp. drop) and read the corresponding friction.

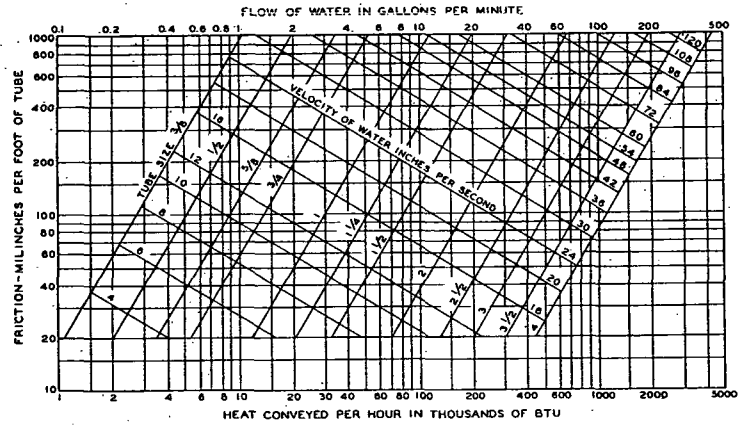
Conversion Ft/(100Ft) to Millinches/Ft

Ft per 100 ft	0.5	1	2	3	4	5
Millinches per ft	60	120	240	360	480	600

Fig. 5. . . . Friction Loss due to Flow of Water in Iron Pipe

Table 1. . . . Carrying Capacity of Standard Black Pipes with Temperature Drop of 20 F Deg*

| Available Head—Feet | | |



Lower scale of chart is based on 20-deg temperature difference between flow and return risers. To find friction when temperature drop is other than 20 deg, multiply the actual heat conveyed by (20 ÷ actual temp. drop) and read the corresponding friction.

Conversion Ft/(100 Ft) to Milinches/Ft						
Ft per 100 ft	0.5	1	2	3	4	5
Milinches per ft	60	120	240	360	480	600

Fig. 6. . . . Friction Loss due to Flow of Water in Type L Copper Tube

Table 2 . . . . Iron and Copper Elbow Equivalents[a, 8, 9, 10]

Fitting	Iron Pipe	Copper Tubing
Elbow, 90-deg	1.0	1.0
Elbow, 45-deg	0.7	0.7
Elbow, 90-deg long turn	0.5	0.5
Elbow, welded, 90-deg	0.5	0.5
Reduced coupling	0.4	0.4
Open return bend	1.0	1.0
Angle radiator valve	2.0	3.0
Radiator or convector	3.0	4.0
Boiler or heater	3.0	4.0
Open gate valve	0.5	0.7
Open globe valve	12.0	17.0

[a] See Table 3 for equivalent length of one elbow.

should be calculated. The friction loss at design flow for all individual sections of pipe and for all fittings in the longest piping circuit should then be summarized.

Table 2 shows the number of elbow equivalents for various fittings. These equivalents can be converted to equivalent feet of pipe by use of Table 3. Fig. 7 shows the elbow equivalents for determining friction loss in tees. If the pipe size calculation indicates a required pump head different from that of a standard pump, pipe sizes in the system may be changed to establish a closer relationship between the two. The relationship between systems and pump heads is discussed in the section Circulating Pumps.

If there are more circuits than one in a system, the friction loss for the longest circuit should be used to determine

Table 3 . . . . Equivalent Length of Pipe for 90-Deg Elbows

Vel. Fps	Pipe Size														
	½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	8	10	12
1	1.2	1.7	2.2	3.0	3.5	4.5	5.4	6.7	7.7	8.6	10.5	12.2	15.4	18.7	22.2
2	1.4	1.9	2.5	3.3	3.9	5.1	6.0	7.5	8.6	9.5	11.7	13.7	17.3	20.8	24.8
3	1.5	2.0	2.7	3.6	4.2	5.4	6.4	8.0	9.2	10.2	12.5	14.6	18.4	22.3	26.5
4	1.5	2.1	2.8	3.7	4.4	5.6	6.7	8.3	9.6	10.6	13.1	15.2	19.2	23.2	27.6
5	1.6	2.2	2.9	3.9	4.5	5.9	7.0	8.7	10.0	11.1	13.6	15.8	19.8	24.2	28.8
6	1.7	2.3	3.0	4.0	4.7	6.0	7.2	8.9	10.3	11.4	14.0	16.3	20.5	24.9	29.6
7	1.7	2.3	3.0	4.1	4.8	6.2	7.4	9.1	10.5	11.7	14.3	16.7	21.0	25.5	30.3
8	1.7	2.4	3.1	4.2	4.9	6.3	7.5	9.3	10.8	11.9	14.6	17.1	21.5	26.1	31.0
9	1.8	2.4	3.2	4.3	5.0	6.4	7.7	9.5	11.0	12.2	14.9	17.4	21.9	26.6	31.6
10	1.8	2.5	3.2	4.3	5.1	6.5	7.8	9.7	11.2	12.4	15.2	17.7	22.2	27.0	32.0

the pump head requirement. Piping in the other circuits should be sized to obtain the same total friction loss.

Effects of Antifreeze Fluids. Antifreeze solutions are sometimes used in heating systems when the danger of freezing exists. They should not be used in direct-fired boiler application. Corrections must be made in flow rates, pipe size, and pump capacity relative to water when antifreeze solutions are used. See Chapter 49, Snow Melting.

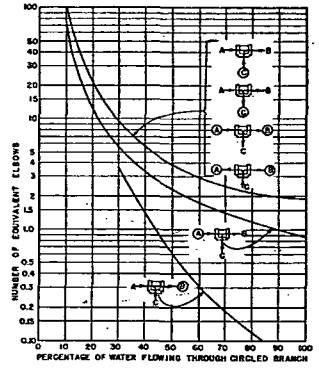
SELECTION OF PIPING ARRANGEMENT

The type of distribution system selected for any particular building will depend primarily on the structural and architectural characteristics of the building, the need for separate control of different zones, the pressure head due to the height of the building, the space available for piping, and the relative cost of different arrangements of piping. Consequently, mains may be run in basement, separate floors, or in space above highest heating units.

There are four distinct arrangements of main piping: (1) the two-pipe reversed-return system, (2) the one-pipe system, (3) the series-loop system, and (4) the two-pipe direct return system. The last is not recommended unless elaborate pipe sizing technique and provision for adjustment are employed.

Two-Pipe Reversed-Return System—Description and Design

The two-pipe reversed-return main system, Fig. 8, has two mains, one supplying water to the heating units or to risers



Notes: 1. The chart is based on straight tees, that is, branches A, B, and C are the same size.

2. Head loss in desired circuit is obtained by selecting proper curve according to illustrations, determining the flow at the circled branch, and multiplying the head loss for the same size elbow at the flow rate in the circled branch by the equivalent elbows indicated.

3. When the size of an outlet is reduced the equivalent bellows shown in the chart do not apply. The maximum loss for any circuit for any flow will not exceed 2 elbow equivalents at the maximum flow (gpm) occurring in any branch of the tee.

4. The top curve of the chart is the average of 4 curves, one for each of the tee circuits illustrated.

Fig. 7. . . . Elbow Equivalents of Tees at Various Flow Conditions[9, 11]

to heating units, and one collecting water returned from these units. The piping is arranged in such a manner that the sum of the equivalent length of supply and return piping to any unit is approximately equal to that of any other unit.

Compared with a one-pipe system, the two-pipe system may:

1. Have a smaller average pipe size. While two mains are required, the sizes of the mains vary as water passes from one main through the heating unit to the other. The maximum size of either main is no larger than the size of a one-pipe main.
2. Be more flexible in its application because larger friction loss may be used in the heat-transfer unit circuits.
3. Require a minimum pumping head because the heating unit resistances are in parallel.
4. Permit adjustment of flow through individual units of radiation over wider limits.

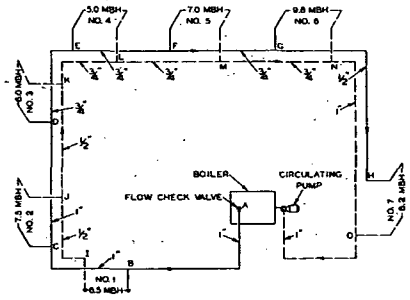


Fig. 8. . . . A Two-Pipe Reversed-Return System

Continuing the comparison, some disadvantages are:

1. Two mains may require more pipe fitting labor than a single main.
2. Some additional pipe may be required to achieve a reversed return.
3. A two-pipe system will be slightly more expensive for small systems.
4. The piping for two mains may be unsightly in occupied spaces.

Important considerations in the design of a two-pipe reversed-return distribution system are:

1. The head which the pump must generate is equivalent to the friction drop through the circuit having the highest resistance. At design flow, the resistance through the distribution system to any one heating unit should be the same as that through any other unit. Where the length of pipe required to obtain distribution to a particular section must be shorter than to other sections, it is desirable to design that section with a higher friction loss.
2. A reversed return, as indicated in the section Adjustment of Flow and Capacity, will reduce the need for adjustments necessary to obtain equal total head through each heating unit circuit.
3. Fan-type air-heating units with extended-surface coils can be used with a higher water temperature drop than that recommended for direct radiation. It is necessary to avoid a difference in air temperature across the face of the coil in such units. One method of minimizing this difference is to install a minimum of two serpentine coils in each unit, one in front of the other. Coil

connections are made so that water travels from top to bottom in one coil and from bottom to top in the other.

The design of a two-pipe reversed-return system will be illustrated by *Example 1*.

Example 1: Design a two-pipe reversed-return hot water heating system for a 37 x 29 ft one-story residence with basement having a heat loss of 50 MBh (50,000 Btu per hr). The heating units are to be convectors operating with a supply water temperature of 210 F and a 20-deg temperature drop. The piping layout and location of convectors are shown in Fig. 8.

Solution: A 50 MBh load with a 20-deg drop requires a water circulation of 5 gpm. The length of the reversed-return circuit is determined as 138 ft of pipe, by following the supply from the boiler to point *G* Fig. 8 and the return from point *N* back to the boiler. The estimated total equivalent length is 138 × 1.5 = 207 ft. This, plus an allowance of the equivalent of 25 ft of pipe for one convector circuit results in a total equivalent length of pipe of 232 ft. Fig. 9 shows that a 1-in. pump has a capacity of 5 gpm at 4.7 ft available head. The preliminary average design friction loss is then 4.7 × 100/232 or 2 ft per 100 ft of pipe (240 milinches per ft). This value can also be determined by reference to Table 1, where, using the vertical column under 5 ft, the next larger equivalent length to 232 ft is 250 ft. To the right in this line read the average design friction loss as 2 ft per 100 ft or 240 milinches per foot.

The piping may be sized initially from Table 1 as follows: In the friction loss column find the value 2 ft per 100 ft. Proceed to right in the same line to the nearest gpm value and then read the pipe size at the top of the column.

The pipe sizes found by this method for the various sections of Fig. 8 are shown in the following table:

Section	MBh	Gpm	Pipe Size inches
Supply			
A–B	50	5.0	1
B–C	43.5	4.35	1
C–D	36	3.6	¾
D–E	30	3.0	¾
E–F	25	2.5	¾
F–G	18	1.8	¾
G–H	8.2	0.82	½
Return			
I–J	6.5	0.65	½
J–K	14	1.4	½
K–L	20	2.0	¾
L–M	25	2.5	¾
M–N	32	3.2	¾
N–O	41.8	4.18	1
O–A	50	5.0	1

The risers to the convectors may also be sized for the same average friction loss, 2 ft/100 ft or 240 milinches per foot, as follows:

Convector No.	1	2	3	4	5	6	7
Load MBh	6.5	7.5	6.0	5.0	7.0	9.8	8.2
Gpm	0.65	0.75	0.60	0.5	0.7	0.98	0.82
Pipe size	½	½	½	½	½	½	½

After the initial pipe sizes have been selected the friction loss in the system should be recalculated as indicated in the following table. The highest pressure drop circuit should be marked off into sections; each of a different flow rate. The total equivalent length of each section should then be determined by adding its elbow equivalent length (see Tables 2 and 3 and Fig. 7) to its actual measured length. Total friction loss is established from

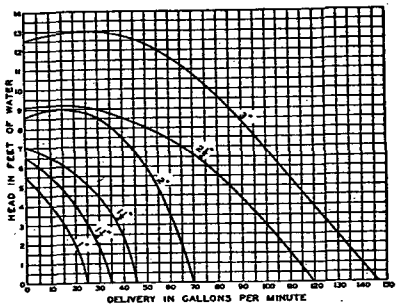


Fig. 9.... Performance Chart for Circulating Pump

charts or tables for each section by multiplying its total equivalent length by the actual friction loss per foot at design flow. The circuit friction pressure drop is the summation of individual section pressure drops as shown in Table 4.

If the total friction loss as determined does not agree with the pump head, some pipe size changes will be necessary to obtain closer agreement. The slight difference in head calculated by milinches and by feet of head is due to use of only 2 significant figures for the latter. This is well within the accuracy required.

As would be expected the calculated head of 4.02 ft is different from the 4.7 ft estimated. This is due to two factors: (1) the difference between the actual and estimated number of fittings and (2) the differences between actual and estimated losses in the different sections as the friction losses were generally below the average friction loss assumed.

The 1-in. pump initially selected would be entirely satisfactory for the installation and would produce a flow slightly in excess of 5 gpm. A procedure to determine the actual flow and head capacity of the pump and system is discussed under Circulating Pumps.

One-Pipe System, Description and Design

The one-pipe system, Fig. 10, has a single main for both supply and return. Special fittings installed at the connections to radiation risers or runouts generate the head necessary to produce flow through the heating unit. In this system the main or circuit loop does not change size from the first to the last radiation unit. The amount of water flowing in the main is constant, except at points where some of it is bypassed through radiation. Since the water temperature in the main drops progressively, some designers consider it necessary to increase the size of heat-transmitting surface accordingly by using a progressively lower design water temperature in sizing the heating units. Most one-pipe heating systems are designed, however, on the basis of a single design water temperature. Generally, the theoretical increase in heat-transmitting surface is needed only when the system is operating at full design output. A water temperature drop in the main of 20 deg or less minimizes the capacity reduction effect.

In comparison with a two-pipe system, the one-pipe system:

1. Permits simpler piping due to the use of one main of uniform size.
2. May be more acceptable when pipe is exposed in occupied spaces.
3. Is very adaptable to small systems or to individual loops

Table 4.... Calculation of Actual Friction Loss for *Example 1*

Section	Flow MBh	Flow Gpm	Friction Loss (Fig. 3) Ft/100 Ft	Friction Loss (Fig. 3) Mi/Ft	Pipe Size In. (Fig. 3)	Fittings Type	Fittings Equiv Ft[b]	Pipe Length Ft	Total Equiv Length of Pipe Ft	Friction Loss Calculation	Friction Loss Mi[c]	Friction Loss Ft
BC	43.5	4.35	1.5	180	1	1 ell	2.4	14	16	0.16 × 1.5 16 × 180	2880	0.24
CD	36.0	3.6	1.0	120	1	—	—	15	15	0.15 × 1.0 15 × 120	1800	0.15
DE	30.0	3.0	2.5	300	¾	1 ell	1.9	8	10	0.10 × 2.5 10 × 300	3000	0.25
EF	25.0	2.5	1.8	210	¾	—	—	10	10	0.10 × 1.8 10 × 210	2100	0.18
FG	18.0	1.8	1.0	120	¾	—	—	14	14	0.14 × 1.0 14 × 120	1680	0.14
GN	9.8	0.98	1.2	150	½	3 ells 1 ¾ in. tee 1 1 in. tee 1 Rad. V	3.6 3.4 4.8 3.6	8	23	0.23 × 1.2 23 × 150	3500	0.26
NO	41.8	4.18	1.4	170	1	1 ell	2.5	31	34	0.34 × 1.4 34 × 170	5780	0.48
OB	50.0	5.00	1.9	230	1	3 ells 2 tees 2 Gate valve 1 Flow valve 1 Boiler	7.5 10.0 2.5 50.0 7.5	38	116	1.16 × 1.9 116 × 230	26700	2.20
Convector friction loss[a]											1200	0.10
Total friction loss											48590	4.02

[a] Data obtained from manufacturers' catalogs.
[b] See also Tables 2 and 3 and Fig. 7.
[c] Mi = milinches.

in large systems, either horisontal or vertical, as a part of a two-pipe system, or to obtain zone control in large combination systems.

4. May be lower in installation cost.
5. Has a simpler problem in regard to water-flow adjustment.

Continuing the comparison with two-pipe systems some disadvantages of one-pipe systems are:

1. The amount of water which can be bypassed by a one-pipe fitting or fittings is limited by the economics of design and the head applied. The fittings are not generally used to provide the water flow necessary for large air-handling units having a high pressure drop.
2. The pump head required is slightly higher than that required for a two-pipe reversed-return system of similar flow requirements.
3. If the progressive increase of radiation size as temperature drops is taken into account, a design complication and additional cost factor are encountered.
4. The main flow design temperature drop should not exceed 20 deg.

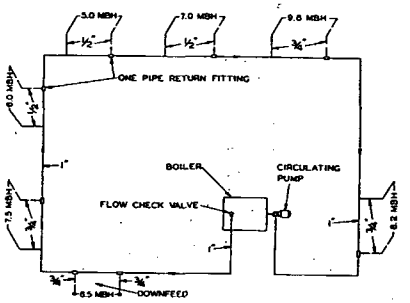


Fig. 10.... A One-Pipe System

Design of a One-Pipe System

The basic design premise of a one-pipe system is that the pressure drop of the one-pipe fitting, or fittings, at the required main flow, be equal to or greater than the radiation circuit pressure drop at its required flow. The system may be divided into two or more loops for smaller main pipe sizes and pumps. Main and radiation circuit temperature drops may be different. For example, a 10-deg temperature drop may be used in the main with a 20-deg drop through the radiation.

Example 2: A one-pipe system is to be designed for the residence discussed in *Example 1*. Fig. 10 shows the piping arrangement for the one-pipe system. The size of the convectors is determined on the basis of a 210 F design supply water temperature and a 20-deg design temperature drop as in *Example 1*.

Solution: For initial selection of pump use a 1-in. pump having a capacity of 5 gpm at 4.7-ft head. The measured pipe length is 140 ft which establishes an equivalent length of 140 × 1.5 =

210 ft. The preliminary design friction loss is then found in Table 1, as in *Example 1*, to be 2¼ ft per 100 ft or 270 mi per ft. At this average design friction loss (see Table 1) a 1-in. main will be used for the required flow of 5 gpm.

From manufacturers' data at 5 gpm flow, the pressure drop from a 1-in. fitting is established at 3000 mi or 0.25 ft. If any heating units are located below the main the gravity head due to temperature difference should be subtracted from available fitting pressure drop. In this example, the riser and convector friction pressure drop should not exceed 3000 mi or 0.25 ft.

The friction loss through the ½-in. risers to the 9.8 MBh convector was calculated to be 0.28 ft or 3450 mi in *Example 1*. This loss plus the 0.1-ft (1200-mi) loss in the convector, equals 0.38 ft (4650 mi) which exceeds the capacity of the fitting. If ¾-in. risers are used, however, the riser pressure drop is considerably reduced. For 9.8 MBh flow the pipe friction drop is 0.34 ft for 100 ft (40 mi per ft). The equivalent length changes from 22 to 26 ft. The riser pressure drop is then 26 × 0.34 or 0.09 ft (26 × 40 or 1040 mi). The head loss through the convector circuit is then 0.09 + 0.1 or 0.19 ft (2240 mi). Since this is below the available fitting pressure drop, the riser size is satisfactory.

Other upfeed risers are sized in a similar manner.

The piping to the 6.5 MBh convector below the main is sized as follows:

The water in this circuit will cool to room temperature during off periods. The center line of the convector is 6½ ft below the main. The thermal head opposing circulation in the convector circuit when one riser is filled with 210 F water and the other has 70 F water is found to be 465 mi per ft or 465 × 5.5 = 2560 mi total. Subtracting 2560 from the 3000 mi pressure drop in the fittings leaves 440 mi available to *start* circulation. The combined convector and riser pressure drop equals 1380 mi even when a 1-in. riser is used. Since the convector and riser pressure drops are greater than the available fitting pressure loss, a change must be made. The total fitting pressure drop could be increased by adding a supply one-pipe fitting, or by increasing fitting pressure drop through use of a larger pump. In this case, the riser is sized on the basis of an added fitting. In accordance with the sizing procedure followed in this example, a ¾-in. riser is adequate.

The pipe sizes selected should be recalculated at the design flow rates either in milinches as shown in the upper part of the following table or in feet per hundred feet as shown in the lower part of the table.

Section	Pipe Size In.	Load MBh	Loss per Foot of Pipe Mi	Length—Ft			Total Friction Loss Mi[a]
				Linear Pipe	Elbow Equiv.	Total Equiv.	
Main	1	50	240	140	87[b]	227	54,500
8 one-pipe fittings at 3000 mi[c]							24,000
Total							78,500
							= 6.55 ft

Section	Pipe Size In.	Load MBh	Loss per 100 Ft of Pipe Ft	Length—Ft			Total Friction Loss Ft
				Linear Pipe	Elbow Equiv.	Total Equiv.	
Main	1	50	2.0	140	87[b]	227	4.54
8 one-pipe fittings at 0.25 ft head[c]							2.0
Total							6.54

[a] Mi = milinches.

[b] The elbow equivalent length is based on: 8 elbows, 2 tees, 2 gate valves, 1 flow check valve and 1 boiler.

[c] The actual value for resistance of the one-pipe fittings should be obtained from the fitting manufacturers. They are frequently lower than the value assumed in this example.

The pump capacity required is 5.0 gpm at 6.54 ft head. The 1½ in. pump as shown in Fig. 9 would meet the requirements of the system based on the assumptions stated for the problem.

Series-Loop System—Description and Design

This system, illustrated in Fig. 11, consists of one or more loops or circuits. In each circuit, as the same water is circulated through the heating units in succession, the size of pipe should not vary materially in any portion of a circuit. The system circuit length thus becomes very important because it influences directly the water flow rate, pressure drop, and temperature drop. Since the water temperature in each

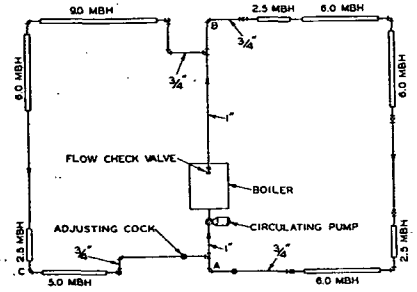


Fig. 11.... A Series-Loop System

successive heating unit decreases progressively, some designers consider it necessary to increase progressively the size of heating units sufficiently to compensate for the drop in temperature. If a small design temperature drop, such as 20 deg or less is used, it may not be necessary to increase the heating units in size toward the end of the circuit.

Various considerations affecting the design of a series-loop system are given in the following paragraphs:

1. It may be possible to eliminate all or large sections of the distribution main and thereby reduce cost. In large buildings it may be necessary to use a partial two-pipe main in combination with the series-loop system for certain sections or divisions of the building.

2. The practicability of the system will depend upon the quantity of water to be circulated through the heating units and the friction loss. Baseboard or finned-type pipe elements for series-loop systems are usually manufactured from standard steel pipe or copper tubing. The friction loss can be determined from Figs. 3 and 4.

3. If a series-loop system serves more than a single space, and if the design temperature drop for the system is in excess of 20 deg, the size of the heating unit should be determined on the basis of the temperature of the water entering the particular heating unit and the temperature drop across the unit. The temperature drop through a particular heating unit is equal to the capacity of that unit divided by the capacity of the total radiation circuit multiplied by the total temperature drop of the circuit.

4. Since the temperature of the water supplied to heating units in the series-loop system cannot be regulated, it is advisable to provide dampers in the cabinets containing finned pipe or baseboard radiation to permit adjustment of the heat delivery by each unit. Valves or adjusting fittings, if used in the water circuit, would affect the performance of the entire loop and would not be a satisfactory means of controlling the heat output of the units.

Certain disadvantages of the series-loop system are:

1. The difficulty of making suitable provision for adjustment and balancing of the system.

2. The total circulating head required may be excessive.

3. The design of the system may be more difficult than for one-pipe or two-pipe main systems.

Example 3: A series-loop system is to be designed for the residence used in *Examples 1* and *2* for two-pipe and one-pipe design. No heat will be supplied to the basement. The total heat loss equals 45,500 Btu. Baseboard-type radiation having a rating of 0.5 MBh per linear foot at an average temperature of 190 F, is selected as the radiation used. The heating units are of baseboard type having a rated output of 0.5 MBh per linear foot at an average temperature of 190 F. They are connected by ¾-in. pipe. A 20-deg design temperature drop will be used.

Solution: For a 45.5 MBh load 4.55 gpm must be circulated for a 20-deg temperature drop. For a single looped main the measured length equals 140 ft. The approximate equivalent length equals 140 × 1.5 or 210 ft. If the 1¼-in. pump, Fig. 9, were initially selected, it would develop a 6-ft head at 4.55 gpm delivery. For an available head of 6 ft and an equivalent length of 218 ft (nearest to 210 ft) Table 1 shows that only 3 gpm will circulate through the ¾-in. main. A larger, and more costly pump may be tried. The 3-in. pump, Fig. 9, has 12.7 ft available head at 4.55 gpm delivery. If 12-ft available head and 250-ft equivalent length are used as the nearest available values, Table 1 indicates that a flow of 4.3 gpm at a velocity of 2.5 fps will be established through the ¾-in. main. However, because of cost, appearance, and temperature drop considerations, this may not be desirable.

If the system is divided into two loops, as shown in Fig. 11, a smaller pump may be used and the system temperature drop decreased.

The 1¼-in. pump at 6-ft head for 4.55 gpm is again selected for preliminary sizing of piping.

The measured supply main length plus the length of the longest loop circuit is 106 ft. The equivalent length equals 106 × 1.5 or 159 ft. For 6-ft available head and 160 ft equivalent length, Table 1 indicates that the friction loss would be 3¾ ft per 100 ft or 450 mi per ft. At this friction loss the supply main would be of 1-in. size for a flow of 7 gpm (which is more than the required 4.5) and the ¾-in. looped main would deliver 3.7 gpm (more than the required 2.25 gpm). Because the flows are more than required, the system temperature drop would be less than 20 deg. A smaller pump could therefore be used if desired.

The pipe sizes should be recalculated at the design flow rates as follows:

Section	Heat Load MBh	Design Flow Gpm	Friction Loss Ft/100 Ft	Pipe Size In.	Equiv. Ft	Total Equiv. Ft	Press Drop Ft
AB	45.5	4.55	1¾	1	36.0[a]		
	2 elbows				5.0		
	2 gate valves				2.5		
	1 flow valve				50.0		
	1 boiler				7.5	101	1.77
BCA	22.5	2.25	1½	¾	70[a]		
	8 elbows				13.6		
	1 cock				1.7		
	2 tees @ 50%				13.6	99	1.5
Total							3.27

[a] Actual pipe length.

Since the total system head loss equals 3.27 ft at a flow of 4.55 gpm and the 1¼-in. pump is a satisfactory selection.

If desired, the actual design temperature drop can be determined. Because the pump head at 4.55 gpm is greater than 3.27 ft, the flow in the system will increase until pump head and flow equal system pressure drop and flow. Construction of a system curve, as described under centrifugal pumps, shows that for the system with the pipe size selected and with the 1¼-in. pump, equilibrium will be reached at 6-ft head and 6.2 gpm flow. The actual temperature drop for design conditions would then be (4.5/6.2) × 20 or 14.5 deg, approximately.

Combination Systems

Three types of systems have been illustrated, namely: one-pipe, two-pipe reversed-return, and series-loop. Actually the design for a particular building may be a composite of two or even all three of these types of systems. For example, a two-pipe reversed-return system of distribution may be used with a number of one-pipe loops in a single building such as an apartment house, or a two-pipe reversed-return system may be used with a main riser at each end of a building with a series-loop system used to serve the space on each floor of the building. In any case, it is well to bear in mind the principles of design involved in each particular type of system for that portion of the installation where it is used.

Adjustment of Flow and Capacity

Where the heating elements and piping are properly sized and arranged, a hot water system is, to a large extent, self-adjusting. This is so because the mean temperature of the water flowing inside the heating unit is the most important factor in determining its output. If the flow of water is within 5 to 15 percent of the correct value for a given heating unit, the difference in mean temperature will not exceed 1 or 2 deg and, consequently, the effect on heat output will be negligible. However, the limitations of commercial pipe sizes and pipe arrangement sometimes result in flow rates considerably different than those required to develop the proper output of the heating units. A means of adjusting the heat output of heating elements and branch circuits must be provided in the system to correct these effects.

Adjusting means may be necessary due to the type of system, for example, the two-pipe system using direct-return mains. In this system the water flow circuits are hydraulically unbalanced, because the water for the first heating load taken off the main is the first to be returned. As a result, the first heating unit will receive a much greater flow than is required to develop its rated output. The flow through the last heating unit can be so low that practically no heat is delivered. The poor distribution of flow in the direct-return system can be corrected by the installation of orifices or of adjusting fittings or valves, by means of which the system is balanced after it is placed in operation. In some cases, proper flow distribution can also be established by careful pipe size selection. That is, the piping can be sized so that the pressure drop through each piping circuit is the same at the design flow rate.

These troubles are minimized by the reversed-return system, Fig. 8. In this system the water for the first heating load taken off the main is the last to be returned, and all circuits are of approximately equal length. Proper hydraulic balance is therefore easily obtained and the system is inherently self-adjusting. While the reversed-return piping circuit is much preferred, in some large installations a saving in pipe can be effected by use of a direct-return system.

When selecting the type of piping arrangement such as one-pipe, two-pipe, series-loop, upfeed, and downfeed, the

adaptability of each arrangement for proper hydraulic balance should be considered.

Air Elimination

If air and various other gases which occur in hot water heating systems are not eliminated from the flow circuit, they may collect in the heating units or piping and reduce the flow of water and the heat output. These gases, designated *air* in this text, are in the piping before it is filled with water, are introduced with the water used to fill and replenish the system, and also are formed to a limited degree from chemical action inside the system and from the movement of low temperature air-saturated water from the

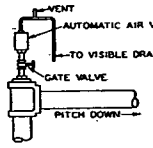


Fig. 12.... Typical Automatic Air Vent Valve Installation

expansion tank to the system.[13] Loss of air by absorption in the water may be prevented by a flexible diaphragm preventing contact of water and air in the expansion tank and having adequate movement to permit the expansion of the water in the system.

The air is eliminated from the flow circuits by trapping it into the expansion tank as explained in connection with discussion of *dip tubes* in the section Boiler Room Piping and

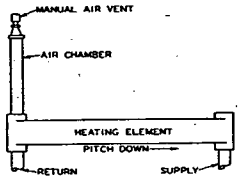


Fig. 13.... Typical Air Vent Valve Installation on Heating Element

by the installation of manual or automatic air vent valves at all high points in the system.

Automatic air-vent valves may be used to eliminate air from the mains and major equipment. Like other mechanical devices, they are subject to malfunction. Small pieces of dirt or scale can lodge on the valve seat, causing it to leak. Corrosion, scale, or sludge in the valve can prevent the valve from opening to eliminate air. All systems should be designed to have a minimum number of automatic air-vent valves, but where a valve is required there should be no hesitancy about using it. In large systems where large numbers of automatic air-vent valves are necessary, provision should be made for service of each valve without draining the system (Fig. 12). The drip from each air vent valve should be piped to a drain to prevent building damage and facilitate inspection for leakage. Automatic air-vent valves should not be installed in piping where temperatures in excess of 215 F can occur without careful analysis of possible temperature-pressure effects.

Each unit of radiation which can act as an air trap should be provided with an air chamber and air-vent valve (Fig. 13). The use of manual air-vent valves or porous plug- or disc-type manual-automatic valves is recommended for radiation.

Piping Principles

Piping should be pitched up in the direction of flow in all cases when pump operation is constant, and wherever possible when pump operation is intermittent to assure consistent air elimination during conditions of either flow or no-flow. Piping is pitched ¼ in. per 10 ft as standard practice.

When changing the size of pipe in horizontal runs, eccentric reducers should be used to keep the top of the pipes in line.

A means of manual capacity adjustment should be provided for each heating unit. Manual valves or adjusting cocks are frequently provided for this purpose. Dampers at the heating element or at the outlet of convection-type units may be used. The heating unit located in the area of a zone thermostat should be equipped with a key-operated adjusting device to discourage tampering by unauthorized personnel.

The piping should be installed to provide for thermal expansion of the pipe without imposing undue stress on connections or equipment. Spring pieces of sufficient length, as required by the diameter of the runout and the amount of expansion to be absorbed, should be installed at connections to radiators and risers. Expansion joints or loops and properly located anchors should be used where required. See Chapter 31, Pipe, Fittings, Welding.

All mains and risers located in exterior walls should be covered with sufficient insulation to prevent freezing. Trunk mains running between buildings must be protected by conduit or installed in trenches below the frost line and properly insulated. When heating mains are run in locations where they could possibly be immersed in water, the conduit should be watertight.

Circulating Pumps

Circulating pumps used in hot water heating systems can vary from small ⅙-hp booster-type pumps with a capacity of 5 gpm at 6- or 7-ft head to pumps handling hundreds, even thousands, of gallons per minute at heads limited only by the pressure characteristics of the system. The selection of a pump for a system is based on an analysis of operational characteristics of the pump and system. Both first cost and operating cost are involved. See discussion under Pipe Sizing Methods, Expansion Tanks and System Pressure Control. Pumps with mechanical seals have found wide acceptance in hot water heating. Pump characteristics, as affected by system characteristics, should be given consideration to insure non-overloading conditions in pump motor operation.

Any given pump, when operated with a given fixed-head system, will operate at only one point on its curve, the intersection of the pump curve with the system curve. Pump curves may be obtained from manufacturers. System curves can be constructed by plotting head versus gpm for different flow rates. The head for various flow rates can be estimated by use of the equation:

$$\frac{H_2}{H_1} = \left[\frac{G_2}{G_1}\right]^2 \quad \text{or} \quad H_2 = H_1\left[\frac{G_2}{G_1}\right]^2 \qquad (3)$$

where

H_1 = calculated system head.
H_2 = head at new flow rate.
G_1 = calculated flow rate.
G_2 = assumed new flow rate.

For economic or for operational reasons two similar pumps are sometimes used in parallel on a single circuit. When so used the combined pump curve showing the relationship of flow to head due to operation of both pumps is established by adding the flow capacity of one pump to the flow capacity of the other at identical heads. If two pumps are placed in series, the relationship of flow to head due to operation of both pumps is established by adding the pump heads at identical flows. When using pumps in series, they must be identical or similar since the maximum flow will be limited by the smaller pump. Gate valves should be provided to enable isolation of either pump when only one is in operation.

The Heat Source or Boiler

The most frequently used source of heat is the 30 psi hot water heating boiler, constructed of cast iron or steel in accordance with the *ASME Boiler and Pressure Vessel Code*, Section IV for Low Pressure Heating Boilers. See Chapter 35, Heating Boilers, Furnaces, Space Heaters. Boilers, classed as low-pressure heating boilers, may be used for hot water with pressures as high as 160 psig if the temperature of the water does not exceed 250 F and if they are constructed according to the ASME code for the pressure at which they are operated.[14]

The ASME code requires installation of a relief valve on each hot water heating boiler to limit the pressure exerted on that boiler to the value for which it was designed and tested.[14] Similar requirements exist for converters and electric water heaters. This operating pressure limit for the boiler is an important factor in the design of the hot water heating system. Together with the size and characteristics of the expansion tank, it determines the permissible height from the boiler to the top element or pipe of the distribution system, as discussed under Expansion Tanks. In addition to the pressure relief valve required by the ASME code, a hot water heating boiler should be equipped with a thermometer, located at or near the top of the boiler, and a reliable pressure or altitude gage with a range at least twice that of the maximum operating pressure or altitude allowable for the boiler.

EXPANSION TANKS AND SYSTEM PRESSURE CONTROL

The objectives of system pressure control are to limit the pressure at all equipment in the system to the allowable working pressure of the equipment; maintain minimum pressure for all normal operating temperatures in order that air can be vented and cavitation at the pump suction and boiling of system water prevented; and accomplish these objectives with as little addition of new water to the system as possible.

An expansion tank is the primary device used to accomplish system pressure control. The excess water volume in the system resulting from increased temperature is stored in the expansion tank during periods of high operating temperatures and returned to the system when the system water temperature is lower. The expansion tank must be designed to store the required volume of water during maximum design operating temperatures without exceeding the maximum operating pressure allowable, and to maintain required minimum pressure when the system is cold. Some designers provide an automatic fill valve to maintain the minimum

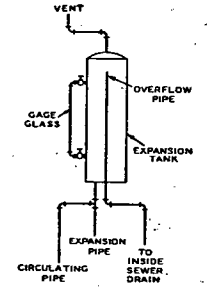


Fig. 14.... An Open Expansion Tank

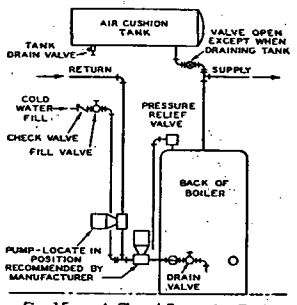


Fig. 15.... A Closed Expansion Tank

pressure in the system by supplying water to make up for losses due to leakage. Others prefer that water added to the system be supplied by a manually operated valve.

The relief valve installed on a hot water boiler in accordance with the ASME code performs the function of limiting the maximum pressure at the boiler. Since it is a safety device, it should not be considered an operating control.

Systems are designated as open (Fig. 14) or closed (Fig. 15).

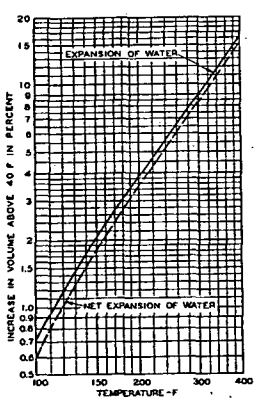
The *open tank* system is vented to the atmosphere and is generally limited to installations having operating tempera-

tures of 180 F or less because of system boiling and tank water evaporation problems. The tank should be at least 3 ft above the high point of the system and should be connected to the suction side of the pump to prevent subatmospheric system pressures caused by pump operation.[15] The system should have an internal overflow drain. Provision must be made to prevent water from freezing in the tank, the tank vent, and the pipe leading to the tank. These provisions are also required by the ASME code. The minimum tank volume should be equal to 6 percent of the total system water volume.

The *closed system* utilizes an airtight tank which provides a means of pressurizing the system for operation over a wide range of conditions. As the excess water due to thermal expansion moves into the expansion tank, it compresses the air trapped therein and increases the pressure on the system. If the tank, or the amount of air in it, is too small, the pressure on the system will exceed the maximum allowable, and cause the relief valve to waste water from the system. When the system cools, the pressure will drop to a value less than minimum, making the venting of air impossible or drawing air into the system if automatic air vent valves are located at a high point of the piping. If the tank is too large, it will cost more and require more space.

Determining Tank Size

The size of a closed expansion tank is determined by: volume of the water in the system; range of water temperatures normal to operation of the system; pressure of the air in the expansion tank when the fill water first enters the tank; relationship of the height of the boiler which usually, but not always, is the item in the system with the lowest working pressure; the expansion tank, and the high point of the system; head of the circulating pump; location of the circulating pump with respect to the expansion tank connection, and the boiler. The expansion tank size for a closed



Note: Net expansion of water equals actual expansion of water less volumetric expansion of metal system.[17]

Fig. 16. . . . Expansion of Water Above 40 F

Table 5. . . . Volume of Water in Standard Pipe and Tube

Nominal Pipe Size Inches	Standard Steel Pipe			Type L Copper Tube	
	Schedule No.	Inside Dia Inches	Gallons per Lin. Ft	Inside Dia Inches	Gallons per Lin. Ft
⅜	—	—	—	0.430	0.0075
½	40	0.622	0.0157	0.545	0.0121
⅝	—	—	—	0.666	0.0181
¾	40	0.824	0.0277	0.785	0.0251
1	40	1.049	0.0449	1.025	0.0429
1¼	40	1.380	0.0779	1.265	0.0653
1½	40	1.610	0.106	1.505	0.0924
2	40	2.067	0.174	1.985	0.161
2½	40	2.469	0.249	2.465	0.248
3	40	3.068	0.384	2.945	0.354
3½	40	3.548	0.514	3.425	0.479
4	40	4.026	0.661	3.905	0.622
5	40	5.047	1.04	4.875	0.970
6	40	6.065	1.50	5.845	1.39
8	30	8.071	2.66	7.725	2.43
10	30	10.136	4.19	9.625	3.78
12	30	12.090	5.96	11.565	5.46

system may be determined from the following ASME formula. This formula should be used only for operating temperatures between 160 F and 280 F.

$$V_t = \frac{(0.00041t - 0.0466)V_s}{\frac{P_s}{P_f} - \frac{P_s}{P_s}} \qquad (4)$$

where

V_t = *minimum* volume of the expansion tank, gallons.
V_s = system volume, gallons.
t = maximum average operating temperature, Fahrenheit.
P_s = pressure in the expansion tank when the water first enters, usually atmospheric pressure, feet of water, absolute.
P_f = initial fill or minimum pressure at tank, feet of water, absolute.
P_s = maximum operating pressure at tank, feet of water, absolute.

A widely used formula recommended for temperatures below 160 F is:

$$V_t = \frac{E}{\frac{P_s}{P_f} - \frac{P_s}{P_s}} \qquad (5)$$

where

E = net expansion of the water[16] in the system when heated from minimum temperatures to maximum temperature, gallons (See Fig. 16).

The volume of the water in a system should be determined from water capacities of heating units as given by manufacturers and the volume of pipe or tube from tables such as Table 5.

Effect of Pump Location

The required minimum pressure, P_f, and the maximum pressure in the expansion tank, P_s, may change for specific systems, depending on the effect of relative pump and tank location on system pressure changes caused by pump operation. Refer to diagrams *A*, *B*, *C*, and *D* of Fig. 17 and following discussion.

The degree of system pressure change caused by pump operation is established by the pump head. Whether the pump head will be added to, or subtracted from the system static pressure at the time of pump operation is determined by the relative pump and tank location. This is so because *the junction of the tank with the system* is a point of *no pressure change* regardless of whether or not the pump operates. Consequently, when the tank is located close to the pump suction, the pump suction pressure is unchanged when the pump starts and the pump head appears as an increase of pressure on the system. Conversely, when the tank is located close to the pump discharge, the pump discharge pressure is unchanged when the pump starts and the pump head appears as a decrease in pressure at the pump suction and on the system.

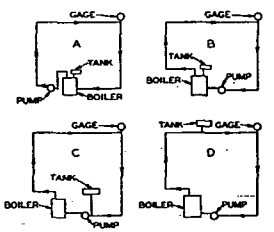


Fig. 17. . . . Effect of Pump Location and Expansion Tank Connection

When the pump discharges away from the boiler and expansion tank (*A*, Fig. 17) all pressure changes created by pump operation are additive. A gage located at any point in the system will show an increase in the pressure on starting the pump equal to the friction pressure drop between the gage and the boiler inlet (tank). Since the tank is located on the boiler, boiler pressures are unaffected by pump operation. The minimum pressure, P_f, required, is equal to height of the system above the expansion tank, plus any pressure required to assure positive air venting and *prevent boiling; if the maximum water temperature exceeds* 212 F. The maximum pressure, P_s, depends upon the pressure rating of that item of equipment having the lowest rated working pressure, usually the boiler or boiler relief valve, and the vertical distance between it and the expansion tank. This arrangement is recommended where high head pumps are used.

When the pump discharges into the boiler and expansion tank (*B*, Fig. 17), all pressure changes induced by pump operation are subtractive. A gage, located at any point in the system, will show a decrease in pressure on starting the pump equal to the pipe friction pressure drop between the gage and boiler outlet (tank). Assuming no friction loss between the discharge of the pump and the expansion tank, the minimum pressure, P_f, must be increased above that discussed for *A*, Fig. 17, by the head of the pump. The maximum pressure, P_s, is determined as described for *A*, Fig. 17. Arrangement *B* results in the largest size expansion tank, but is commonly used in residential and other small installations where water volumes, system heights, and pump heads are all low.

When the pump discharges into a boiler and the tank is connected to the pump suction line as in *C*, Fig. 17, the boiler is subjected to an increase in pressure equal to the pump head when the pump operates. The required minimum pressure, P_f, is as described for *A*, Fig. 17. The tank final operating pressure, P_s, equals the boiler working pressure, as discussed under *A*, Fig. 17, minus the pump head. With arrangement *C* the size of the expansion tank will be smaller than for arrangement *B*, but larger than for arrangement *A*.

Attic tanks are used for the purpose of tank size reduction. When used, as shown in *D*, Fig. 17, a gage located at any point between the tank and the discharge of the pump will show an increase in pressure when the pump operates, while any point between the tank and the suction line of the pump will show a decrease in pressure. This decrease in pressure on the suction side of the tank connection requires that a pressure equal to friction loss between the tank connection and the suction of the pump should be added to the vertical distance the tank is located below the high point of the system to obtain the minimum tank pressure, P_f. Since the pump is located between the expansion tank connection and the boiler, the maximum tank pressure, P_s, would be obtained as described under *C*, Fig. 17, except that pump effect on the boiler pressure *is reduced by the friction* loss between the tank connection and the boiler. If the pump were discharging from the boiler instead of into it, the pump operation would not affect P_s.

The following *Examples 4, 5,* and *6* illustrate the principles applying to sizing of closed expansion tanks.

The system used in the examples has: (1) a water volume of 1000 gal, (2) a high point of the system at the top of a return riser 25 ft from the top of the boiler directly above the boiler room, (3) a circulating pump having a 20-ft head, and a 30 psi boiler with an ASME rated relief valve as the item of equipment having the lowest pressure rating. The P_s is atmospheric pressure, 34 ft of water, absolute, in all examples. The design average water temperature is 200 F. Friction losses between the high point of the system and the boiler are assumed to be negligible.

Example 4: Determine the size of expansion tank required when connected as shown in *A*, Fig. 17.

Solution: Installation is as shown in *A*, Fig. 17, with the expansion tank at the same level as the relief valve. The net expansion of the water in the system is represented by the term $(0.00041\ t - 0.0466)V_s$ of Equation 4 and, hence, is

$$[(0.00041)(200) - 0.0466]\ (1000) = 35.4 \text{ gal.}$$

As stated, P_s is 34 ft of water, absolute. Referring to the discussion for *A*, Fig. 17, P_f, the minimum pressure, is equal to the height of the system above the expansion tank, 25 ft, plus 4 ft for positive venting, a total of 29 ft of water, gage, or 63 ft of water, absolute. P_s is equal to the relief valve setting, 30 psi × 2.31 ft of water, 69.3 ft of water, gage, or 103.3 ft of water, absolute.

Substituting in Equation 4:

$$V_t = \frac{35.4}{\frac{34}{63} - \frac{34}{103.3}} = 170 \text{ gal.}$$

Example 5: Determine the size of expansion tank required when connected as shown in *B*, Fig. 17.

Solution: All conditions remain the same, except that the expansion tank connection is at the discharge of the pump rather than at the suction. When the pump operates, the pressure at the top of the return riser is reduced by an amount equal to the head of the pump. Therefore, to maintain a positive pressure at the top of the system, the minimum pressure must be in-

creased an amount equal to the head of the pump, 20 ft of water. P_f, the minimum pressure, equals 29 plus 20, or 49 ft of water, gage, or 83 ft of water, absolute. The required volume by Equation 4 is

$$V_t = \frac{35.4}{\frac{34}{83} - \frac{34}{103.3}} = 440 \text{ gal.}$$

Example 6: Determine the size of expansion tank required when it is connected as shown in *D*, Fig. 17.

Solution: The expansion tank is located at the top of the system with the result that the height of the system has no effect on the required minimum pressure, P_f. However, the tank is located far enough from the pump so that friction loss in the piping between the pump and the tank must be considered in determining the pressure effect of pump operation on system pressures. Assume that the tank is located at a point such that the friction loss between the tank connection to the system and the suction of the pump is 8 ft, with the resistance in the return riser to the boiler still considered negligible.

The pressure in the expansion tank remains the same whether the pump runs or not. Therefore, since 8 ft of friction loss exists between the expansion tank and the pump suction at the top of the return riser, the pressure in the expansion tank must be increased by that amount to maintain a positive air venting pressure at that point.

P_f, the minimum pressure in the tank should be the sum of 4 ft for positive venting pressure and 8 ft for pump effect, or 12 ft water, gage, which is 46 ft of water, absolute. Since the pump is located between the expansion tank connection and the boiler, P_s, the maximum pressure *in the tank*, must be less than the operating pressure of the relief valve by an amount equal to the effect of pump operation on the boiler pressure. In this case, the effect of the pump is reduced by the friction loss between the expansion tank and the boiler. The head of the pump, 20 ft, less friction as discussed, 8 ft, *is* equal to 12 ft.

P_s, then, is 69.3 less 12, which is 57.3 ft of water, gage, or 91.3 ft of water, absolute. Other factors remain as discussed in the other examples. Substituting in Equation 4 the required expansion tank size is:

$$V_t = \frac{35.4}{\frac{34}{46} - \frac{34}{91.3}} = 96 \text{ gal.}$$

In *Examples 4, 5,* and *6*, the calculated size of the expansion tank has been indicated. It is customary practice to select a standard tank size not less than that calculated.

The two following important factors in tank sizing have not been illustrated in the *Examples 4 to 6*:

1. *Use of compressed air to increase the amount of air in the tank.* This factor affects the value of P_s by increasing it above atmospheric pressure. The use of compressed air to *charge* the expansion tank reduces the required tank volume.

2. *Effect of a design maximum temperature higher than 212 F.* The effect of design temperatures above 212 as used by some designers is given in the discussion of *A*, Fig. 17. It is possible to obtain proper operation of the expansion tank without adding the pressure required to maintain liquid water at temperatures in excess of 212 F. Two things are necessary on this basis of design: (*a*) the pressure increase in the system as the temperature increases from 40 F to 212 F and to the final design temperature must be sufficient to keep the pressure above that corresponding to the temperature of the water in order to prevent boiling and (*b*) the operating personnel for the system must be instructed regarding the proper pressure for various system temperatures in order that proper pressure control of the system may be maintained. An automatic fill valve cannot be used to maintain the minimum pressure for such a system.

In small systems where the volume of water is small and where the pressures due to pump and static heads are low, expansion tanks are small and no critical problems are encountered. However, for large systems the size of the expansion tank and the problems of pressure control can become critical. The following means of reducing expansion tank size and establishing satisfactory pressure control may be investigated in addition to the methods discussed under *A* to *D*, Fig. 17:

1. Compressed air, if available, can be admitted to the expansion tank to obtain the desired pressure.

2. The head of the pump may be reduced by the use of larger pipe sizes, or greater design temperature drop.

3. The boiler may be constructed for pressures higher than 30 psig.

4. The steam boiler may be used with a steam-to-water heat exchanger to heat the water circulated.

The point of connection of the expansion tank to the system should be studied carefully to guard against the possibility of isolating the tank from a hot boiler or any part of the system by normal operation of automatic, check, or manual valves.

BOILER ROOM PIPING

The arrangement of equipment, its capacity, and the piping arrangement for its connection will vary, depending upon the size and characteristics of the systems which it serves. Most of the principles involved in these arrangements have been discussed. Fig. 18 shows the piping and specialties considered necessary for the most basic system. The *ASME Low Pressure Heating Boiler Code* requires that provision must be made for draining a closed expansion tank without emptying the system. A valve is commonly included in the expansion piping to the tank but it must not close a connection to the safety relief valve. Fig. 19 shows a more elaborate multizone system. Most systems will fall between these extremes as the engineer fits the design to the requirements of the installation.

One device in boiler piping which facilitates elimination of air before it can enter the mains is the dip tube. Air will separate from the water in a heating system at a point where the velocity and pressure are the lowest and where the temperature is the highest. Although the pressure at the boiler is generally high, it is a logical place to perform the separation. Fig. 18 shows the use of a dip tube. This extension of the outlet pipe forms a dead space at the highest and hottest part of the boiler. The air will tend to separate from the

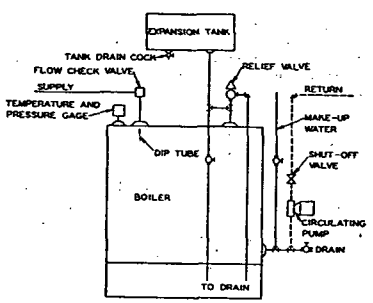


Fig. 18. . . . Basic Boiler Connections for a Hot Water System

water in this space. The air can be conducted to the expansion tank by a connection from the tank to the highest point of the boiler (see Fig. 18). The connecting pipe should be as direct as possible and be pitched upward continuously to the tank. If the expansion tank is not connected directly to the boiler, an automatic air vent valve should be installed at the high point of the boiler, instead.

In general, the manufacturer's recommendation on the number of outlets to be used for a particular boiler for a given application should be followed. This applies also to return connections.

The prevention of thermal shock through proper use of return connections should be considered in piping arrangement. Where more than one pump are used for circulation, the discharge of the pumps should be joined before being discharged into the boiler. A check valve should be installed in the discharge of each pump, unless another method of preventing backflow through the pump circuit is provided. Where temperatures higher than normal are maintained in the boiler, the connection of the return to the boiler should follow the manufacturer's recommendations for such service.

Where a multiple boiler installation is used, provisions should be made to prevent the bypassing of water through a boiler which is not being maintained at control temperature. In some cases this provision takes the form of automatic control valves interlocked with the boiler temperature control to be open only when the temperature in the particular boiler is being maintained at operating conditions. In other cases, manual valves are installed in the boiler returns and opened only when the burner switch which places the firing device at the command of the boiler temperature control is in the *ON* position.

Weighted check valves, or flow check valves, should be installed in the supply piping from the boiler to prevent gravity circulation during periods when the circulating pumps are not operating. If the piping is arranged with separate supplies to individual zones, a weighted check valve should be installed in the supply piping to each zone.

THE USE OF STEAM WITH HOT WATER HEATING SYSTEMS

The use of a steam boiler or district steam with a converter (heat exchanger) has many advantages for tall buildings. The piping for the steam used in this type of a system should follow the principles outlined in Chapter 26, Steam Heating Systems. In tall buildings the system can be zoned vertically as shown in Fig. 20, that is, with a given number of floors per zone and a converter for each zone. This type of design makes the limitation of maximum pressure in the system possible. With a converter-type system, the volume of water in the boiler is not a part of the volume of water causing thermal expansion in the system. Since the boiler in a hot water system may hold 30 to 50 percent of the total water volume of the system, the reduction in the size of the expansion tank and the problems involved therewith are of major importance.

The use of steam in extended-surface coils for heating air will often be more desirable in a system where steam is available. This type of installation will result in less expensive coils, and simplified coil controls.

TEMPERATURE CONTROL

Methods of temperature control for hot water heating systems, including zone controls, are covered in Chapter 43, Automatic Control.

SYSTEM ADJUSTMENT

Two common methods of determining the effects of capacity adjustment are: the measurement of space tempera-

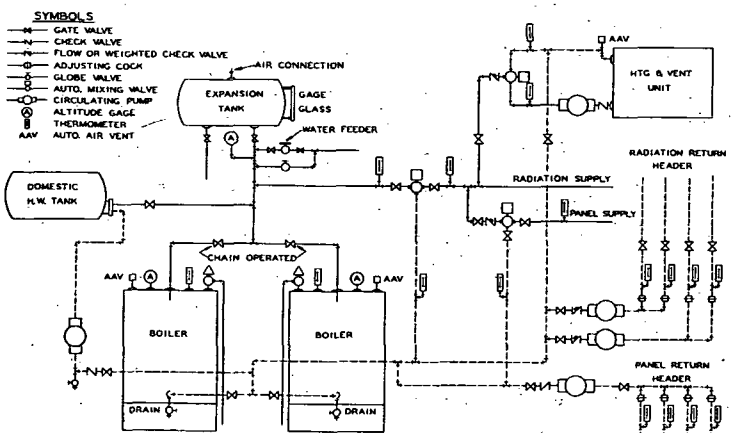
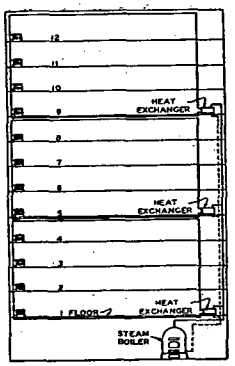


Fig. 19. . . . Boiler Piping for Multiple-Zone Multiple-Purpose Heating System

tures by room thermometers, and the determination of the temperature drop of water flowing through a heating unit or zone by means of surface contact thermometers or thermometers installed in the piping. When adjustment is made by means of water temperature drop, the capacity of the units after adjustment must equal the heat delivery actually required. Control is then accomplished by making the temperature drop through all units equal.



(Provision should be made for expansion in each closed circuit)

Fig. 20. . . . Vertical Zoning of Hot Water Heating System in a 12-Story Building

The procedure for making a permanent adjustment of heat distribution in a larger system should be as follows:

1. Select a time or day when conditions of heat gain are minimum; i.e., nighttime or an overcast day. Outdoor temperatures should be such as to require at least 50 percent of the system capacity to maintain the design indoor temperature.

2. Place the system in operation and make certain that all valves, adjusting fittings, and dampers are in the open position. Automatic control which might reduce the flow or capacity of any unit should be rendered inoperative. Doors and openings between rooms should be closed.

3. Prepare a form for recording data on temperatures in spaces or at the supply and return of heating elements or of zones.

4. After the system has reached equilibrium a complete record should be made of temperatures throughout the system.

5. An initial adjustment of flow regulating devices in the system should be made on the basis of the record of the original readings. A new set of temperature readings should be recorded after sufficient time has been allowed to establish a new equilibrium throughout the building.

6. Continue adjustments of flow control devices until a satisfactory condition is obtained.

7. When a satisfactory adjustment has been accomplished, it is advisable to *mark the position of each of the adjusting fittings or valves.* This facilitates return to proper control settings if the flow controls are disturbed by accidental or emergency changes at any time.

CARE AND MAINTENANCE OF SYSTEMS

A hot water heating system should last during the life of the building if it is designed, installed, and maintained properly. Maintenance instructions for individual elements of the system such as burners, motors, pumps, and accessories may be obtained from the manufacturers who supply this equipment. Two factors contribute in a most important way to the satisfactory operation and life of the hot water heating system: (1) the proper cleaning of the system when installed; and (2) a minimum change of water in the system, except as required by periodic maintenance of the boiler or draining of the expansion tank.

INITIAL CLEANING OF SYSTEM

Rules that should be followed for the initial cleaning of the system are:

1. All equipment and piping should be thoroughly cleaned of iron cuttings and other refuse during assembly and installation.

2. When installation is complete, the system should be filled with a solution of 1 lb of trisodium phosphate per 50 gal of water and should be operated for 24 hr at maximum temperature with all pumps operating. The system should then be drained and thoroughly flushed with water before refilling.

In order to minimize the addition of water to the heating system, the operation of air-vent valves, relief valves, and the tightness of the system generally should be subject to continuous inspection. Drips from automatic air-vent valves should discharge in places where leakage is readily detected. Continual operation of the pressure relief valve should be a signal for the inspection, repair, or replacement of the relief valve, or a check on the operation of the expansion tank or the automatic water feeder, if one is provided.

REFERENCES

[1] J. H. Keenan and F. G. Keyes: *Thermodynamic Properties of Steam* (John Wiley & Sons, New York, 1936, 1st ed.).

[2] F. E. Giesecke and J. S. Hopper: Friction heads in standard six-inch pipe (ASHVE TRANSACTIONS, Vol. 47, 1941, p. 71).

[3] F. E. Giesecke and J. S. Hopper: Comparative study of friction heads in screwed and welded elbows (ASHVE TRANSACTIONS, Vol. 48, 1942, p. 201).

[4] F. E. Giesecke: Friction heads due to water flow in copper, brass, and other smooth pipes (ASHVE TRANSACTIONS, Vol. 49, 1943, p. 175).

[5] F. E. Giesecke: Determination of pipe sizes for hot water heating systems (ASHVE TRANSACTIONS, Vol. 21, 1915, p. 473).

[6] F. E. Giesecke: Friction of water in iron pipes and elbows (ASHVE TRANSACTIONS, Vol. 23, 1917, p. 499).

[7] F. E. Giesecke: Effect of temperature upon the friction of water in pipes (ASHVE TRANSACTIONS, Vol. 31, 1925, p. 9).

[8] F. E. Giesecke: Friction of water in elbows (ASHVE TRANSACTIONS, Vol. 32, 1926, p. 303).

[9] F. E. Giesecke and W. H. Badgett: Friction heads in one-inch standard cast-iron tees (ASHVE TRANSACTIONS, Vol. 37, 1931, p. 395).

[10] F. E. Giesecke and W. H. Badgett: Loss of head in copper pipe and fittings (ASHVE TRANSACTIONS, Vol. 38, 1932, p. 529).

[11] F. E. Giesecke and W. H. Badgett: Supplementary friction heads in one-inch cast-iron tees (ASHVE TRANSACTIONS, Vol. 38, 1932, p. 111).

[12] F. E. Giesecke: Two methods of figuring the friction loss in pipe lines (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 49).

[13] L. N. Montgomery and W. S. Harris: Sources of vent gas in a hot water heating system (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 483).

[14] Low-pressure heating boilers (*ASME Boiler and Pressure Vessel Code*, with 1954 and 1955 addenda, Section IV).

[15] R. C. Chewning and R. W. Peterson: Consider pump heads when sizing hot water expansion tanks (*Heating, Piping and Air Conditioning*, July 1955, p. 106).

[16] H. A. Lockhart and G. F. Carlson: Compression tank selection for hot water heating systems (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 55).

[17] Oscar Faber and J. R. Kell: *Heating and Air Conditioning of Buildings* (Architectural Press, London).

CHAPTER 29

HIGH-TEMPERATURE WATER SYSTEMS

Features; Basic System; Design Considerations: Boilers, Boiler Piping and Controls, Direct-Contact Heaters and Auxiliary Heat Exchangers, Boiler Feed Pumps, Circulating Pumps; Distribution Piping Design: District Distribution, Pipe, Valves and Fittings, Thermometers and Pressure Gages; Space Heating Equipment; Temperature Control; Water Treatment; Storage

HIGH-TEMPERATURE water systems discussed in this chapter are those operating with supply water at temperatures exceeding 250 F. Operating temperatures range from 250 to 430 F, with pressures from 55 to 350 psig. High-temperature water systems used in the United States are predominantly closed-type systems.

This chapter is intended to give a broad general idea of principles and practices that apply to high-temperature systems and distinguish these systems from the systems operating below 250 F.

FEATURES OF SYSTEMS

The following are among the outstanding features of high temperature as compared to low temperature systems:

1. It is common practice to use greater temperature drops.

2. Supply and return piping may be given the same pitch or grade, or may be run level. Grading is desirable but a definite minimum pitch is not required.

3. Piping may slope up or down or run at a variety of elevations to suit the terrain and the architectural and structural requirements without provision for drainage at each low point except for emptying sections of piping for shutdown. This feature may reduce the excavations required.

4. The pressure in any part of the system must always be above the pressure corresponding to the temperature at saturation in the system in order to prevent flashing of the water into steam.

5. Processes requiring different temperatures of water may be served at their required temperatures by use of heat exchangers, by regulating the flow to certain processes, by placing some processes in series with others, etc.

6. Heat may be stored in the mains and boiler by build-up of temperature in the return main during periods of light load.

THE BASIC SYSTEM

High-temperature water systems are basically similar to the conventional forced hot water heating systems since they require boilers or direct contact heaters for heating the water, expansion or pressurizing tanks or both, circulating pumps, distribution piping, and heat transfer or emission equipment, as shown in Fig. 1. The principal differences from the low temperature system are the higher pressure used, the consequently heavier equipment, and the manner in which pressure is maintained on the water. When cushioned by steam in the boiler, the system in effect is a hot water system using a boiler producing steam to heat the water. The supply piping draws water from the boiler below its water level so that water flow may occur even though steam is kept above the surface of the water. For an equivalent heat load the rate of water flow through the boiler is 6 to 10 times that for steam systems.

Water is circulated in a closed circuit from the central station to areas where heat is required and back to the boiler.

The size of installation, the extent of the load, and the operating temperature will govern central boiler-plant design and construction. Some determining considerations are:

1. Type of load: heating, process, or both.

2. Distance from heating plant to space or process requiring heat.

3. Terrain of land on which buildings are located.

4. Zoning requirements based on occupancy and load distribution.

5. Quantity of steam used for power equipment, if required.

Variations in elevation may make it desirable to locate the boiler house on the highest ground in order to obtain maximum static pressure head on the system. Boiler auxiliaries such as boiler feed pumps, pressure tanks, economizers, and system circulating pumps are usually located in the boiler house, as indicated in Fig. 2.

DESIGN CONSIDERATIONS

Factors to be considered in selecting the type of boiler are (1) the maximum operating pressure of the system, (2) the water temperature requirements during a 24-hr operating period, and (3) the size of the load. Process loads may require water at a given minimum supply temperature continuously, while space heating and other loads may permit a lower water temperature at night or an off period during each 24-hr operating period.

Water temperature requirements also influence the manner of pressurizing the system. Simplicity of operation is an important consideration in selecting a method of pressurizing when boilers are not operated during the night. Variations in supply water temperature and temperature drop also should be taken into account in selecting a pressurizing method.

Theoretically, water temperatures up to about 350 F may be provided using boilers and piping suitable for 125 psig, but practically, maximum water temperatures will be limited

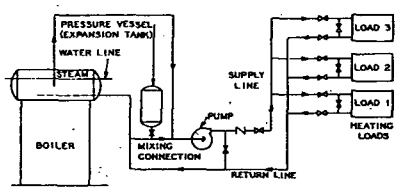


Fig. 1. . . . Elements of a High-Temperature Water System

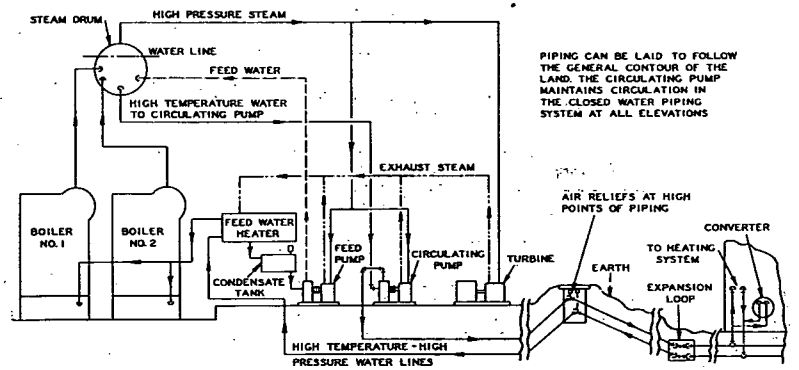


Fig. 2 .... Profile of High-Temperature Water System

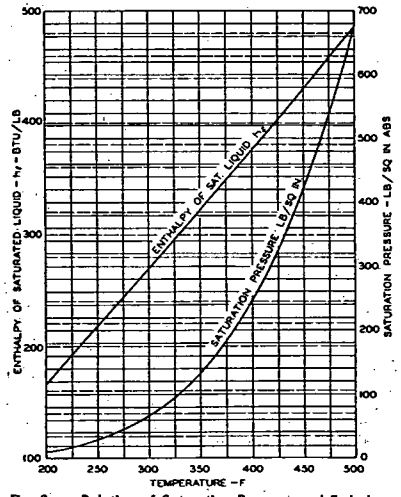


Fig. 3 .... Relation of Saturation Pressure and Enthalpy to Water Temperature

Table 1 .... Properties of Water (212 F to 400 F)

Temperature F	Pressure* Psia	Density Lb/Cu Ft	Specific Heat Btu/(lb)(F)	Total Heat Above 32 F	
				Btu/Lb*	Btu/Cu Ft
212	14.70	59.81	1.007	180.07	10,770
220	17.19	59.63	1.009	188.13	11,216
230	20.78	59.38	1.010	198.23	11,770
240	24.97	59.10	1.012	208.34	12,313
250	29.83	58.82	1.015	218.48	12,851
260	35.43	58.51	1.017	228.64	13,378
270	41.86	58.24	1.020	238.84	13,910
280	49.20	57.94	1.022	249.06	14,430
290	57.56	57.64	1.025	259.31	14,947
300	67.01	57.31	1.032	269.59	15,450
310	77.68	56.98	1.035	279.92	15,950
320	89.66	56.66	1.040	290.28	16,437
330	103.06	56.31	1.042	300.68	16,931
340	118.01	55.96	1.047	311.13	17,409
350	134.63	55.59	1.052	321.63	17,879
360	153.04	55.22	1.057	332.18	18,343
370	173.37	54.85	1.062	342.79	18,802
380	195.77	54.47	1.070	353.45	19,252
390	220.37	54.05	1.077	364.17	19,681
400	247.31	53.65	1.085	374.97	20,117

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by the system design and elevation characteristics to values between 300 F to 325 F. Most systems are being designed with the system circulating pump in the supply line and with a steam drum for pressurization, expansion, and temperature control. In this type of system, the temperature of the water from the steam drum cannot exceed the temperature of the steam in the drum which corresponds to its pressure at saturation. The point of maximum pressure is at the discharge of the circulating pump. If this pressure is to be maintained below 125 psig, the pressure in the drum, corresponding to the water temperature, cannot exceed 125 psig minus the sum of the pump head and the pressure head due to the difference in elevation between the drum and the circulating pump.

Elevation effects and the pressures required to prevent the flashing of water to steam in the supply system can also limit the maximum water temperature that may be used and must, therefore, be studied in evaluating the temperature-pressure relationships of the system.

The properties of water which govern the features of the design are:

1. Temperature vs pressure at saturation (see Fig. 3).
2. Density or specific volume (v_s) vs temperature.
3. Enthalpy or sensible heat (h_f) vs temperature.
4. Viscosity vs temperature.

The relation of temperature and pressure, specific volume, and enthalpy are all discussed in Chapter 3. Data for the normal temperature-pressure ranges are given in Table 3 of Chapter 3. The properties of water for the range 212 F to 400 F are shown in Table 1.

Water temperatures of 350 to 400 F requiring 250 psig working pressure will largely be limited to district and institutional installations. In such applications the heat quantities and distances to be covered usually justify the added cost of the heavier pipe, fittings and special valves.

Boilers

The boiler plants of the larger high-temperature water systems are comparable to boiler plants of power generating stations operating within the same pressure range. The boilers should be selected for size and type in keeping with the load and design pressures. Both steam for power or processing and high-temperature water for heating may be supplied from the same boiler plant. The load is determined by the heat required for:

1. Space heating and winter air conditioning.
2. Generated steam required for power.
3. Water heating or heat processes.

The boilers may be water-tube, horizontal return-tube or Scotch marine type, and may be equipped with any conventional fuel-firing apparatus. Water-tube boilers are favored for higher pressure ranges—200 psig and upward. Boilers may be of either forced circulation or gravity circulation type. The circulating pump of forced circulation boilers must operate continuously during operation of the boiler. Water-tube boilers frequently require separate vessels for the pressure cushions, depending on the volume of the steam drum relative to system requirements. When systems using fire-tube boilers are cushioned by steam, expansion space may be available within the boiler itself.[1] A separate vessel usually is used when the system is cushioned by air or other gas.

Proper distribution of return water and of water flow is essential in all types of boilers to prevent tube failures due to overheating or unequal expansion in boiler tubes.

Expansion Pressure Cushion

Four methods are used to produce and maintain the necessary pressures on the system:

1. A steam cushion in the steam space of the boiler or a steam drum.
2. A separate expansion tank for steam pressurizing.
3. Compression tank with compressed air or an inert gas.
4. An automatic pressure pump.[2, 3, 4]

A steam cushion or a pneumatic cushion is generally favored because it may also accommodate the expansion. The *steam cushion* method for maintaining pressure above the water level in the boiler requires space to accommodate the expansion of the water volume within the system when heated to the system operating temperature. Systems which do not use steam for maintaining the required pressure, or in which the boiler steam space is inadequate to accommodate the change in water volume, will require additional tanks or steam drums for expansion.

The arrangement of the tanks for pneumatic cushioning is similar to the closed expansion tanks when air pressures are used for cushioning or pressurizing. A suitable means of balancing or a connection from the bottom of the tank should be made to the main system return header. (See Fig. 4.)

A major problem when pressurizing by steam is to maintain a proper boiler water level while providing for the change in water volume due to expansion and contraction. Boiler water level and flow connections should be designed to utilize the steam drum in adjusting to changes in water

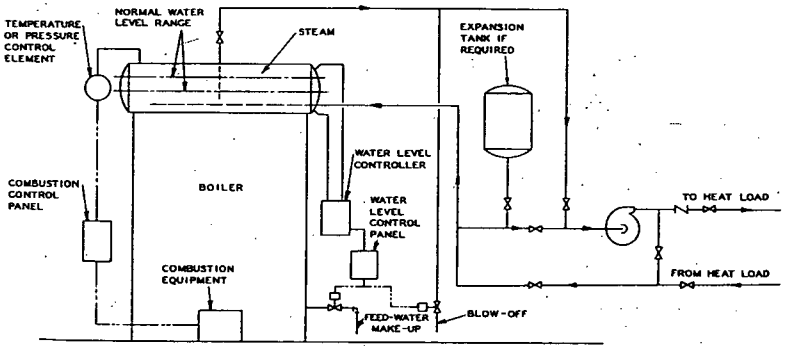


Fig. 4 .... Schematic Arrangement of Combustion and Feed-Water Controls

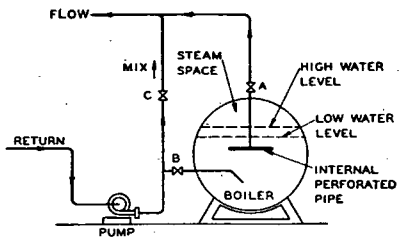


Fig. 5 .... Arrangement of Boiler Piping

volume. The problem of change in water expansion and contraction is minimized by rarely closing down a large system after it is in operation.

When pressurizing with a boiler feed pump automatically operated by pressure control, a relief valve for the maximum system pressure is provided. The pressure control is set to operate the boiler feed pump and feed water from the make-up tank to the boiler whenever the pressure falls to the regulator *cut-in point* setting, and stop when the *cut-out point* is reached. If the pressure increases beyond the safe limit of relief valve setting, excess water will flow through the relief valve to the make-up water tank. Pressurizing may also be done by a continuously operating boiler feed pump.

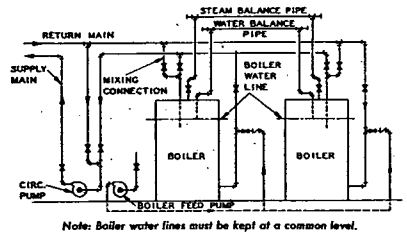


Fig. 6 .... Piping Connections for Two or More Boilers in High-Temperature Water Systems Pressurized by Steam

Boiler Piping

When the circulation pumps discharge to the system supply piping, the flow main should not rise above the boiler water line high enough to reduce the total pressure sufficiently to permit flashing in the piping. Whether the pump is connected so as to deliver water to the system supply piping or to the boiler inlet line, the arrangement is a complete circuit and the water is returned to the boiler for reheating. The returned water is brought into the boiler below the water line as shown in Fig. 5.[8]

When two or more boilers are used to supply a common load, means must be provided to equalize the flow of water to each boiler (see Fig. 6). Automatic water control is desirable. Balance pipes to maintain equal pressures must be liberal in size as shown in the following table.

Boiler Rating Millions Btuh	Balance Pipe Dia, Inches
2.5	3
5	3½
10	4
15	5
20	6
30	8

Valves should be used in connections to individual boilers when more than one boiler is required as in power plant practice.

Natural or forced circulation may be used to produce internal boiler flow. Internal baffles or distribution pipes are arranged to prevent stratification and promote circulation in the boilers. Arrangements for this purpose vary with the

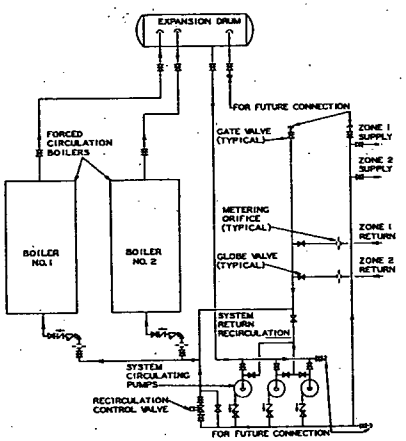


Fig. 7 .... High-Temperature Water Piping for Combined Pump System

various designs. In natural circulation, convection currents and the system circulating pump circulate the water through the boiler. Fig. 7 shows the piping arrangement for the system which is also known as the combined pump system. This method should be used only with boilers having a comparatively low pumping head loss.[8]

In forced circulation boilers a circulating pump takes the water from the steam drum and discharges it back into the boiler, the proper water path being obtained by suitable boiler design. This is referred to as the separate pump system because the boiler circulating pump is separate from the distribution system pumps. One pump is normally used for each boiler to draw water from the expansion drum, circulate it through the boiler and return it to the expansion drum (Fig. 8). The system circulating pumps draw water from the expansion drum and circulate it through the distribution system only. The system return is connected to the suction side of boiler circulating pumps or to the expansion drum. This method permits the use of boilers with either high or low head loss.

With steam pressurizing, the water line must be held within proper limits to always provide a water seal at the entrance to the supply flow pipe. Otherwise, flashing or steam bubbling could occur in the supply pipe as the water level rises and falls.

Boiler Controls

Some of the automatic controls indicated in Fig. 4 and extensively used for operating boiler plants are:

1. Automatic combustion and safety control: control of fuel and combustion with flow changes.
2. Water level control: control of make-up and shutdown on low water.
3. Safety valve (steam), automatic relief valve (hot water): on large water systems small additional relief valves should be provided to relieve minor excess pressures.

On installations large enough for high-temperature water heating to be economic, automatic control is usually a worthwhile investment. When package type boilers are used, automatic combustion control and water level control are usually components of the boiler package.

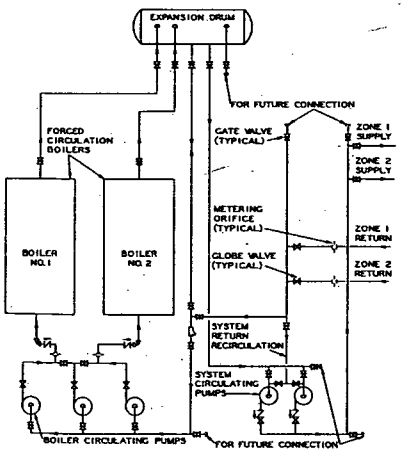


Fig. 8 .... High-Temperature Water Piping for Separate Pump System

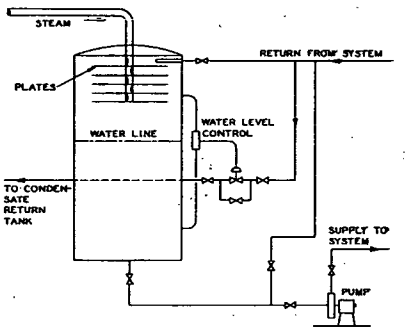


Fig. 9 .... Direct Contact Heater for Generating High-Temperature Water

Direct-Contact Heaters and Auxiliary Heat Exchangers

High-temperature water may be obtained from direct-contact heaters in which steam from turbine exhaust, extractors, or steam boilers is mixed with return water from the system (see Fig. 9). The mixture takes place in the upper part of the heater where the water cascading from horizontal baffles comes in direct contact with steam.

The steam in the upper part of the heater serves as the steam cushion for pressurizing the system. The lower part of the heater serves as the expansion tank for the system.

Where the water heater and the boiler operate under the same pressure, the surplus water is usually returned to the boiler through a pipeline connecting the outlet of the high-temperature water circulating pump to the boiler. This

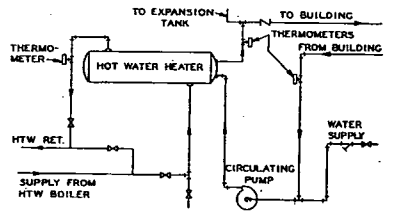


Fig. 10 .... High-Temperature Heat Exchanger Heating Water for Low-Temperature Heating System

system is also applicable where mixed steam and high-temperature water services are required.[1]

Fig. 10 shows a method by which high-temperature water in a heat exchanger heats water for a comfort heating system which uses supply water temperatures of 180 F to 240 F. With high-temperature water at 300 to 350 F a temperature differential between entering and leaving water of approximately 100 deg may be used. Automatic control of such generators or converters is preferable and is discussed in a later section, Temperature Control.

Fig. 11 shows a tubular heat exchanger in which low- or medium-pressure steam is produced by heat transferred from high-temperature water circulated through the tubes. In order to produce the required quantity of steam the mean water temperature in the tubes may need to be within 20 to 30 deg of the original temperature of water in the high temperature system.

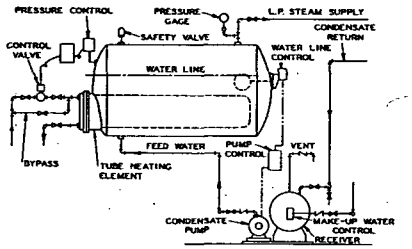


Fig. 11 .... Heat Exchanger Used as Steam Generator

Boiler Feed Pumps

Boiler feed pumps should be provided for supplying make-up water to the boilers where local pressures are, or may be, below the boiler pressure. If the boiler plant supplies steam for heating only, the pump sizes will be small. Boiler feed pumps should, however, have reserve capacity and should have a hot well or receiver whose water level is controlled by automatic means. When steam is used for power purposes, boiler feed pumps must not only return condensate but also supply make-up water to replace wasted condensate.

Circulating Pumps

Circulating pumps for temperatures above 240 F should be built for continuous operation at high temperature. The preferable types have bearings and stuffing boxes that are water cooled. The bed plate and mountings should be designed to permit expansion and contraction without causing misalignment. Since continuity of service depends upon the circulating pumps, they should be installed in duplicate to provide standby service.

It is good practice to locate the circulating pump in the supply line in order to maintain all parts of the distributing system at pressures exceeding boiler pressure. This location is also preferred since water in the system would flash into steam if the pressure in the piping decreased below the pressure corresponding to the temperature of the water at saturation. Connections which permit some return water to bypass the boiler and enter the supply line as a means of preventing flash steam in the flow main are very important.

Sufficient net positive suction head (NPSH) should be available at the pump suction flange. Proper design of the pump piping in this respect will avoid *cavitation*.[2] In selecting a pump its pumping head and water circulation rate must be carefully evaluated in order to permit selection of the proper driving motor. The effect of density variation between the high and low temperature must be considered. Pump characteristics as affected by system characteristics, should be given consideration to insure nonoverloading conditions in pump motor operation.

DISTRIBUTION PIPING DESIGN

The arrangement of piping and necessity for zoning will be determined by such factors as the hours of operation, the nature of the space occupancy, and the required water temperature. These factors, together with the heat requirements of the equipment served, will establish the temperature drop which, in common practice, may range from 80 to 150 deg.

Economic considerations require that the savings due to use of small pipe and high velocity circulation be balanced against the cost of valves, pumps, etc., that are suitable for operation at the high friction loss and pressure. The best overall investment will be secured when there is a proper relation between first cost, operating cost (including that of the operating personnel), and the fixed charges.

Fig. 12 shows the relation (based on the Fanning formula in Equation 8 of Chapter 4) of the water flow in thousands of pounds per hour to the friction loss in feet of 62 F water at a mean temperature of 300 F in Schedule 40 commercial steel pipe in sizes ½ to 12 in. The temperature 300 F is a normal mean between flow and return temperatures in the system. Correction of friction loss for other temperatures may be obtained from Fig. 13. Allowance for friction loss in fittings may be obtained from Table 2.

District Distribution

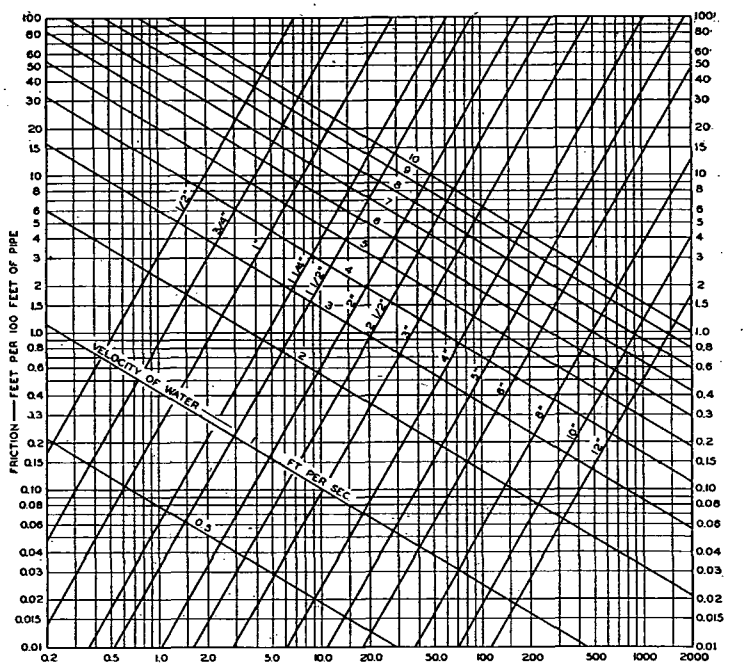
The conventional conduit or tunnel distribution systems are employed with similar techniques used for installation. Although the mains may be run at different elevations at will, it is important that high points be vented and the low points drained by the air collection and venting arrangement described earlier. Grading is not critical. See Chapter 27, District Heating.

The piping serving each building or process load should have a valved bypass connection between the supply and the return pipe in addition to the usual stop valves as indicated in Fig. 1.

Pipe, Valves, and Fittings

All pipe, valves, and fittings used in high-temperature systems should comply with the requirements of the *American Standard Code for Pressure Piping*, ASA B 31.1—1955. This code states that hot water systems shall be designed for the highest pressure and temperature actually existing in the piping under normal operation. This pressure is the cushion pressure plus pump head plus static pressure.

Individual heating equipment units should preferably be installed with a lock-shield valve for balancing the flow and a separate valve for shut-off. These may be placed in the most convenient position for accessibility. If the unit is to be isolated for service, valves will be needed in both the supply and return piping to the unit. Valve trim should be of stainless steel or other alloy that will prevent corrosion.



(Friction given in feet of 62 F water per 100 ft of pipe.)

Fig. 12 .... Friction Loss for 300 F Water in Commercial Steel Pipe (Schedule 40)

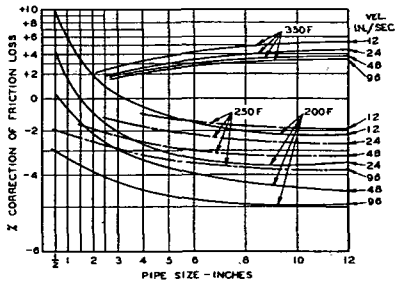
Thermostatically operated valves in the control equipment should be of the electric-motor-operated or pneumatic type. Their selection is discussed in a later section, Temperature Control. (See also Chapter 43, Automatic Control.)

High points in piping should be provided with chambers and air vents for collecting and removing air and low points should have provision for sludge removal, as illustrated in Fig. 14.

Loop-type expansion joints, in which the expansion is absorbed by deflection of the pipe loop, are preferable to the packed-metal type. If mechanical expansion joints are used, they should be of the guided type.

As high-temperature water is more penetrating than low-temperature water, leakage due partly to capillary action should not be ignored because even a small amount of leakage vaporizes immediately. This slight leakage becomes noticeable only on the outside of the gland and stem of the valve where thin deposits of salt are left after evaporation.

The insulation of high-temperature water lines is similar to that of steam lines.[3] The selection of the economical thickness of pipe insulation is discussed in Chapter 32.



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Fig. 13 .... Correction of 300 F Friction Loss for Other Water Temperatures

Thermometers and Pressure Gages

Pressure gages should be installed in the pump discharge and suction and at locations where pressure readings will assist operation and maintenance. Thermometers (preferably dial type) or thermometer wells should be installed in the flow and return pipes, the pump suction and discharge, and at any other points of major temperature change or where temperatures are of importance in operating the system. Thermometers and gages are desirable in the piping at the entrance to each building, hot water converter, or steam converter. (See Chapter 44.)

SPACE HEATING EQUIPMENT

In industrial areas, space heating equipment may often be operated with high-temperature water due to its availability. Convectors and radiators may require water temperatures in the low and medium temperature range (120 to 180 F or 200 to 250 F) depending upon how close they are to the occupants and upon the pressure for which they are designed. The water velocity through the heating equipment affects its capacity. This must be taken into account in selecting the equipment because, if a large water-temperature drop is used, the rate of circulation is reduced and consequently the velocity of flow may be reduced sufficiently to lower appreciably the rate of heat transfer. (See Chapter 5.)

TEMPERATURE CONTROL

The principles discussed under Control of Flow in Chapter 43 apply to high-temperature water systems, but added consideration should be given to the selection and location of control components.

Valve Selection

Valve selection and sizing are very important because of the relatively high temperature drops and smaller flows encountered in high-temperature water systems. To obtain good control, the valve must be sized so that it is effective over its full range of stem travel. It is recommended that the valve size be selected on the basis of a pressure drop at full flow equal to approximately 50 percent of the pressure difference between supply and return mains at the location of the equipment served. The pressure drop across the valve must not be sufficient to result in a downstream pressure below the saturation pressure at the temperature existing at any point or flashing into steam will result. A valve with equal percentage flow characteristics is desirable. Sometimes two small valves are used rather than one large valve to improve the controllability of the system. Stainless steel trim is recommended, and all valve body materials and packing should be suitable for the high temperatures and pressures encountered. The valve should have a close-off rating at least equal to the maximum head produced by the circulating pump. Generally, two-way valves are more desirable than the three-way type because of the equal percentage flow characteristics and the smaller capacities available in two-way valves. Single-seated valves are recommended in preference to double-seated valves since the latter do not provide tight closing.

Table 2.... Head Loss in Fittings[a]
(For turbulent flow)

Fittings	Pipe Size, Inches: Feet of Same Size Pipe Having Equal Loss[a]												
	10	8	6	5	4	3	2½	2	1½	1¼	1	¾	½
Globe Valve	340	270	200	170	135	105	83	70	55	45	36	28	21
Gate Valve	7	5.75	4.5	3.5	3	2.2	1.7	1.5	1.2	1	0.8	0.6	0.5
90° Welding Elbow	11	8.5	6.5	5.5	4.5	3.5	2.6	2.2	1.8	1.5	1.2	0.9	0.7
45° Welding Elbow	7	6	4.5	3.6	3	2.4	1.7	1.5	1.2	1	0.8	0.6	0.5
22½° Welding Elbow	5	4	3	2.5	2	1.6	1.3	1.1	0.9	0.7	0.6	0.5	0.3
Lateral[b]	26	20	16	13	11	8	6.5	5.5	4	3.5	2.8	2.2	1.6
Lateral[c]	17	14	10	8.5	7	5	4	3.5	2.8	2.5	1.8	1.5	1.1
Welding Tee[c]	13	10	8	7	5.5	4	3.2	2.6	2	1.7	1.4	1.2	0.8

Concentric Reducers[d] (From \ to)	10	8	6	5	4	3	2½	2	1½	1¼	1	¾	½
10		1	2.5	3	2.8								
8	3.5		1.2	2	2.2	2.2			CONTRACTING				
6	10	3.5		0.5	1.3	1.8	1.7						
5	12	7.5	1.5		0.5	1.2	1.4	1.5					
4	12	9.5	4.5	1.5		0.6	1.2	1.4	1.5				
3		10	7	5	1.8		0	0.8	1	1			
2½			7.5	6	3.8	0.7		0	0.7	0.8	0.75		
2				6	5	2.5	0.8		0	0.6	0.6	0.6	
1½					5	3.8	2.8	9		0	0.5	0.5	0.5
1¼						4	3	2	0.3		0	0	0.3
1	ENLARGING						3	2.5	1.25	0.5		0	0.3
¾								2.5	2	1.3	0.5		0
½									2	1.8	1.5	0.8	

[a] Compiled from data published by *Tube Turns*, Inc., a Division of the National Cylinder Gas Co. Used by permission.
[b] Water deflected through branch.
[c] Water flowing straight through fitting.
[d] For eccentric reducers, the values read for concentric reducers must be increased by 33⅓%.
[e] Equivalent length of pipe having a Fanning friction coefficient of 0.008.

Valve Location

Control valves preferably should be located in the return lines from heat transfer units in order to reduce the valve operating temperature and to prevent the high-temperature water from flashing to steam at the lower discharge pressure. Sometimes valves are located as low as practicable to increase the static head at the valve and further minimize the possibility of flashing.

Typical Applications

A typical application is the control of the temperature of water leaving a heat exchanger in which high-temperature water is used as the heating medium. The temperature measuring element should be located where it can best detect changes and thus prevent overheating of the outlet water and possible flashing. When the measuring element is located in the leaving water, there must be a continuous flow through the exchanger and past the element. If the water leaving the exchanger is used for space heating, the set point of the thermostat in this water (which controls the temperature of the leaving water by controlling the inlet temperature to the exchanger) can be readjusted in accordance with outdoor temperature.

Another typical application is the control of a low- or medium-pressure steam generator, usually less than 50 psig, using high-temperature water as the source of heat. In this application a proportional pressure controller in the steam generator supply positions a high-temperature water valve located in the return line to maintain the desired steam pressure.

High-temperature water is sometimes used to supply coils in air handling units where heating surfaces are so placed that possibility of personal contact is avoided. Careful sizing of coil and valve is required to obtain stable control. In some cases face and bypass damper control is used,

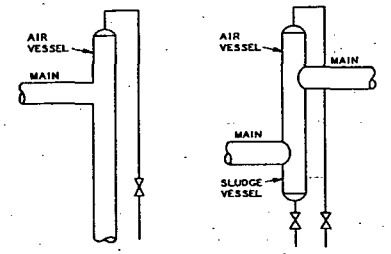


Fig. 14.... Methods of Venting High-Temperature Water Mains

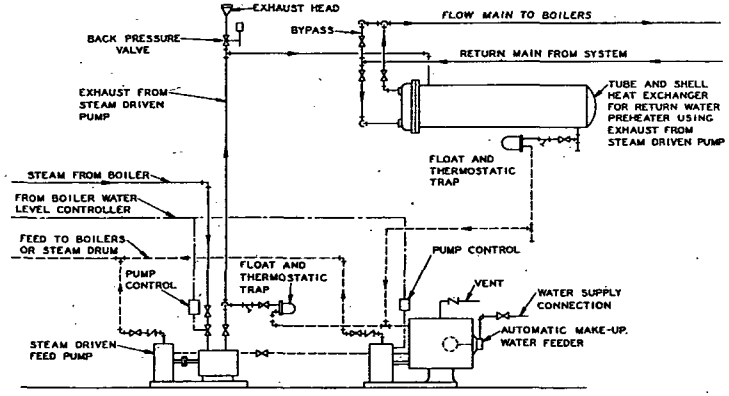


Fig. 15.... Arrangement of Boiler Feed Where Exhaust Steam Is Available

because then the control of water flow through the coil is not as critical.

In some cases face and bypass damper control is used in conjunction with valve control. The coil valve, operating in sequence with the damper, must be closed when the face damper is closed. The coil arrangement should be such as to minimize air temperature pick-up when the bypass damper is at or near the open position.

WATER TREATMENT

Water treatment is seldom required where there is continual re-use of the water. Where none of the water is lost, except such minor amounts as may accompany venting or be due to occasional leakage at stuffing boxes, it is usually sufficient to make the water slightly alkaline when the system is first filled. Where the make-up water is appreciable, treatment appropriate to the mineral content and quantity is indicated. See Closed Recirculation Systems in Chapter 55.

It is desirable that systems with steam driven auxiliaries be arranged to conserve the heat in the exhaust. Steam driven auxiliaries can rarely be justified unless the exhaust steam can be used. Usually this is accomplished with a minor addition in capital expenditure. Fig. 15 shows an arrangement for boiler feed where exhaust steam is available.

STORAGE

High-temperature water systems may be operated to even out the peak loads and low loads within 24-hr cycles by storage. Storage is usually accomplished by the bypassing of water from the flow to the return mains and thereby storing heat in the return main for future use, or storing heat in accumulators. Systems which experience normal peaks may obtain as much as 15 percent added capacity through such heat storage.

REFERENCES

[1] J. G. Frost: High temperature hot water boiler plant at Dartmouth (*The Engineering Journal of Canada*, June 1953).

[2] O. S. Lieberg: High temperature water (*Air Conditioning, Heating and Ventilating*, September 1955, p. 83; November 1955, p. 85; January 1956, p. 69). Also published as a separate bulletin.

[3] E. A. Pierce: High pressure hot water systems (*The Heating and Ventilating Engineer & Journal of Air Conditioning*, London, 1942).

[4] J. R. Kell: High pressure hot water (*Heating, Piping and Air Conditioning*, April 1948, p. 93; June 1948, p. 97; August 1948, p. 91; October 1948, p. 85).

[5] George Applegate, Jr.: British and European design and construction methods (ASHAE Journal Section, *Heating, Piping and Air Conditioning*, March 1958, p. 169).

[6] C. A. Carter and B. L. Sturtevant: Design of high temperature water systems for military installations (ASHAE Journal Section, *Heating, Piping and Air Conditioning*, February 1958, p. 109).

[7] E. G. Hansen and William Liddy: A flexible high pressure hot water and steam boiler plant (*Power*, May 1958, p. 109).

[8] Centrifugal Pump Section, *Standards of the Hydraulic Institute*, p. B(Vi)-9 (Hydraulic Institute, New York, 1955).

[9] Oscar Faber and J. R. Kell: *Heating and Air Conditioning of Buildings* (Architectural Press, London).

BIBLIOGRAPHY

J. H. Keenan and F. G. Keyes: *Thermodynamic Properties of Steam* (John Wiley & Sons, Inc., New York, 1936, 1st ed.).

S. R. Lewis: Hot-water system of advantage in meeting plant heating requirements (*Heating, Piping and Air Conditioning*, May 1938, p. 319).

K. Aschof: Die Calliqua-Heisswasserheizung (*Die Wärme*: Zeitschrift für Dampfkessel & Machinenbetrieb, October 1931).

F. R. L. White: Some notes on high-pressure hot-water heating (*Journal of the Institution of Heating and Ventilating Engineers*, 1945).

P. L. Geiringer: High pressure hot water (*Heating, Piping and Air Conditioning*, May 1948, p. 103).

T. W. Reynolds: The high temperature water system (*Heating and Ventilating*, September 1951).

High Temperature High Pressure Hot Water Heating Manual No. 2685 (Dunham-Bush, Inc., New York, 1955).

Charles Broder: Heating and air-conditioning a civilian airport (ASHAE Journal Section, *Heating, Piping and Air Conditioning*, March 1958, p. 147).

S. W. Brown: Marine applications of high-temperature water (ASHAE Journal Section, *Heating, Piping and Air Conditioning*, March 1958, p. 161).

E. M. Thompson: Economic evaluation of high-temperature water (ASHAE Journal Section, *Heating, Piping and Air Conditioning*, April 1958, p. 140).

CHAPTER 30

PANEL HEATING

Application Methods: Embedded Piping for Ceilings, Walls, or Floors; Warm Air and Electrically Heated Ceilings, Walls, or Floors; Output from Panel Surfaces: Radiation, Convection and Combined Heat Transfer, Panel Heat Losses; Design of Panel Heating Systems: Warm Water Panels for Plaster, Metal, and Concrete Ceilings; Wall Panels and Concrete Floor Panels, Installation Details and Accessories, Controls; Warm Air and Electric Panels

IN this chapter the term, Panel Heating, is used to describe a method of space heating in which heat is supplied by large heated areas of interior room surfaces operating at relatively low surface temperatures (80 to 125 F). The heating elements usually consist of warm water piping, warm air ducts, or low temperature electrical resistance elements embedded in, or located behind, ceiling, wall, or floor surfaces.

Panel heating may be considered as another method of convenient and effective space heating. The heat loss requirements may be calculated in the conventional manner except that the heat loss through the area occupied by the heated panel need not be included. An assumed or computed reverse heat loss from the panel, however, should be included in determining heating main size and the boiler load. The heat release from the panel is expressed in terms of hourly heat output per square foot of surface. The room air temperatures to be maintained are approximately the same as those maintained by heating systems employing cast-iron radiators, convectors, or warm air ducts.

This chapter does not include a separate discussion of such topics as the influence of radiation on human comfort, the mechanisms by which human beings release heat, and other similar topics that apply to all methods of heating interior spaces for human comfort. The reader is referred to Chapter 6 for a detailed discussion of these subjects.

APPLICATION METHODS

The great majority of panel installations of the past 50 years (which is the period of the modern utilization of this method of heating) have used warm water as the heating medium which is circulated in embedded piping. More recently, the use of warm air ducts, and embedded electrical heating elements, has come into favor, especially where specific local factors have influenced such use. Steam has been used only occasionally because of the problems which result from its higher temperature.

When the heating medium is warm water, both ferrous (steel or wrought iron) or non-ferrous (generally copper or aluminum) pipe or tube are used widely in ceiling, wall, or floor panel construction. Tube sizes used are ⅜, ½, ⅝, and ¾ in. OD, while piping is generally ½, ¾, or 1 in. IPS. Where coils are embedded in concrete or plaster, no threaded joints should be used for either pipe coils or mains. The construction should be of all-welded type. Changes in direction should be made by bending the pipe itself, rather than by use of fittings. Solder-joint fittings are used for non-ferrous heating coils and piping. It is recommended that a medium temperature solder of 95 percent tin—5 percent antimony, or capillary brazing alloys, be used. All piping should be subjected to a hydrostatic test of at least three times the working pressure, but not less than 150 psig.

The most common forms of panels applied in panel heating are: (1) embedded piping for ceilings; (2) embedded piping for walls; (3) embedded piping for floors; (4) air heated ceilings, walls, or floors; (5) electrically heated ceilings, walls, or floors.

Embedded Piping for Ceilings

When piping is embedded in ceilings, the construction used is generally one of the following:

a. Pipe or tube is embedded in the lower portion of a concrete slab, generally very close to its lower surface. If plaster is to be applied to the concrete, the piping may be placed directly on the wood forms. If the slab is to be used without plaster finish, then the piping should be installed not less than ¾ in. above the undersurface of the slab. Fig. 1 shows this method of construction. The minimum coverage must be in compliance with the local building code requirements.

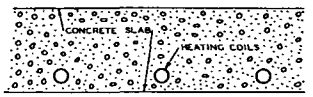


Fig. 1 . . . . Coils in Structural Concrete Slab

b. Pipe or tube is embedded in a metal lath and plaster ceiling. If the lath is suspended to form a hung ceiling, both the lath and the heating coils are securely wired to the supporting members in such a way that the lath is below, but in good contact with the coils, as shown in Fig. 2. Plaster is then applied to the metal lath, care being taken to embed the coil, as shown in Fig. 2.

c. Copper tube of the smaller diameters is attached to the underside of wire lath or gypsum lath. Plaster is then applied to the lath to embed the tube, as shown in Fig. 3.

d. Other forms of ceiling construction are prefabricated panels of metal, composition board, wood paneling, etc., having warm water piping, tube, or channels built into the panel sections.

Coils are usually of the sinuous type, although some header or grid type coils have been used in ceilings. Coils may be of either ferrous or non-ferrous pipe or tube, with coil pipes spaced from 4½ to 9 in. on centers, depending on the required output, pipe or tube size, and other factors.

Where plastering is applied to pipe coils, a standard three-coat gypsum plastering specification[1] is followed, with a minimum of ⅜ in. of cover below the tubes when the tubes

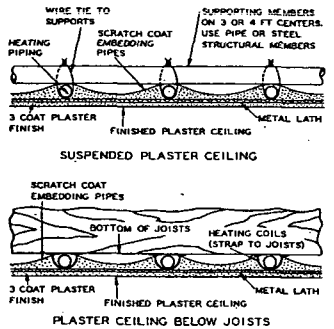


Fig. 2 .... Coils in Plaster Above Lath

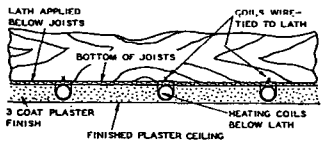


Fig. 3 .... Coils in Plaster Below Lath

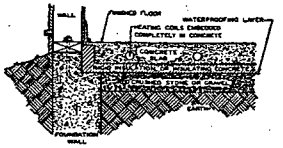


Fig. 4 .... Coils in Floor Slab on Grade

are installed below the lath. Generally, the surface temperature of plaster panels should not exceed 120 F, and this is usually met by limiting the water temperature in the pipes or tubes in contact with the plaster to a maximum temperature of 140 F. Insulation should be placed above the coils to reduce the *reverse loss* which is the difference between the heat supplied to the coil and the net useful output to the heated room.

In order to protect the plaster installation and to assure proper air drying of the plaster, it is recommended that no heat be applied to the panels for two weeks after all plastering work has been completed. When the system is started for the first time, the water supplied to the panels should be at a temperature not more than 20 deg above the prevailing room temperature at that time (but not in excess of 90 F). Water should be circulated at this temperature for about two days. Then the water temperature should be increased at a rate of approximately 5 deg increase per day to 140 F.

During the air-drying and preliminary warming-up periods, adequate ventilation should be provided to carry off moisture from the panels. No paint or paper should be applied to the panels until these periods have been completed. No paint or paper should be applied while the panels are being operated. After paint and paper have been applied, a further shorter warm-up period, similar to that for the first time starting, is also recommended.

Embedded Piping for Walls

Although not so universally used as ceiling panels, wall panels may be constructed by any of the methods outlined for ceilings.

Embedded Piping for Floors

The construction for piping embedded in floors will depend upon whether (*a*) the floor is laid on grade, or (*b*) the floor is above grade.

a. Both ferrous and non-ferrous pipe and tube are used in floor slabs which rest on grade. The coils are constructed as either sinuous-continuous pipe coils, or arranged as header coils with the pipes spaced from 6 to 18 in. on centers. The coils are generally installed with 1½ to 4 in. of cover above the coils. It is recommended that insulation be used to reduce the perimeter and reverse losses. Fig. 4 shows the application of pipe coils in slabs resting on grade. Coils should be embedded completely and should not rest on an interface. Any supports used for positioning the heating coils should be non-absorbent and inorganic. It is suggested that reinforcing steel, angle iron, pieces of pipe, or stone concrete mounds be used. No wood, brick, concrete block, or similar materials should be used for support of coils. Generally, a waterproofing layer is desirable to protect insulation and piping.

b. Where the coils are embedded in structural load supporting slabs above grade, construction codes may affect their position. Otherwise, the coil piping is installed in the same manner as described for slabs resting on grade.

c. A warm-up and start-up period for concrete panels should be similar to that outlined for plaster panels.

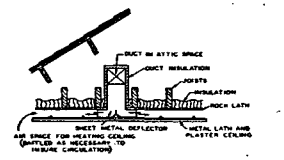


Fig. 5 .... Warm Air Plaster Ceiling Construction

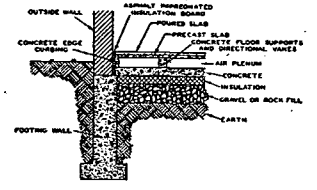


Fig. 6 .... Warm Air Floor Panel Construction

Air-Heated Ceilings, Walls, and Floors

Several methods have been devised to warm the interior room surfaces by circulating heated air through passages behind these surfaces. In some cases, the heated air is recirculated in a closed system. In others, all or a part of the air is passed through the room on its way back to the furnace to provide supplementary heating and ventilation.[2] Figs. 5 and 6 indicate two common types of construction. Care must be exercised to assure compliance with any building codes that might apply. (See also section on Warm Air Ceiling Panel Systems in Chapter 18.)

Electrically Heated Ceilings, Walls, or Floors

Several different forms of electric resistance units are available for heating the interior room surfaces. These include: (1) resistance cables that may be embedded in a manner similar to hot water piping in concrete or plaster; (2) prefabricated electric heating panels to be attached to room surfaces; and (3) electrically heated fabrics or other materials for application to, or incorporation into, finished room surfaces. Figs. 7 and 8 indicate two methods of installation. The constructions of electric panels for ceilings, walls, and floors are described in greater detail in Chapter 17, Electric Heating.

HEAT OUTPUT FROM PANEL SURFACES

A heated panel transfers heat to a room by convection and radiation. In the following paragraphs, the two transfer mechanisms are first considered separately and then combined to facilitate design calculations.

Radiation Transfer

The radiation transfer can be evaluated by means of the Stefan and Boltzmann equation:

$$q_r = 0.173 F_e F_r \left[\left(\frac{T_p}{100} \right)^4 - \left(\frac{T_r}{100} \right)^4 \right] \qquad (1)$$

where

q_r = heat transfer by radiation, Btu per (hour) (square foot).
T_p = temperature of panel heated surface, Fahrenheit, absolute.

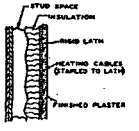


Fig. 7 .... Electric Heating Cables in Plaster

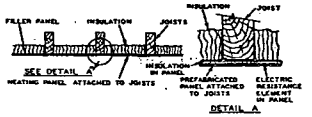


Fig. 8 .... Prefabricated Electric Panel

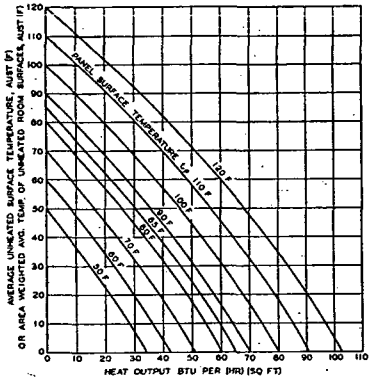


Fig. 9 .... Heat Output by Radiation

T_r = mean radiant temperature of all unheated surfaces, Fahrenheit, absolute.
F_e = the configuration factor (dimensionless).
F_r = the emissivity factor (dimensionless).

A combined configuration and emissivity factor for a simple, box-like room in which there is a *uniformly* heated ceiling, floor, or wall, all other surfaces are at another temperature, and all surfaces are perfectly diffusing is given by Hottel[3] as:

$$F_e = F_e F_r = \frac{1}{\frac{1}{F_{1-2}} + \left(\frac{1}{e_1} - 1 \right) + \frac{A_1}{A_2} \left(\frac{1}{e_2} - 1 \right)} \qquad (2)$$

where

F_e = combined configuration and emissivity factor.
F_{1-2} = view factor = 1.0.
e_1 and e_2 = emissivities of the surfaces.
A_1 and A_2 = areas of the surfaces.

In practice the emissivity of conventional, non-metallic, non-reflecting surfaces will be found to be about 0.9. When this emissivity is used in Equation 2, the combined factor will be found to be about 0.87 for most rooms. Substituting this value in Equation 1, the constant becomes about 0.15 and the equation can be rewritten:

$$q_r = 0.15 \left[\left(\frac{t_p + 460}{100} \right)^4 - \left(\frac{\text{AUST} + 460}{100} \right)^4 \right] \qquad (3)$$

where

t_p = temperature of panel surface, Fahrenheit.
AUST = area-weighted average temperature of unheated surfaces in room, Fahrenheit.

The actual radiation transfer in a room may be somewhat different from that given by Equation 3 because of non-uniform temperatures, irregular room surfaces, variations in

emissivity of materials, etc. It is generally agreed, however, that the equation is accurate to within 10 percent when used in conventional heating and cooling calculations. Tests[4] conducted in the ASHAE Environment Laboratory[5] showed that the value of the constant of Equation 3 was 0.152 in the test room. The design information in this chapter is based on that value of the constant.

Radiation exchange calculated from Equation 3 is given in Fig. 9. The values apply to either ceiling, floor, or wall panel radiation output.

Convection Transfer

Convection values of heat transfer are not easily established. Convection in a panel heated space is usually considered to be of the *natural* type, that is, air motion is generated by the warming (by conduction) of the boundary layer of air which starts moving as soon as its temperature exceeds that of the surrounding air. In practice, however, there are many factors which interfere with or affect *natural* convection. The configuration of the room and the space connected to it and the variation in temperature throughout the two spaces determine the *natural* convection. Infiltration, ventilation, and the movement of persons may serve to introduce some *forced* convection which can disturb the natural process.

The laboratory approach has been to measure the *natural* convection in a bare, box-like room which is sealed against infiltration and in which all surfaces, except the heated panel, are at one uniform temperature. Convection was also measured with varying amounts of infiltration and a correction determined for use in practical problems.[6, 7]

Tests[4] conducted in the ASHAE Environment Laboratory indicated that the natural convection from floors and ceilings can be calculated from Equations 4 to 9 which follow.

Natural convection from heated ceiling

$$q_c = 0.041 \frac{(t_p - t_a)^{1.25}}{D_e^{0.25}} \qquad (4)$$

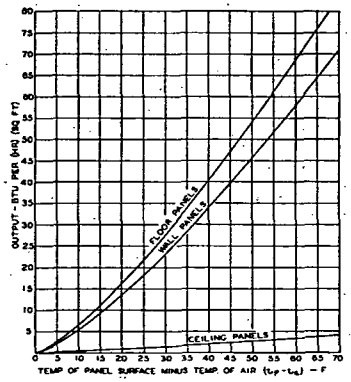
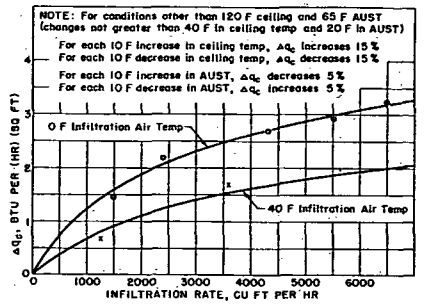


Fig. 10 . . . . Heat Output by Convection from Floor and Ceiling Panels



Ceiling at 120 F, all other surfaces at 65 F (24½ x 12 x 8 ft high room)

Fig. 11 . . . . Additional Heat Flow from Ceiling due to Infiltration Air[6]

Natural convection from heated floor

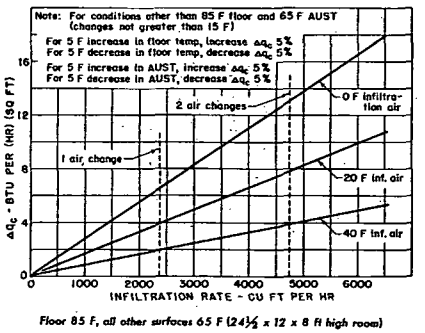
$$q_c = 0.39 \frac{(t_p - t_a)^{1.25}}{D_e^{0.50}} \qquad (5)$$

Natural convection from heated wall

$$q_c = 0.29 \frac{(t_p - t_a)^{1.25}}{H^{0.05}} \qquad (6)$$

where

q_c = heat transfer by convection, Btu per (hour) (square foot).
t_p = temperature of panel surface, Fahrenheit.
t_a = temperature of the air, Fahrenheit.
D_e = equivalent diameter of panel (area × 4 ÷ perimeter), feet.
H = height of wall panel, feet.



Floor 85 F, all other surfaces 65 F (24½ x 12 x 8 ft high room)

Fig. 12 . . . . Additional Heat Output from Floor due to Infiltration Air

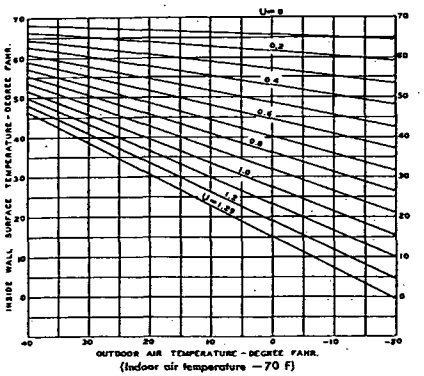


Fig. 13 . . . . Relation of Inside Surface Temperature to Overall Coefficient of Heat Transfer

Measurements of panel performance in furnished test rooms which did not have uniform temperature surfaces showed variations that are not large enough to be significant in heating practice.[6] Other tests[8] established that the effect of room size was also usually insignificant. The convection equations can therefore be simplified to:

Natural convection from heated ceiling

$$q_c = 0.021 (t_p - t_a)^{1.25} \qquad (7)$$

Natural convection from heated floor

$$q_c = 0.32 (t_p - t_a)^{1.25} \qquad (8)$$

Natural convection from heated wall

$$q_c = 0.26 (t_p - t_a)^{1.25} \qquad (9)$$

Fig. 10 shows panel convection outputs calculated from these equations.

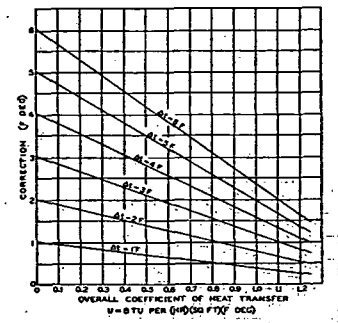
Infiltration and ventilation increase the convection from panels. Figs. 11 and 12 show the amount of increase Δq_c observed in the ASHAE Environment Laboratory.[6, 7] Equations 7 and 8 and Figs. 11 and 12 were used in the development of the design data for this chapter.

Combined Heat Transfer

The heat transfer from a panel to a room can be determined by adding the radiant heat transfer from Fig. 9 to the convective heat transfer from Fig. 10.

Use of Fig. 9 requires the calculation of the AUST, the area-weighted average surface temperature of the unheated surfaces in the room. In calculating AUST, the surface temperature of *inside* walls is assumed to be the same as the room air temperature. The surface temperatures of *outside* walls and exposed floors or ceilings can be obtained from Fig. 13 for a 70 F room air temperature. Corrections for other temperatures may be obtained from Fig. 14.

The combined heat transfer for ceiling panels and floor panels in rooms in which the air temperature is 70 to 76 F can be read directly from Figs. 15 and 16, respectively. These two diagrams apply to rooms in which the AUST



$\Delta t = t_a - 70$
If Δt is positive: $t_{ai}' = t_{ai} +$ correction
If Δt is negative: $t_{ai}' = t_{ai} -$ correction
t_a = indoor air temperature
t_{ai} = inside wall surface temperature based on t_a = 70 F
t_{ai}' = actual inside wall surface temperature

Fig. 14 . . . . Inside Wall Surface Temperature Correction for Air Temperatures Other Than 70 F

does not differ greatly from room air temperatures. Tests[6, 7] showed that the two temperatures are normally nearly equal.

Effect of Floor Coverings

Floor coverings may have a pronounced effect upon the performance of a floor panel system. The added thermal resistance of the floor covering causes a reduction in upward heat flow, and increases the heat flow to the underside of the

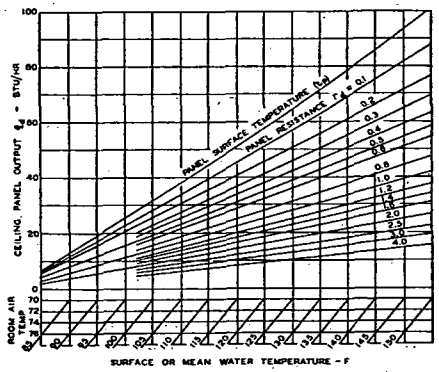


Fig. 15 . . . . Ceiling Panel Design Graph Showing Panel Surface Temperature and Mean Water Temperature vs Output Downward

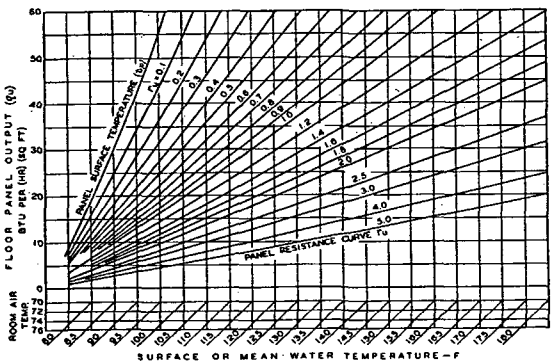


Fig. 16 . . . . Floor Panel Design Graph Showing Panel Surface Temperature and Mean Water Temperature vs Output Upward

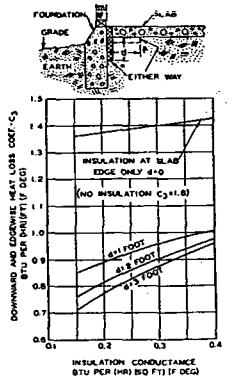


Fig. 17 . . . . Downward and Edgewise Heat Loss Coefficient for Concrete Floor Slabs on Grade

slab. To maintain a given upward heat flow after a floor covering has been added, the temperature of the heating medium must be increased. Data on the thermal resistance of common floor coverings are given in Table 4, and the application of the data is illustrated in *Example 2*.

Where covered and bare floor panels exist on the same system, it is necessary to maintain the water temperature high enough to satisfy the covered panels, and balance the system by throttling the flow to the bare slabs. When possible, it is desirable to divide the covered and bare panels into *separate zones*.

Panel Heat Losses

Heat transferred from the upper surface of ceiling panels, the back surface of wall panels, the underside of floor panels, or the edges of any panel is considered a panel heat loss. Panel heat losses are part of the building heat loss if the heat is transferred outside of the building. If the heat is transferred to another heated space, the panel loss is a source of heat for the space and is not a part of the building heat loss. In either case, the magnitude of the panel loss should be determined.

Panel heat loss to the space outside the room should be kept to a reasonable amount by insulation. Panel heat loss to heated spaces may require reduction by insulation if the amount of heat transferred is excessive or if objectionable temperatures will be developed. For example, a floor panel may overheat the basement below and a ceiling panel may cause the temperature of a floor surface above it to be too high for comfort.

The heat loss from most panels can be calculated by using the coefficients given in Chapter 9, Heat Transmission Coefficients of Building Materials. These coefficients should not be used to determine the downward heat loss from panels built on grade because the heat flow from them is not uniform.[10, 11, 12] The heat loss from panels built on grade can be estimated from Fig. 17.

DESIGN OF PANEL HEATING SYSTEMS

Design Steps

Panel design requires specification of the following: panel area, size and location of the heating elements in the panel, insulation on the reverse side and edge of the panel, required input to panel, and temperature of the heating elements. The procedure is summarized as follows:

1. Calculate the hourly rate of heat loss for each room.
2. Determine the available area for panels in each room.
3. Calculate the required unit panel output.
4. Determine the required panel surface temperature.
5. Select the means of heating the panel and the size and location of the heating elements.
6. Select the insulation for the reverse side and edge of the panel.
7. Determine the panel heat loss and the required input to the panel.
8. Determine the other temperatures which are required or developed.
9. Design the system for heating the panels in accordance with conventional practice.

In the steps outlined for design, the effect of each assumption or choice on comfort should be considered carefully. Tests are now being made at the ASHAE Research Laboratory to identify and measure the factors which contribute to comfort and to establish comfort design principles. Until the results are available the following general rules may be followed:

1. Place panels near the cold areas where the heat losses occur.
2. Do not use high temperature ceiling panels in very low ceilings.
3. Keep floor temperatures at or below recommended limits.

Warm Water Panels

This section contains a simplified procedure for the thermal design of water-heated panels for use in residences and commercial buildings. The procedures are based primarily on the experimental data obtained at the ASHAE Research Laboratory. This work has been reported in a series of research papers which are listed in the references and bibliography at the end of this chapter.

A panel designed by these procedures will maintain the desired room air temperature for the selected outdoor conditions. Room air temperature is the selected criterion of comfort. The design procedure is restricted to situations in which the area-weighted average temperature of unheated surfaces of walls, glass, and floor or ceiling does not differ greatly from room air temperature. Room-scale tests, which simulated various conditions of construction and outdoor temperature, have shown that this near-equality of the two temperatures normally prevails.

The procedures are applicable within the following range:

Outdoor design conditions: Temperatures as low as −30 F.

Room air temperature: 70 to 76 F.

Air changes: No more than two air changes per hour.

Room dimensions: Rooms having normal proportions; ceiling height between 7 and 12 ft.

Room construction: Any type of wall construction and any amount of glass area. (Both, however, have an effect upon comfort.) Conventional interior finishes and furnishings.

Design Examples

In the design examples which follow use is made of the letter symbols shown in the table Letter Symbols for Examples of Design Methods.

Letter Symbols for Examples of Design Methods

A_p = panel area, square feet.

C_1 = coefficient of heat transfer from the *upper* surface of the concrete slab which forms the ceiling panel to air above panel at point t_b, Btu per (hour) (square foot) (Fahrenheit degree temperature difference between panel surface and air).

C_2 = coefficient of heat transfer from *lower* surface of concrete slab to air below the panel at point t_b, Btu per (hour) (square foot) (Fahrenheit degree temperature difference between panel surface and air).

C_3 = coefficient of downward and edgewise heat loss of exposed slab, Btu per (hour) (linear foot of exposed slab perimeter) (Fahrenheit degree difference between concrete surface and outdoor air).

P = length of exposed edge of slab, feet.

q_d = downward heat flow from panel, Btu per (hour) (square foot).

q_{de} = apportioned downward and edgewise heat flow from panel, Btu per (hour) (square foot).

q_u = upward heat flow from panel, Btu per (hour) (square foot).

r_d = total thermal resistance of panel to downward heat flow, (Fahrenheit degree) (hour) (square foot) per Btu.

r_{dc} = thermal resistance of material between the underside of the concrete slab and the ceiling surface below, (Fahrenheit degree) (hour) (square foot) per Btu.

r_{dc} = thermal resistance of bare concrete panel to downward heat flow, (Fahrenheit) (hour) (square foot) per Btu.

r_u = total thermal resistance of panel to upward heat flow (Fahrenheit degree) (hour) (square foot) per Btu.

r_{uc} = thermal resistance of floor covering, (Fahrenheit degree) (hour) (square foot) per Btu.

r_{uc} = thermal resistance of bare concrete slab to upward heat flow, (Fahrenheit degree) (hour) (square foot) per Btu.

t_a = design room air temperature, Fahrenheit.

t_{ao} = outdoor design air temperature, Fahrenheit.

t_b = air temperature above or below panel at point to which U, C_1, or C_2 is taken, Fahrenheit.

t_i = inlet water temperature, Fahrenheit.

t_o = outlet water temperature, Fahrenheit.

t_{mw} = mean water temperature, Fahrenheit.

t_{max} = maximum water temperature permissible for a given construction, Fahrenheit.

t_{mwo} = design mean water temperature (selected for each zone), Fahrenheit.

t_p = panel surface temperature, (exposed surface) Fahrenheit.

t_s = surface temperature of top of concrete slab, Fahrenheit.

U = overall coefficient of heat transfer for the given construction between room air and the point t_b, Btu per (hour) (square foot) (Fahrenheit degree temperature difference).

Procedure for Plaster Ceiling Panels[13]

The procedure for designing a plaster ceiling panel will be illustrated by *Example 1*.

Example 1: Three rooms, A, B, and C, are to have a common water supply temperature; that is, they represent a single zone. They are to be maintained at 72 F air temperature when the outdoor air temperature is zero F. The ceilings of rooms A and B have floors above them with the space heated to 72 F and an air-to-air U value of 0.25 Btu per (hr) (sq ft) (F deg). The ceiling of room C has insulation in the joist spaces and an uninsulated attic space with a combined U value of 0.05 from room C to outdoor air.

Step 1. Heat Loss

Calculate the heat loss of each room by methods outlined in *Chapter 12, Heating Load*, but *do not include* any heat loss through the area covered by the panel.

Room dimensions and calculated heat losses are as follows:

Room	Dimensions Feet	Heat Loss Btu per hour
Room A	11 x 12 x 8	6300
Room B	11 x 12 x 8	2500
Room C	15 x 21 x 8	8000

Step 2. Required Panel Output

Divide the heat loss of each room by the maximum ceiling area in the room which can be used as a heating panel. The result is the minimum heat output per square foot of panel that will satisfy the requirements of the room. The panel that requires the highest output per square foot will generally control the design, because the temperature of the fluid in the system must be high enough to produce the required out-

put from that panel. The required panel output is given in the following table.

Available Panel Area Sq Ft	Required Panel Output, q_d Btu/(hr) (sq ft)
Room A 132 sq ft	$\frac{6300}{132} = 47.7$
Room B 127 sq ft	$\frac{2500}{127} = 19.7$
Room C 306 sq ft	$\frac{8000}{306} = 26.1$

Step 3. Panel Surface Temperature

From Fig. 15 find the panel surface temperature needed to yield the required heat output to each room, using the output determined in Step 2 and the design room air temperature. This use of Fig. 15 for Room A is illustrated in Fig. 15A.

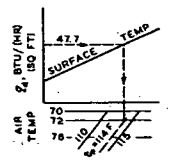


Fig. 15A . . . . Determination of Panel Surface Temperature for Room A from Fig. 15

The values of t_p determined for Rooms A, B, and C are as follows:

Room A t_p = 114 F
Room B t_p = 92 F
Room C t_p = 97 F

Step 4. Upward Heat Flow

Determine the upward heat flow from each panel using Fig. 18. The upward heat flow must be determined in order to obtain the downward panel resistance and to select the proper size of boiler. The use of Fig. 18 is illustrated for Room A in Fig. 18A. Note that the U values shown on Fig. 18 are those

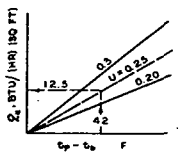


Fig. 18A . . . . Determination of Upward Heat Flow for Room A from Fig. 18

for the entire structure of which the panel is a part and are taken from room air (temperature, t_a) to air above the panel (temperature, t_b) and for that reason $q_u \neq U(t_p - t_a)$. U values may be obtained from Chapter 9, Heat Transmission Coefficients of Building Materials.

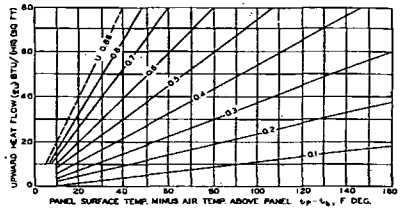


Fig. 18 . . . . Upward Heat Flow from Plaster Ceiling Panel

The values of q_u for Rooms A, B, and C as determined from Fig. 18 are as follows:

Room	U	$t_p - t_a$ F Deg	q_u Btu/(hr)(sq ft)
A	0.25	42	12.5
B	0.25	20	6.0
C	0.05	97	5.0

Step 5. Downward Panel Resistance

Assume tentative pipe or tube spacings for the panel construction to be used from those listed in Table 1, choosing the closer spacings when higher outputs are required. Also in Table 1 find the resistance of each panel (r_d), using heat flow ratios calculated from the flow rates determined in Steps 2 and 4, interpolating as required.

Room	Spacing (Inches)	Heat Flow Ratio Resistance (r_d)	
A	4½	$\frac{12.5}{47.7} = 0.26$	0.35
B	9	$\frac{6}{19.7} = 0.30$	0.90
C	9	$\frac{5}{26.1} = 0.19$	0.85

Step 6. Mean Water Temperature

For the required panel output (q_d) found in Step 2, the panel resistance (r_d) found in Step 5, and the room air temperature (t_a), find the mean water temperature (t_{mw}) from Fig. 15. This use of Fig. 15 is illustrated in Fig. 15B.

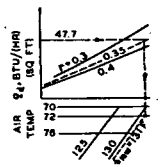
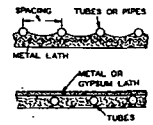
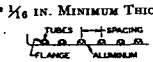


Fig. 15B . . . . Determination of Mean Water Temperature for Room A from Fig. 15

Table 1 . . . . Thermal Resistance of Plaster and Metal Ceiling Panels

	Spacing in.	Thermal Resistance to Downward Heat Flow (r_d) (F deg) (hr) (sq ft) per Btu					
		Heat Flow Ratio, q_u/q_d					
		0	0.2	0.4	0.6	0.8	1.0[a]
Plaster Panels[b] Standard Gypsum Plaster, Three Coats	4½	0.30	0.34	0.38	0.42	0.46	0.50
							
⅜ in. (nom.) non-ferrous tube or ½ in. (nom.) ferrous pipe *above* metal lath tied at 8-inch intervals with good tube embedment, or ⅜ in. (nom.) nonferrous tube *below* metal or gypsum lath.	6	0.45	0.51	0.57	0.63	0.69	0.75
	9	0.75	0.85	0.95	1.05	1.15	1.25
	12	1.15	1.29	1.43	1.57	1.71	1.85
Aluminum Panel[c] 1/16 in. Minimum Thickness 							
pipe or tube installed so that excellent heat conduction to ceiling surface is obtained.	3	0.07	0.07	0.07	0.07	0.07	0.07

[a] Any ceiling panel also acts as a floor panel to the extent of its upward heat flow. If the upward heat flow is high and the space above is occupied, check floor surface temperature for possible foot discomfort (see Reference 7). Also check effect on heating requirements of the space above. It is not good practice to have the major portion of the upper room's heating requirements supplied by the upward heat flow of a ceiling panel below.

[b] Recommended maximum inlet water temperature (t_{mwa}) 140 F.

[c] Recommended maximum inlet water temperature (t_{mwa}) 220 F.

Values of t_{mw} for Rooms A, B, and C as determined from Fig. 15 are as follows.

Room A t_{mw} = 131 F, Room B t_{mw} = 110.5 F, Room C t_{mw} = 119 F

Step 7. Design Mean Water Temperature

Select a single design mean water temperature (t_{mwa}) for each group of rooms which is to comprise a zone, choosing the highest mean water temperature (t_{mw}) of the group subject to the following:

a. If $t_{mw} + \frac{(t_i - t_a)}{2}$ is equal to or less than t_{max} (see Nomenclature and Table 1, this mean water temperature (t_{mw}) is an acceptable design mean water temperature (t_{mwa}). Proceed to Step 8.

b. If $t_{mw} + \frac{(t_i - t_a)}{2}$ is greater than t_{max}, go back to Step 5 and select a panel construction which has a lower panel resistance (r_d). This can be accomplished by either or both of the following:

1. Reducing the tube spacing,

2. Decreasing the upward heat flow (q_u) by providing additional insulation above the panel.

If the required mean water temperature is still too high apply either or both of the following:

1. Reduce the heat loss of the room,

2. Provide supplementary heating.

In Example 1, assume $t_i - t_a$ = 15 F deg and t_{max} = 140 F. Then since t_{mw} = 131 F, $t_{mw} + \frac{1}{2}(t_i - t_a)$ = 138.5 F and is less than t_{max}. Thus 131 F can be used as the design mean water temperature t_{mwa}.

Step 8. Design Panel Output

From Fig. 15, find the panel output (q_d) for design mean water temperature (t_{mwa}), room air temperature (t_a), and panel resistance (r_d). Fig. 15C shows how Fig. 15 is used to find q_d for Room B in this step.

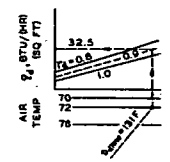


Fig. 15C . . . . Determination of Design Panel Output for Room B from Fig. 15

The mean water temperature (t_{mwa}) found for Room A in Step 6 was used as the design mean water temperature (t_{mwa}). The panel output for Room A which was determined in Step 2 is therefore the design panel output for this room. Design panel outputs for Rooms B and C can be found from Fig. 15 using the design mean water temperature (t_{mwa}).

Room	Design Panel Output (q_d) Btu per (hr) (sq ft)
A	47.7
B	32.5
C	33.0

Step 9. Design Panel Area

Divide the room heat loss found in Step 1 by the design panel output found in Step 8.

Room	Design Panel Area (A_p) Sq ft
A	132 (Unchanged from Step 2)
B	77
C	242

Step 10. Total Panel Output

Add the heat flow upward (q_u) to the design panel output (q_d) and multiply by the design panel area to obtain the total panel output. If the design panel output is different from the panel output (q_d) used in Steps 3 and 4, the heat flow upward (q_u) should be redetermined.

Room	q_u	q_d	$q_u + q_d$	A_p	Total Panel Output $A_p(q_u + q_d)$ Btu/hr
A	12.5	47.7	60.2	132	7946
B	9.5†	32.5	42.0	77	3234
C	5.5†	33.0	38.5	242	9317

† redetermined

Step 11. Fluid Circuit

Design the fluid circuit (panel piping and mains) for a temperature drop of 10 to 20 F deg between the water inlet and outlet of the panel (see Chapter 28, Hot Water Heating Systems).

Step 12. Boiler Size

Size the boiler according to method contained in Chapter 35, Heating Boilers, Furnaces, and Space Heaters. The net Btu rating of the boiler should equal or exceed the total output of all panels plus any other loads on the boiler.

Procedure for Metal Ceiling Panels[12]

Follow the procedure for Hot Water Plaster Ceiling Panels.

Procedure for Concrete Ceiling Panels[13]

Concrete ceiling panels are distinguished from concrete floor panels in intermediate floors by the position of the tubes in the concrete slabs (see Tables 2 and 3). Both types of panels have heat outputs in both directions in amounts determined by the thermal resistances and the temperature differences in the two directions. The effect of both outputs on heating requirements of the spaces and the comfort of the occupants should always be considered. (See section on Procedure for Concrete Floor Panels—Intermediate Slab.)

The procedure for hot water plaster ceiling panels cannot be applied in ite entirety to concrete ceiling panels because some of the simplifying assumptions regarding the upward heat flow from plaster panels are not valid for concrete panels. The necessary modification of the plaster ceiling panel procedure is as follows:

Step 1. Heat Loss

Follow the procedure for Plaster Ceiling Panels, Step 1.

Step 2. Required Panel Output

Follow the procedure for Plaster Ceiling Panels, Step 2.

Table 2 .... Thermal Resistance of Concrete Ceiling Panels

Panel Construction	Spacing in.	Thermal Resistance (F deg)(sq ft)(hr) per Btu					
		Heat Flow Ratio q_u/q_d					
		0		0.5[a]		1.0[a]	
		up	down	up	down	up	down
		r_{ua}	r_d	r_u	r_d	r_u	r_d
6-in. Concrete Slab—1-in. cover							
½-in. (nom.) nonferrous tube	9	3.6	0.30	0.9	0.35	0.7	0.45
	12	5.1	0.35	1.1	0.45	0.9	0.55
½-in. (nom.) ferrous pipe or ¾-in. (nom.) nonferrous tube	9	2.6	0.25	0.7	0.30	0.6	0.35
	12	4.0	0.30	0.9	0.40	0.8	0.50
¾-in. (nom.) ferrous pipe or 1-in. (nom.) nonferrous tube	9	2.1	0.20	0.6	0.25	0.6	0.30
	12	3.3	0.30	0.8	0.35	0.7	0.40
	15	4.5	0.35	1.0	0.45	0.8	0.55
1-in. (nom.) ferrous pipe	9	1.6	0.20	0.5	0.25	0.5	0.25
	12	2.6	0.25	0.7	0.30	0.6	0.35
	15	3.6	0.30	0.9	0.40	0.7	0.45
8-in. Concrete Slab—1-in. Cover							
½-in. (nom.) nonferrous tube	9	3.6	0.30	1.0	0.35	0.8	0.40
	12	5.2	0.35	1.2	0.45	1.0	0.55
½-in. (nom.) ferrous pipe or ¾-in. (nom.) nonferrous tube	9	2.9	0.25	0.9	0.30	0.8	0.35
	12	4.0	0.30	1.1	0.40	0.9	0.45
¾-in. (nom.) ferrous pipe or 1-in. (nom.) nonferrous tube	9	2.2	0.20	0.8	0.30	0.7	0.30
	12	3.3	0.30	1.0	0.35	0.8	0.40
	15	4.3	0.35	1.1	0.40	0.9	0.50
1-in. (nom.) ferrous pipe	9	1.7	0.20	0.7	0.25	0.7	0.25
	12	2.7	0.25	0.9	0.30	0.8	0.35
	15	3.7	0.30	1.0	0.40	0.9	0.45

[a] Any ceiling panel also acts as a floor panel to the extent of its upward heat flow. If the upward heat flow is high and the space above is occupied, check floor surface temperature for possible foot discomfort (see Reference 7). Also check effect on heating requirements of the space above. It is not good practice to have the major portion of the upper room's heating requirements supplied by the upward heat flow of a ceiling panel below.

Step 3. Panel Surface Temperature

Follow the procedure for Plaster Ceiling Panels, Step 3.

Step 4. Upward Heat Flow Estimate

a. If the upper surface of the slab is exposed to form a floor, find the heat flow upward from Fig. 16, using the panel surface temperature (t_p) found in Step 3, and the air temperature of the space above.

b. If the upper surface of the slab is not exposed, use the equation:

$$q_u = C_1(t_b - t_p) \qquad (10)$$

Table 3 .... Thermal Resistance of Bare Concrete Floor Panels

Panel Construction	Spacing in.	Thermal Resistance (F deg)(sq ft)(hr)/Btu							
		Heat Flow Ratio q_u/q_d or q_w/q_d							
		1		3		5		10	
		up	down	up	down	up	down	up	down
4-in. Concrete Slab—2-in. Cover		r_{ua}	r_{da}	r_{ua}	r_{da}	r_{ua}	r_{da}	r_{ua}	r_{da}
½-in. (nom.) nonferrous tube	9	0.57	0.52	0.46	0.84	0.43	1.17	0.42	1.97
	12	0.73	0.68	0.58	1.16	0.54	1.65	0.51	2.86
½-in. (nom.) ferrous pipe or ¾-in. (nom.) nonferrous tube	9	0.49	0.42	0.41	0.66	0.39	0.90	0.38	1.80
	12	0.63	0.55	0.50	0.93	0.48	1.30	0.46	2.35
6-in. Concrete Slab—2-in. Cover									
½-in. (nom.) nonferrous tube	9	0.59	0.70	0.47	1.05	0.45	1.39	0.43	2.25
	12	0.78	0.90	0.60	1.40	0.56	1.97	0.54	3.21
¾-in. (nom.) nonferrous tube	9	0.51	0.61	0.43	0.87	0.41	1.13	0.40	1.78
	12	0.68	0.78	0.54	1.23	0.51	1.63	0.49	2.61
¾-in. (nom.) ferrous pipe	9	0.47	0.55	0.40	0.77	0.39	0.98	0.38	1.50
	12	0.63	0.71	0.50	1.07	0.48	1.44	0.46	2.36
1-in. (nom.) nonferrous tube or 1-in. (nom.) ferrous pipe	12	0.59	0.66	0.48	0.98	0.46	1.30	0.45	2.11
	15	0.73	0.83	0.57	1.21	0.54	1.73	0.51	2.74

Step 5. Upward and Downward Panel Resistance

Follow the procedure for Plaster Panels, Step 5, using Table 2 to find both resistances (r_{ua} and r_d).

Step 6. Mean Water Temperature and Upward Heat Flow

a. Follow the procedure for Plaster Panels, Step 6.

b. Find the heat flow upward from the panel (q_u) from Fig. 16. Add to the thermal resistance of the slab to upward heat flow (r_{ua}), the resistance to heat flow (r_{ua}) of any material between the upper surface of the slab and the space above to obtain the resistance (r_u) to be used in Fig. 16. The mean water temperature (t_{mw}) found and the air temperature of the space above the panel are the other two factors to be used.

Step 7. Design Mean Water Temperature

Select the highest mean water temperature (t_{mw}) as the design mean water temperature (t_{dmw}).

Step 8. Design Panel Output

Follow the procedure for Plaster Panels, Step 8.

Step 9. Design Panel Area

Follow the procedure for Plaster Panels, Step 9.

Step 10. Total Panel Output

Follow the procedure for Plaster Panels, Step 10.

Step 11. Fluid Circuit

Follow the procedure for Plaster Panels, Step 11.

Step 12. Boiler Size

Follow the procedure for Plaster Panels, Step 12.

Procedure for Plaster Wall Panels

A design graph has not been prepared for wall panels but a design can be approximated using the equations of heat transfer from walls together with the thermal resistance properties of plaster ceilings from Table 2. The procedure for Plaster Ceiling Panels is used as a guide.

Step 1. Heat Loss

Follow the procedure for Plaster Ceiling Panels, Step 1.

Step 2. Required Panel Output

Follow the procedure for Plaster Ceiling Panels, Step 2.

Step 3. Panel Surface Temperature

Assume a trial panel surface temperature and determine the resulting heat output from Figs. 9 and 10 as explained in the accompanying section of the test. (Additional heat flow due to the infiltration can be estimated from Fig. 12, if desired.) Assume successive trial panel surface temperatures until the temperature is found at which the combined heat transfer from the panel equals the output determined in Step 2.

Step 4. Reverse-Side Heat Flow

Follow the procedure for Plaster Ceiling Panels, Step 4.

Step 5. Panel Resistance

Follow the procedure for Plaster Ceiling Panels, Step 5.

Step 6. Mean Water Temperature

For the required panel output found in Step 2, the panel resistance found in Step 5, and the room air temperature, calculate the required mean water temperature as follows:

$$t_{mw} = t_p + r_d(q_d) \qquad (11)$$

Step 7. Design Mean Water Temperature

Follow the procedure for Plaster Ceiling Panels, Step 7.

Step 8. Design Panel Output

From equations, Step 6, and Figs. 9 and 10 find the panel output for design mean water temperature t_{dmw} by successive trials.

Step 9. Design Panel Area

Follow the procedure for Plaster Ceiling Panels, Step 9.

Step 10. Total Panel Output

Follow the procedure for Plaster Ceiling Panels, Step 10.

Step 11. Fluid Circuit

Follow the procedure for Plaster Ceiling Panels, Step 11.

Step 12. Boiler Size

Follow the procedure for Plaster Ceiling Panels, Step 12.

Procedure for Hot Water Concrete Floor Panels (Slab-On-Grade)[12]

The procedure for designing a concrete floor panel (slab-on-grade) will be illustrated by *Example 2*.

Example 2: Three rooms, A, B, and C, are to have a common water supply temperature; that is, they represent a single zone. They are to be maintained at 72 F air temperature

when the outdoor air temperature is zero F. The floors in the three rooms are covered with heavy wall-to-wall carpeting, without a pad. Insulation which has a conductance of 0.4 Btu per (hr) (sq ft) (F deg) is placed along the edge of the slab and extends downward 2 ft below it.

Step 1. Heat Loss

Calculate the heat loss of each room, following the recommendations contained in Chapter 12, Heating Load, but do not include any heat loss through the area covered by the panel.

If very large rooms are involved, the rooms should be subdivided into areas having somewhat similar heat requirements, i.e., separate the interior areas requiring little or no heat input, from the exterior areas directly influenced by outdoor weather conditions. Each such area should then be treated as a separate room for the purpose of design.

Room dimensions, exposure, and calculated heat losses are as follows:

Room	Dimensions Feet	Exposed Wall Feet	Heat Loss Btu per hr
A*	11 x 12 x 8	23	3300
B	11 x 12 x 8	12	2500
C	15 x 21 x 8	21	8000

* Compare with Room A of *Example 1*. The rooms are the same except that in this example a lower heat loss is assumed to permit the room to be heated by a floor panel alone without developing an inadmissibly high panel surface temperature (Step 2). The lower heat loss is presumably the result of adding insulation to Room A.

Step 2. Required Panel Output

Divide the heat loss of each room by the maximum floor area in the room which can be used as a heating panel. The result is the minimum heat output per square foot of panel that will satisfy the requirements of the room. The panel that requires the highest output per square foot will generally control the design, because the temperature of the water in the system must be high enough to produce the required output from that panel.

Room	Available Panel Area Sq ft	Required Panel Output (q_u) Btu/(hr) (sq ft)
A	132	25.0
B	127	19.7
C	306	26.1

Step 3. Panel Surface Temperature

From Fig. 16 find the panel surface temperature needed to yield the required heat output to each room, using the output determined in Step 2 and the design room air temperature.

Floor panel surface temperatures in excess of about 85 F are not recommended because of the probability of discomfort to the feet. If the required heat output cannot be obtained from an 85 F floor panel, heat losses should be reduced or supplementary heating should be provided.

Room	Panel Surface Temperature (t_p)
A	85.0 F
B	82.5 F
C	85.3 F

Table 4 . . . . Thermal Resistance of Floor Coverings

Description	Resistance, r_{ss} (F deg)(hr)/(sq ft)/Btu
Bare concrete, no covering	0.00
Asphalt tile	0.05
Rubber tile	0.05
Light carpet	0.6
Light carpet with rubber pad	1.0
Light carpet with light pad	1.4
Light carpet with heavy pad	1.7
Heavy carpet	0.8
Heavy carpet with rubber pad	1.2
Heavy carpet with light pad	1.6
Heavy carpet with heavy pad	1.9

Step 4. Downward and Edgewise Heat Flow

a. Determine the temperature of the surface of the concrete slab (t_s) by adding to the panel surface temperature, the temperature difference caused by the thermal resistance of the floor covering, if any:

$$t_s = t_p + (q_u \times r_{ss}) \qquad (12)$$

The value of (r_{ss}) for various floor coverings is given in Table 4.

Room	t_p	q_u	r_{ss}	t_s
A	85.0 F	25.0	0.8	105.0 F
B	82.5 F	19.7	0.8	98.3 F
C	85.3 F	26.1	0.8	106.2 F

b. Determine the downward and edgewise heat loss coefficient C_s from Fig. 17 for the insulation to be used. Insulation with a conductance of 0.4 Btu per (hr) (sq ft) (F deg) extending two feet below the slab results in a slab downward and edgewise heat loss coefficient of 0.97 Btu per (hr)(ft) (F deg)

c. Apportion the downward and edgewise heat loss uniformly across the panel as follows:

$$q_{de} = \frac{P \times C_s(t_s - t_{ma})}{A_p} \qquad (13)$$

Room	P	$t_s - t_{ma}$	A_p	q_{de}
A	23	105.0 F	132	17.7
B	12	98.3 F	127	9.0
C	21	106.2 F	306	7.1

Step 5. Upward Panel Resistance

Assume a tentative pipe or tube size and a spacing for each panel, choosing closer spacings and larger pipe or tube when higher heat outputs are required. From Table 3, find the thermal resistance (r_{ss}) of the slab of each panel, using heat flow ratios (q_u/q_{de}) calculated from the flow rates determined in Steps 2 and 4, interpolating as required. To the slab resistance (r_{ss}), add the resistance of the floor covering (r_{sc}, Step 4) to obtain the panel resistance to upward heat flow (r_u).

Room	Pipe or Tube Spacing	q_u/q_{de}	r_{ss}	r_{sc}	r_u
A	9 in.	$\frac{25}{17.7} = 1.41$	0.47	0.8	1.27
B	12 in.	$\frac{19.7}{9.0} = 2.19$	0.55	0.8	1.35
C	9 in.	$\frac{26.1}{7.1} = 3.68$	0.40	0.8	1.20

All rooms: ½-in. pipe or ⅝-in. tube; 4-in. concrete slab.

Step 6. Mean Water Temperature

For the required panel output (q_u) found in Step 2, the panel resistance (r_u) found in Step 5, and the room air temperature (t_a), find the mean water temperature (t_{mw}) from Fig. 16.

Room	t_{mw}
A	116.5
B	109.0
C	116.5

Step 7. Design Mean Water Temperature

Select a single design mean water temperature (t_{dmw}) for each group of rooms which is to comprise a zone, choosing the highest mean water temperature (t_{mw}) of the group. In the example, 116.5 F is chosen.

Step 8. Design Panel Output

From Fig. 16 find the panel output (q_u) for design mean water temperature (t_{dmw}), room air temperature (t_a), and panel resistance (r_u). Also from Fig. 16 find the design panel surface temperature. If it exceeds 85 F (see Step 3) in a room, go back to Step 5 and choose a wider spacing or smaller pipe or tube for that room.

Room	(q_u)	t_p
A	25.0	85.0 (unchanged)
B	24.0	84.5
C	26.1	85.3 (unchanged)

Step 9. Design Panel Area

Divide the room heat loss found in Step 1 by the design panel output found in Step 8.

Room	A_p
A	132 (unchanged)
B	104
C	306 (unchanged)

Step 10. Total Panel Output

Add the apportioned downward and edgewise heat flow (q_{de}) to the design panel output (q_u) and multiply by the design panel area to obtain the total panel output. If the design panel output is appreciably different from the panel output (q_u) used in Steps 3 and 4, the apportioned downward and edgewise heat flow (q_{de}) should be redetermined.

Room	q_u	q_{de}	$q_u + q_{de}$	A_p	$A_p(q_u + q_{de})$
A	25.0*	17.7*	42.7	132	5636
B	24.0	11.6**	35.6	104	3702
C	26.1*	7.1*	33.2	306	10159

* Unchanged. ** Redetermined.

Step 11. Fluid Circuit

Design the fluid circuit (panel piping and mains) for a temperature drop of 10 to 20 F deg between the water inlet (t_i) and outlet (t_o) of the panel (see Chapter 23, Hot Water Heating Systems).

Step 12. Boiler Size

Select the size of the boiler according to method contained in Chapter 35, Heating Boilers, Furnaces, and Space Heaters. The net Btu rating of the boiler should equal or exceed the total output of all panels plus any other loads on the boiler.

Procedure for Hot Water Concrete Floor Panels (Intermediate Slab)[12]

The procedure for designing a hot water concrete floor panel of intermediate slab type is given in the following 12 steps.

Step 1. Heat Loss

Follow the procedure for slab-on-grade construction, Step 1.

Step 2. Required Panel Output

Follow the procedure for slab-on-grade construction, Step 2.

Step 3. Panel Surface Temperature

Follow the procedure for slab-on-grade construction, Step 3.

Step 4. Downward Heat Flow Estimate

A. Follow the procedure for slab-on-grade construction, Step 4, part 1.

B. Estimate the heat flow downward as follows:

1. If the underside of the concrete slab is exposed to form a ceiling, find the heat output downward (q_d) from Fig. 15, using the slab surface temperature (t_s) found in Step 3 as the panel surface temperature (t_p), and the air temperature of the space below.

2. If the underside of the concrete slab is not exposed, use the equation:

$$q_d = C_s(t_s - t_b) \qquad (14)$$

Step 5. Upward and Downward Panel Resistance

Follow the procedure for slab-on-grade construction, Step 5, using Table 3 to find both resistances (r_{ss} and r_{sc}).

Step 6. Mean Water Temperature and Downward Heat Flow

a. Follow the procedure for slab-on-grade construction, Step 6.

b. Use Fig. 15 to find the downward heat flow (q_d). Add to the slab resistance to downward heat flow, (r_{ss}), the resistance to heat flow, (r_{sc}), of any material between the underside of the slab and the ceiling surface, to find the total resistance to downward heat flow (r_d) to be used in Fig. 15. The water temperature (t_{mw}) found above and the air temperature (t_b) of the space below the ceiling are the other two factors to be used.

If the heat flow downward (q_d) differs appreciably from the estimate made in Step 4, repeat Steps 5 and 6 using the calculated value.

Step 7. Design Mean Water Temperature

Follow the procedure for slab-on-grade construction, Step 7.

Step 8. Design Panel Output

Follow the procedure for slab-on-grade construction, Step 8.

Step 9. Design Panel Area

Follow the procedure for slab-on-grade construction, Step 9.

Step 10. Total Panel Output

Follow the procedure for slab-on-grade construction, Step 10, substituting heat flow downward (q_d) for the apportioned downward and edgewise heat flow (q_{de}).

Step 11. Fluid Circuit

Follow the procedure for slab-on-grade construction, Step 11.

Step 12. Boiler Size

Follow the procedure for slab-on-grade construction, Step 12.

Installation Details and Accessories

The design of the two-pipe, direct return forced circulation system which is used in a panel heating system is described generally in Chapter 28, Hot Water Heating Systems. In addition to the usual valves, a balancing valve should be installed in the supply or return connection of each panel in order to permit the heat output of the panel to be balanced with other panels in the same zone.

While all piping should be designed to minimize trapping of air or water, few panel heating systems can be installed to be completely free of air traps. For that reason, panel heating systems cannot usually be filled by flooding nor drained by gravity alone. The system can be filled and the air removed if the system is flushed with a sufficient flow of water. To empty the system a flow of compressed gas is used to assist gravity drainage of the water. Provisions for both flushing and blowing should be included in the piping. Devices for the collection and venting of the gases that will appear during operation should also be provided.

Controls

Automatic controls for panel heating differ somewhat from those described for convective heating because of the thermal inertia characteristics of the panel heating surface and the increase in the mean radiant temperature within the space under increasing loads for panel heating. However, many of the control principles for hot water heating systems described in Chapter 28 will also apply to panel heating. (See also Chapter 43.)

Panels such as concrete slabs have large heat storage capacity and continue to emit heat long after the room thermostat has shut off the supply of heating medium. In addition, there is a considerable time lag between thermostat demand and heat delivery to the space due to the large part of the heat which must first be stored in the thermally heavy radiant surface. This inertia will cause uncomfortable variations in space conditions unless controls for detecting load changes as early as possible are provided.

In general, the temperature of the heating medium supplied to the panel surface should be varied in accordance with outdoor temperature but precautions must be taken to prevent the introduction of excessively hot water which might damage the panels in the event of control failure. A manual boiler bypass or other means of reducing the water temperature may be necessary to prevent too rapid drying out of new panels (see section, Embedded Piping for Ceiling Panels, in this chapter).

Due to the increase in MRT (mean radiant temperature) within a panel-heated space which necessarily takes place as the heating load increases, the air temperature under such conditions, theoretically, should be lowered in the order of 1 or 2 degrees to maintain comfort. In ordinary structures with normal infiltration loads the required reduction in air temperature is small and, consequently, a conventional room thermostat may be used.

In panel heating systems, lowered night temperatures will produce unsatisfactory results with heavy panels such as concrete floors. These panels cannot respond to either quick increase or decrease in heating demand within the relatively short time required, with the result that there will be a very slow reduction of space temperature at night and a correspondingly slow pickup in the morning. Panels of light weight construction, such as plaster or metal ceilings and walls, may respond to changes in demand with sufficient rapidity to obtain moderately satisfactory results from lowered night temperatures. However, very little fuel saving can be expected even with the light panels unless the lowered temperature is maintained for long periods. If reduced nonoccupancy temperatures are employed, some means of providing a higher-than-normal rate of heat input for rapid warm-up is necessary, or a long warmup period should be provided, as explained in Chapter 28.

Warm Air Panels

The first three steps in the design of warm air panels are the same as those outlined for warm water panels and the same performance curves can be used. The balance of the design can be determined from the data in the Chapter on Forced Warm Air Systems and Manual 7-A of the National Warm Air Heating and Air Conditioning Association.

Electric Panels

Electric panel heating systems can be designed by using part of the procedure for warm water panels as a guide. See also Chapter 17, Electric Heating.

Step 1. Heat Loss

Follow the procedure for warm water panels, Step 1.

Step 2. Required Panel Output

Follow the procedure for warm water panels, Step 2.

Step 3. Panel Surface Temperature

Follow the procedure for warm water panels, Step 3.

Step 4. Panel Heat Loss

Determine the heat loss from the panel, using the panel surface temperature found in Step 3 and the factors in Chapter 9, Heat Transmission Coefficients of Building Materials.

REFERENCES

[1] *Standard Specifications for Gypsum Plastering, Including Requirements for Lathing and Plastering* (American Standards Association, A 42.1, 1946).

[2] *Manual 7A Design and Installation of Warm Air Ceiling Panel Systems* (National Warm Air Heating and Air Conditioning Association).

[3] W. H. McAdams: Chapters 4 and 7 (*Heat Transmission*, McGraw-Hill Book Co., New York, 1954, 3rd ed.).

[4] T. C. Min, L. F. Schutrum, G. V. Parmelee, and J. D. Vouris: ASHAE Research Report No. 1576—Natural convection and radiation in a panel heated room (ASHAE Transactions, Vol. 62, 1956, p. 337).

[5] Cyril Tasker, C. M. Humphreys, G. V. Parmelee, and L. F. Schutrum: ASHVE Research Report No. 1444—The ASHVE Environment Laboratory (ASHVE Transactions, Vol. 58, 1952, p. 139).

[6] L. F. Schutrum, G. V. Parmelee, and C. M. Humphreys: ASHVE Research Report No. 1473—Heat exchanges in a ceiling panel heated room (ASHVE Transactions, Vol. 59, 1953, p. 197).

[7] L. F. Schutrum, G. V. Parmelee, and C. M. Humphreys: ASHVE Research Report No. 1490—Heat exchanges in a floor panel heated room (ASHVE Transactions, Vol. 59, 1953, p. 495).

[8] L. F. Schutrum and C. M. Humphreys: ASHVE Research Report No. 1499—Effects of non-uniformity and furnishings on panel heating performance (ASHVE Transactions, Vol. 60, 1954, p. 121).

[9] L. F. Schutrum and J. D. Vouris: ASHVE Research Report No. 1516—Effects of room size and non-uniformity of panel temperature on panel performance (ASHVE Transactions, Vol. 60, 1954, p. 455).

[10] H. H. Macey: Heat loss through a solid floor (*Institute of Fuel Journal*, 22-128, p. 369).

[11] E. L. Sartain and W. S. Harris: Performance of covered hot water floor panels, Part I—Thermal characteristics (ASHAE Transactions, Vol. 62, 1956, p. 55).

[12] A Subcommittee of the TAC on Panel Heating and Cooling, R. L. Maher, Chairman; W. P. Chapman; H. T. Gilkey; P. B. Gordon; E. F. Snyder; and J. M. Van Nieukerken; and by ASHAE Laboratory Staff Members, L. F. Schutrum and C. M. Humphreys: ASHAE Research Report No. 1600—Thermal design of warm water concrete floor panels (ASHAE Transactions, Vol. 63, 1957, p. 239).

[13] A Subcommittee of the TAC on Panel Heating and Cooling, R. L. Maher, Chairman; W. P. Chapman; H. T. Gilkey; P. B. Gordon; E. F. Snyder; and J. M. Van Nieukerken; and by ASHAE Laboratory Staff Members, L. F. Schutrum, G. V. Parmelee, and C. M. Humphreys: ASHAE Research Report No. 1559—Thermal design of warm water ceiling panels (ASHAE Transactions, Vol. 62, 1956, p. 71).

BIBLIOGRAPHY

B. F. Raber and F. W. Hutchinson: Trend curves for estimating performance of panel heating systems (ASHVE Transactions, Vol. 48, 1942, p. 425).

B. F. Raber and F. W. Hutchinson: ASHVE Research Report No. 1192—Panel heating and cooling performance studies (ASHVE Transactions, Vol. 48, 1942, p. 35).

F. C. Houghten, Carl Gutberlet, and E. C. Hach: ASHVE Research Report No. 1193—Radiation as a factor in the feeling of warmth in convection radiator and panel heated rooms (ASHVE Transactions, Vol. 48, 1942, p. 55).

B. F. Raber and F. W. Hutchinson: Panel heating and cooling, analysis (ASHVE Transactions, Vol. 47, 1941, p. 285).

E. J. Rodee: Operating results of a residence radiant wall heating system (ASHVE Transactions, Vol. 47, 1941, p. 123).

H. F. Randolph and J. B. Wallace: Performance of a residential panel heating system (ASHVE Transactions, Vol. 49, 1943, p. 235).

Radiant heating (*Heating and Ventilating*, March 1941, p. 35).

F. E. Giesecke: Radiant heating and cooling (*Heating, Piping and Air Conditioning*, June, July, August, September and October 1940).

T. N. Adlam: Calculations for radiant heating (*Heating and Ventilating*, October 1931).

C. O. Mackay, L. T. Wright, Jr., R. E. Clark, and N. R. Gay: *Radiant Heating and Cooling* (Cornell University, *Engineering Experiment Station Bulletin* No. 32, 1943).

R. G. Vanderweil: Design method for panel heating systems using copper tubing (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, November 1947, p. 123).

J. M. Ayres and B. W. Levy: Air temperature gradients in a panel heated room (ASHVE Transactions, Vol. 54, 1948, p. 131).

W. P. Chapman and R. E. Fischer: Graphical solution of radiant panel areas (*Heating and Ventilating*, January 1948, p. 88).

D. L. Mills and L. J. LaTart: Embedding coils in radiant heating panels (*Heating and Ventilating*, December 1947, p. 75).

D. L. Mills and L. J. LaTart: Radiant heat with copper tubing (*Heating and Ventilating*, November 1947, p. 95).

D. L. Mills and L. J. LaTart: Panel heat with copper tubing—experiment in practice (*Heating and Ventilating*, October 1947, p. 65).

B. F. Raber and F. W. Hutchinson: Experimental studies on panel heating tube spacing (ASHVE Transactions, Vol. 53, 1947, p. 369).

S. Konzo: Panel heating—a basic discussion (*American Artisan*, October 1946, p. 68).

John E. Peterson: Solar house heated by a warm air floor panel (*American Artisan*, December 1946, p. 83).

A. B. Algren: Design data for a warm air floor panel (*American Artisan*, January 1947, p. 141).

C. S. Leopold: The mechanism of heat transfer, panel cooling, heat storage (*Refrigerating Engineering*, July 1947, p. 33).

Radiant Heating—Simplified Design and Installation (Copper and Brass Research Association, 1949).

C. M. Humphreys, H. B. Nottage, C. V. Franks, R. G. Huebscher, L. F. Schutrum, and D. W. Locklin: ASHVE Research Report No. 1387—Laboratory studies on heat flow within a concrete panel (ASHVE Transactions, Vol. 56, p. 175).

L. E. Hulbert, H. B. Nottage, and C. V. Franks: ASHVE Research Report No. 1388—Heat flow analysis in panel heating or cooling sections (ASHVE Transactions, Vol. 56, 1950, p. 189).

C. F. Kayan: Electric analogger studies on panels with imbedded tubes (ASHVE Transactions, Vol. 56, 1950, p. 205).

A. B. Algren: ASHVE Research Report No. 1345—Ground temperature distribution with a floor panel heating system (ASHVE Transactions, Vol. 54, 1948, p. 321).

F. W. Hutchinson, D. L. Mills, and L. J. LaTart: Losses from a floor-type panel heating system (ASHVE Transactions, Vol. 57, 1951, p. 37).

C. M. Humphreys, C. V. Franks, and L. F. Schutrum: ASHVE Research Report No. 1418—Field studies of heat losses from concrete floor panels (ASHVE Transactions, Vol. 57, 1951, p. 221).

N. S. Billington: Losses from heated floors (*Journal of the Institution of Heating and Ventilating Engineers*, June 1953).

E. L. Sartain and W. S. Harris: Heat flow characteristics of hot water floor panels (ASHAE Transactions, Vol. 60, 1954, p. 103).

F. C. Houghten, S. I. Taimuty, Carl Gutberlet, and C. J. Brown: ASHVE Research Report No. 1213—Heat loss through basement walls and floors (ASHVE Transactions, Vol. 48, 1942, p. 369).

R. S. Dill, W. C. Robinson, and H. E. Robinson: *Measurements of Heat Losses from Slab Floor* (National Bureau of Standards, *Building Materials and Structures Report* BMS 103).

H. A. Bareither, A. N. Fleming, and B. E. Alberty: Temperature and heat loss characteristics of concrete floors laid on ground (*American Artisan*, March 1950).

N. S. Billington: Heat loss through solid ground floors (*Journal of the Institution of Heating and Ventilating Engineers*, November 1951).

J. R. Jamieson, R. W. Roose, and S. Konzo: Warm-air perimeter heating: Part III—Heat losses from floor slab (ASHVE Transactions, Vol. 58, 1952, p. 217).

M. Baker, J. M. O'Byrne, and A. M. Levy: Estimating the heat loss from slab floors and basements (*Heating, Piping and Air Conditioning*, November 1952).

N. S. Billington and E. W. Shaw: Experiments with intermittently operated floor panels (*Journal of the Institution of Heating and Ventilating Engineers*, June 1952).

N. S. Billington: Heat loss through solid ground floors—II (*Journal of the Institution of Heating and Ventilating Engineers*, November 1952).

H. R. Martin, P. R. Achenbach, and R. S. Dill: *Effect of Edge Insulation Upon Temperature and Condensation on Concrete-Slab Floors* (National Bureau of Standards, *Building Materials and Structures Report* No. 138, October 1953).

Aydin Umur, G. V. Parmelee, and L. F. Schutrum: ASHAE Research Report No. 1528—Measurement of angular emissivity (ASHAE Transactions, Vol. 61, 1955, p. 111).

C. M. Humphreys, C. V. Franks, and L. F. Schutrum: ASHVE Research Report No. 1426—Laboratory studies of the thermal characteristics of plaster panels (ASHVE Transactions, Vol. 57, 1951, p. 363).

L. F. Schutrum and C. M. Humphreys: ASHVE Research

REPORT No. 1491—Further studies of the thermal characteristics of plaster panels (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 511).

H. B. Nottage, C. V. Franks, L. E. Hulbert, and L. F. Schutrum: ASHVE RESEARCH REPORT No. 1492—Heat flow analysis in panel heating or cooling sections, case II—Floor slab on earth with uniformly spaced pipes or tubes at the slab-earth interface (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 527).

C. M. Humphreys, H. B. Nottage, C. V. Franks, R. G. Huebscher, L. F. Schutrum, and D. W. Locklin: ASHVE RESEARCH REPORT No. 1387—Laboratory studies on heat flow within a concrete panel (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 175).

L. F. Schutrum and T. C. Min: ASHAE RESEARCH REPORT No. 1598—Cold wall effects in a ceiling-panel-heated room (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 187).

E. L. Sartain and W. S. Harris: Performance of covered hot water floor panels, Part II—Room conditions (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 209).

CHAPTER 31

PIPE, FITTINGS, WELDING

Pipe Materials, Types of Pipe, Commercial Pipe Dimensions, Threading Practice, Pipe Fittings, Flange Facings and Gaskets, Valves, Welding, Expansion and Flexibility, Hangers and Supports

IMPORTANT considerations in the selection and installation of pipe and fittings for heating, ventilating, and air conditioning are dealt with in this chapter.

PIPE MATERIALS

Use of corrosion-resistant materials for pipe, including special alloy steels, copper, and red brass, has increased considerably during the past few years. The following brief discussion indicates the variety of pipe materials and the types of pipe available.

Wrought-Steel Pipe. Because of its lower cost, the great bulk of wrought pipe used for heating and ventilating work at the present time is of wrought steel. The material used for steel pipe is a mild steel made by the acid-bessemer, the open-hearth, or the electric-furnace process. Ordinary wrought-steel pipe is made either by shaping sheets of metal into cylindrical form and welding the edges together, or by forming or drawing from a solid billet. The former is known as *welded pipe*, the latter as *seamless pipe*.

Many types of welded pipe are available, although the smaller sizes most frequently used in heating and ventilating work are made by the lap-weld, resistance-weld, or butt-weld process. While the lap-weld and resistance-weld processes produce a better weld than the butt type, lap-weld and resistance-weld pipe are seldom manufactured in nominal pipe sizes less than 2 in. Seamless pipe can be obtained in the small sizes at a somewhat higher cost, and is listed as pressure tubing below 2 in.

Seamless steel pipe is frequently used for high-pressure work or where pipe is desired for close coiling, cold bending, or other severe forming operations. Its advantages are its greater strength which permits use of a thinner wall and, in the small sizes, its freedom from the occasional tendency of welded pipe to split at the weld when bent.

Wrought-Iron Pipe. Wrought-iron pipe may be identified by the spiral line marked into each length, either knurled into the metal or painted on it in red or other bright color. Otherwise, there is little difference in the appearance of wrought-iron and steel pipe, although microscopic examination of polished and etched specimens will readily disclose different grain structures.

Cast-Ferrous Pipe. There are now available several types of cast-ferrous metal pipe made of a good grade of cast-iron with or without additions of nickel, chromium, or other alloy. This pipe is available in sizes from 1½ in. to 6 in., and in standard lengths of 5 or 6 ft, with external and internal diameters closely approximating those of extra strong wrought pipe. Cast-ferrous pipe may be obtained coupled, beveled for welding, of with ends plain or grooved for the several types of couplings. It is easily cut and threaded as well as welded. The fact that it is readily welded enables the manufacturers to supply the pipe in any lengths practicable for handling.

Alloy Metal Pipe. Both iron and steel pipe are available in the alloy class. In the case of iron pipe when copper and molybdenum are added, the material is known as alloy wrought-iron pipe.

By common custom steel is considered to be an alloy when the maximum range of alloying elements exceed certain limits such as 1.65 percent manganese, 0.60 silicon, or 0.60 copper. Also, a steel is an alloy if any of the following elements are specified in minimal quantities or within limits: Al, B, Cr (up to 3.99), Co, Cb, Mo, Ni, Ti, W, V, Zr, or any other element added to obtain a certain alloying effect. Small quantities of certain elements are present in alloy steels and are considered as incidental and may be present to the following maximum amounts: copper—0.35 percent, nickel—0.25 percent, chromium—0.20 percent, and molybdenum—0.06 percent.

Commercially, steel is considered stainless when chromium exceeds 3.99 percent regardless of the inclusion of other alloying elements. With *tubing*, however, an alloy becomes stainless when it contains a minimum of 10.5 percent chromium.

The alloy and stainless steels are used for high temperature piping and for highly corrosive fluids and gases.

Copper Pipe and Fittings. Owing to inherent resistance to corrosion, copper and red brass pipe have always been used in heating, ventilating, and water supply installations, but the cost with standard dimensions for threaded connections has been high. The introduction of fittings which permit erection by soldering or sweating, allows the use of tube with thinner walls than would be possible with threaded connections, thereby reducing the cost of installations.

COMMERCIAL PIPE DIMENSIONS

Commercial pipe dimensions are covered in the following specifications of the American Society for Testing Materials: (1) steel—ASTM A-120 and A-53, (2) wrought iron—ASTM A-72, and (3) copper tube—ASTM A-88.

Tables 1 and 2 are based on these specifications.

In addition to the specifications for these three materials, there are standards for copper or red brass IPS, clay or concrete, and cast-iron pipe. Copies of the following specifications may be obtained from the American Society for Testing Materials: (1) steel, wrought-iron, and cast-iron—Series A, (2) copper and red brass—Series B, and (3) clay and concrete—Series C.

Table 3 lists the sizes available in each of the specifications and classes of steel pipe. Note that butt-weld is not available above 4-in. size and that lap-weld, electric-resistance-weld, and seamless are not available below 2-in. size.

Table 1 . . . . Dimensions and Properties of Steel Pipe

Nominal[a] Size	ASTM[b] Schedule	Diameter OD in.	Diameter ID in.	Wall Thickness in.	Surface Area Sq Ft/Lin Ft OD	Surface Area Sq Ft/Lin Ft ID	Section Area Sq In. OD	Section Area Sq In. ID	Area of Metal Sq In.	Volume Gal/Lin Ft	Weight[c] (plain end) Lb/Lin Ft	Working Pressure[d] Psig
⅛	40 (s)	0.405	0.269	0.068	0.106	0.0704	0.129	0.0568	0.0720	0.00295	0.244	314 (a)
	80 (x)	0.405	0.215	0.095	0.106	0.0563	0.129	0.0363	0.0925	0.00189	0.314	1084 (a)
¼	40 (s)	0.540	0.364	0.088	0.141	0.0953	0.229	0.104	0.125	0.00541	0.424	649 (a)
	80 (x)	0.540	0.302	0.119	0.141	0.0791	0.229	0.0716	0.157	0.00372	0.535	1353 (a)
⅜	40 (s)	0.675	0.493	0.091	0.177	0.129	0.358	0.191	0.167	0.00992	0.567	574 (a)
	80 (x)	0.675	0.423	0.126	0.177	0.111	0.358	0.140	0.217	0.00730	0.738	1191 (a)
½	40 (s)	0.840	0.622	0.109	0.220	0.163	0.554	0.304	0.250	0.0158	0.850	697 (a)
	80 (x)	0.840	0.546	0.147	0.220	0.143	0.554	0.234	0.320	0.0122	1.09	1266 (a)
	XX	0.840	0.252	0.294	0.220	0.0660	0.554	0.0499	0.504	0.00259	1.71	3824 (a)
¾	40 (s)	1.050	0.824	0.113	0.275	0.216	0.886	0.533	0.333	0.0277	1.13	604 (a)
	80 (x)	1.050	0.742	0.154	0.275	0.194	0.866	0.432	0.434	0.0225	1.47	1078 (a)
	XX	1.050	0.434	0.308	0.275	0.114	0.866	0.148	0.718	0.00768	2.44	3134 (a)
1	40 (s)	1.315	1.049	0.133	0.344	0.275	1.36	0.864	0.494	0.0449	1.68	651 (a)
	80 (x)	1.315	0.957	0.179	0.344	0.251	1.36	0.719	0.639	0.0374	2.17	1083 (a)
	XX	1.315	0.599	0.358	0.344	0.157	1.36	0.282	1.08	0.0146	3.66	2963 (a)
1¼	40 (s)	1.660	1.380	0.140	0.435	0.361	2.16	1.50	0.669	0.0777	2.27	440 (a)
	80 (x)	1.660	1.278	0.191	0.435	0.335	2.16	1.28	0.881	0.0666	3.00	805 (a)
	XX	1.660	0.896	0.382	0.435	0.235	2.16	0.630	1.53	0.0328	5.21	2318 (a)
1½	40 (s)	1.900	1.610	0.145	0.497	0.421	2.84	2.04	0.800	0.1058	2.72	417 (a)
	80 (x)	1.900	1.500	0.200	0.497	0.393	2.84	1.77	1.07	0.0918	3.65	756 (a)
	XX	1.900	1.100	0.400	0.497	0.288	2.84	0.950	1.89	0.0494	6.41	2122 (a)
2	40 (s)	2.375	2.067	0.154	0.622	0.541	4.43	3.36	1.07	0.174	3.65	376 (a)
	80 (x)	2.375	1.939	0.218	0.622	0.508	4.43	2.95	1.48	0.153	5.02	690 (a)
	XX	2.375	1.503	0.436	0.622	0.393	4.43	1.77	2.66	0.0922	9.03	1861 (a)
2½	40 (s)	2.875	2.469	0.203	0.753	0.646	6.49	4.79	1.70	0.249	5.79	505 (a)
	80 (x)	2.875	2.323	0.276	0.753	0.608	6.49	4.24	2.25	0.220	7.66	806 (a)
	XX	2.875	1.771	0.552	0.753	0.364	6.49	2.46	4.03	0.128	13.7	2048 (a)
3	40 (s)	3.500	3.068	0.216	0.916	0.803	9.62	7.39	2.23	0.384	7.57	454 (a)
	80 (x)	3.500	2.900	0.300	0.916	0.759	9.62	6.61	3.02	0.343	10.3	734 (a)
	XX	3.500	2.300	0.600	0.916	0.602	9.62	4.15	5.47	0.216	18.5	1829 (a)
3½	40 (s)	4.000	3.548	0.226	1.05	0.929	12.6	9.89	2.68	0.514	9.11	425 (a)
	80 (x)	*4.000*	*3.364*	*0.318*	*1.05*	*0.881*	*12.6*	*8.89*	3.68	0.462	12.5	692 (a)
	XX	4.000	2.728	0.636	1.05	0.714	12.6	5.85	6.72	0.304	22.9	1699 (a)
4	40 (s)	4.500	4.026	0.237	1.18	1.05	15.9	12.7	3.17	0.661	10.8	403 (a)
	80 (x)	4.500	3.826	0.337	1.18	1.00	15.9	11.5	4.41	0.597	14.9	663 (a)
	XX	4.500	3.152	0.674	1.18	0.825	15.9	7.80	8.10	0.405	27.5	1602 (a)
5	40 (s)	5.563	5.047	0.258	1.46	1.32	24.3	20.0	4.30	1.04	14.6	498 (b)
	80 (x)	5.563	4.813	0.375	1.46	1.26	24.3	18.2	6.11	0.945	20.8	825 (b)
	XX	5.563	4.063	0.750	1.46	1.06	24.3	13.0	11.3	0.673	38.6	1951 (b)
6	40 (s)	6.625	6.065	0.280	1.73	1.59	34.5	28.9	5.58	1.50	18.0	467 (b)
	80 (x)	6.625	5.761	0.432	1.73	1.51	34.5	26.1	8.40	1.35	28.6	825 (b)
	XX	6.625	4.897	0.864	1.73	1.28	34.5	18.8	15.6	0.978	53.1	1912 (b)
8	30 (s)	8.625	8.071	0.277	2.26	2.11	58.4	51.2	7.26	2.66	24.7	351 (b)
	40 (s)	8.625	7.981	0.322	2.26	2.09	58.4	*50.0*	8.40	2.60	28.6	431 (b)
	80 (x)	8.625	7.625	0.500	2.26	2.00	58.4	45.7	12.8	2.37	43.4	753 (b)
	XX	8.625	6.875	0.875	2.26	1.80	58.4	37.1	21.3	1.93	72.4	1460 (b)
10	(s)	10.750	10.192	0.279	2.81	2.67	90.8	81.6	9.18	4.24	31.2	285 (b)
	30 (s)	10.750	10.136	0.307	2.81	2.65	90.8	80.7	10.1	4.19	34.2	324 (b)
	40 (s)	10.750	10.020	0.365	2.81	2.62	90.8	78.9	11.9	4.10	40.5	405 (b)
	60 (x)	10.750	9.750	0.500	2.81	2.55	90.8	74.7	16.1	3.88	54.7	600 (b)
12	30 (s)	12.750	12.090	0.330	3.34	3.17	128.	115.	12.9	5.96	43.8	299 (b)
	(s)	12.750	12.000	0.375	3.34	3.14	128.	113.	14.6	5.88	49.6	352 (b)
	(x)	12.750	11.750	0.500	3.34	3.08	128.	108.	19.2	5.63	65.4	503 (b)
14	30 (s)	14.000	13.250	0.375	3.67	3.46	154.	138.	16.0	7.17	54.6	458 (c)
	(x)	14.000	13.000	0.500	3.67	3.15	154.	133.	21.2	6.70	72.1	653 (c)

Table 1 . . . . Dimensions and Properties of Steel Pipe (Concluded)

Nominal[a] Size	ASTM[b] Schedule	Diameter OD in.	Diameter ID in.	Wall Thickness in.	Surface Area Sq Ft/Lin Ft OD	Surface Area Sq Ft/Lin Ft ID	Section Area Sq In. OD	Section Area Sq In. ID	Area of Metal Sq In.	Volume Gal/Lin Ft	Weight[c] (plain end) Lb/Lin Ft	Working Pressure[d] Psig
16	30 (s)	16.000	15.250	0.375	4.18	3.99	201.	183.	18.4	9.48	62.4	400 (c)
	40 (x)	16.000	15.000	0.500	4.18	3.93	201.	177.	24.3	9.18	82.8	570 (c)
18	(s)	18.000	17.250	0.375	4.71	4.52	254.	234.	20.7	12.1	70.6	355 (c)
	(x)	18.000	17.000	0.500	4.71	4.45	254.	227.	27.4	11.8	93.5	506 (c)
20	20 (s)	20.000	19.250	0.375	5.23	4.51	314.	291.	23.2	15.2	78.6	319 (c)
	30 (s)	20.000	19.000	0.500	5.23	4.97	314.	284.	30.6	14.7	104.2	454 (c)
24	20	24.000	23.250	0.375	6.29	6.08	452.	426.	26.8	22.1	94.6	265 (c)
	(s)	24.000	23.000	0.500	6.29	6.03	452.	415.	36.9	21.5	125.5	378 (c)

[a] 3½ double extra strong is no longer considered in *ASTM* specification but some pipe of this size is still manufactured.

[a] The sizes for wrought iron are approximately the same except wall thickness is slightly heavier. See *ASTM* A-72.

[b] *American Society for Testing Materials* Schedule. The numbers 30, 40, etc., refer to the *ASTM* Schedule; the letter (s) refers to the former designation *Standard Weight*; the letter (x) refers to the former designation *Extra Strong*; the letters XX refer to the former designation *Double Extra Strong*.

[c] Weight per foot is based on plain end pipe. Threaded and coupled (*T and C*) *pipe* is slightly heavier.

[d] Working pressure for welded joints—see formula in Table 3.

(1) Working pressure based on an allowable fiber stress of 6225 psi (for 250 F).

(2) Working pressure based on an allowable fiber stress of 8400 psi (for 250 F).

(3) Working pressure based on an allowable fiber stress of 12000 psi (for 250 F).

Note: Standard-weight pipe is generally furnished with threaded ends in random lengths of 16 to 22 ft., although when ordered with plain ends, 5 percent may be in lengths of 12 to 16 ft. Five percent of the total number of lengths ordered may be jointers which are two pieces coupled together. Extra-strong pipe is generally furnished with plain ends in random lengths of 12 to 22 ft., although 5 percent may be in lengths of 6 to 12 ft.

Listed in Table 3 is the formula for allowable working pressures as given in the *American Standard Code for Pressure Piping*.[1] In this formula is a factor *c*, which is intended as an allowance for corrosion, mechanical injury, manufacturing tolerances, etc. At times this factor seems too conservative; for example, with ⅛-in., Schedule 40, steel pipe, the allowable working pressure is 314 psi. Yet this pipe is tested at 700 psi. The reasoning in establishing the factor *c* which influences the working pressure, is that after nominal use the pipe may not be as sturdy as it was at the time of manufacture. Note that the joint factor, *e*, is not included for plain end non-ferrous pipe as listed in Table 2.

In addition to IPS copper pipe, several varieties of copper tubing are in use with either flared or compression couplings or soldered joints. Dimensions of copper water tube intended for plumbing, underground water service, fuel-oil lines, gas lines, etc., have been standardized by the U. S. Government and the American Society for Testing Materials.

Copper tube is classified in accordance with the wall thickness as follows:

Type K—Heavy Wall.
Type L—Medium Wall.
Type M—Light Wall.
Type DWV—Light Wall—for drainage, waste, and vent service.

In general, Type K is used for underground services and for general plumbing service where corrosion conditions are severe. Type L is used for general plumbing and heating service where conditions may be considered normal. Type M is used for sanitary drainage and other non-pressure applications, for heating, and sometimes for other services less severe than those for which Types K or L are recommended. Type DWV is of lighter weight than Type M, and is used for sanitary drainage and other non-pressure applications. Type DWV tube should not be used underground. Type L is suitable for underground drainage applications. Unless adequately protected, no copper tube should be embedded in a cinder fill. Where flexibility is essential as in hidden replacement work, or where as few joints as possible are desired as in fuel-oil lines, the soft temper is commonly used. In new or exposed work copper tube of a hard temper is generally used. All three types are extensively used with soldered fittings.

Standard dimensions, weights, and diameter and wall-thickness tolerances for these classes of copper tube are obtainable from Table 2. Copper pipe is also available with dimensions of steel pipe.

In refrigeration lines, used in connection with air-conditioning equipment, copper tube is used extensively. For refrigeration use where tube absolutely free from scale and dirt is required, bright annealed copper tube that has been cleaned, dehydrated, and sealed is used. This tube is available in a variety of sizes and wall thicknesses.

THREADING PRACTICE

In all threaded pipe for heating and ventilating installations the American Standard taper pipe thread, ASA B2.1-1945 is used. This thread is cut with a taper of 1 in 16 measured on the diameter of the pipe so as to secure a tight joint. The number of threads per inch varies with the pipe size. Threads for fittings are the same, except that it is regular practice to furnish straight tapped couplings for Schedule 40 pipe 2 in. and smaller. For steam pressures in excess of 25 psi, it is recommended that taper-tapped couplings be used to obtain a tight joint. These may be secured by ordering *line* pipe[a] which is used for oil piping, *the couplings of which* are provided with taper-tapped threads and may be used with regular mill-threaded standard weight pipe. Thread lengths should be in accordance with ASA B2.1. Right-hand threads are used unless otherwise ordered. To facilitate drainage, some elbows have the thread tapped at an angle to provide a pitch of the connecting pipe of ¼ in. to the foot. These elbows are known to the trade as

Table 2 .... Dimensions and Properties of Copper Tube

(All Types Except DWV are based on ASTM B-88)

Nominal Size	Type	Diameter OD in.	Diameter ID in.	Wall Thickness in.	Surface Area Sq Ft/Lin Ft OD	Surface Area Sq Ft/Lin Ft ID	Section Area Sq In. OD	Section Area Sq In. ID	Area of Metal Sq In.	Volume Gal/Lin Ft	Weight[a] Lb/Lin Ft	Working Pressure[b] Psig
¼	K[c]	0.375	0.305	0.035	0.0982	0.0798	0.110	0.0730	0.0374	0.00379	0.145	918
	L[c]	0.375	0.315	0.030	0.0982	0.0825	0.110	0.0779	0.0324	0.00404	0.126	764
⅜	K	0.500	0.402	0.049	0.131	0.105	0.196	0.127	0.0695	0.00660	0.269	988
	L	0.500	0.430	0.035	0.131	0.113	0.196	0.145	0.0512	0.00753	0.198	677
½	K	0.625	0.527	0.049	0.164	0.138	0.306	0.218	0.0887	0.0113	0.344	779
	L	0.625	0.545	0.040	0.164	0.143	0.306	0.233	0.0735	0.0121	0.285	625
⅝	K	0.750	0.652	0.049	0.193	0.171	0.441	0.334	0.108	0.0174	0.418	643
	L	0.750	0.666	0.042	0.193	0.174	0.441	0.348	0.0934	0.0181	0.362	547
¾	K	0.875	0.745	0.065	0.229	0.195	0.601	0.436	0.165	0.0227	0.641	747
	L	0.875	0.785	0.045	0.229	0.206	0.601	0.480	0.117	0.0250	0.455	497
1	K	1.125	0.995	0.065	0.295	0.260	0.994	0.778	0.216	0.0405	0.839	574
	L	1.125	1.025	0.050	0.295	0.268	0.994	0.852	0.169	0.0442	0.655	432
1¼	K	1.375	1.245	0.065	0.360	0.326	1.48	1.22	0.268	0.0634	1.04	466
	L	1.375	1.265	0.055	0.360	0.331	1.48	1.26	0.228	0.0655	0.884	387
	M	1.375	1.291	0.042	0.360	0.338	1.48	1.31	0.176	0.0681	0.682	293
	DWV	1.375	1.295	0.040	0.360	0.339	1.48	1.32	0.163	0.0684	0.650	—
1½	K	1.625	1.481	0.072	0.425	0.388	2.07	1.72	0.351	0.0894	1.36	421
	L	1.625	1.505	0.060	0.425	0.394	2.07	1.78	0.295	0.0925	1.14	359
	M	1.625	1.527	0.049	0.425	0.400	2.07	1.83	0.243	0.0950	0.940	289
	DWV	1.625	1.541	0.042	0.425	0.403	2.07	1.86	0.205	0.0969	0.809	—
2	K	2.125	1.959	0.083	0.556	0.513	3.56	3.01	0.532	0.157	2.06	376
	L	2.125	1.985	0.070	0.556	0.520	3.56	3.10	0.452	0.161	1.75	316
	M	2.125	2.009	0.058	0.556	0.526	3.56	3.17	0.377	0.164	1.46	255
	DWV	2.125	2.041	0.042	0.556	0.534	3.56	3.27	0.288	0.142	1.07	—
2½	K	2.625	2.435	0.095	0.687	0.638	5.41	4.66	0.755	0.242	2.93	352
	L	2.625	2.465	0.080	0.687	0.645	5.41	4.77	0.640	0.247	2.48	295
	M	2.625	2.495	0.065	0.687	0.653	5.41	4.89	0.523	0.254	2.03	234
3	K	3.125	2.907	0.109	0.818	0.761	7.67	6.64	1.03	0.345	4.00	343
	L	3.125	2.945	0.090	0.818	0.771	7.67	6.81	0.858	0.354	3.33	278
	M	3.125	2.981	0.072	0.818	0.780	7.67	6.98	0.691	0.362	2.68	220
	DWV	3.125	3.035	0.045	0.818	0.796	7.67	7.23	0.435	0.376	1.69	—
3½	K	3.625	3.385	0.120	0.949	0.886	10.3	9.00	1.32	0.468	5.12	324
	L	3.625	3.425	0.100	0.949	0.897	10.3	9.21	1.11	0.478	4.29	268
	M	3.625	3.459	0.083	0.949	0.906	10.3	9.40	0.924	0.489	3.58	218
4	K	4.125	3.857	0.134	1.08	1.01	13.3	11.7	1.68	0.607	6.51	135
	L	4.125	3.905	0.110	1.08	1.02	13.3	12.0	1.39	0.623	5.38	256
	M	4.125	3.935	0.095	1.08	1.03	13.3	12.2	1.20	0.634	4.66	217
	DWV	4.125	4.009	0.058	1.08	1.05	13.3	12.6	0.67	0.656	2.87	—
5	K	5.125	4.805	0.160	1.34	1.26	20.7	18.1	2.50	0.940	9.67	307
	L	5.125	4.875	0.125	1.34	1.28	20.7	18.7	1.96	0.971	7.61	234
	M	5.125	4.907	0.109	1.34	1.29	20.7	18.9	1.72	0.981	6.66	203
6	K	6.225	5.741	0.192	1.60	1.50	29.4	25.9	3.50	1.35	13.9	308
	L	6.125	5.845	0.140	1.60	1.53	29.4	26.8	2.63	1.39	10.2	221
	M	6.125	5.881	0.122	1.60	1.54	29.4	27.2	2.30	1.42	8.92	190
	DWV	6.125	5.959	0.083	1.60	1.56	29.4	27.9	1.58	1.55	6.10	—
8	K	8.125	7.583	0.271	2.13	1.99	51.8	45.2	5.69	2.34	25.9	330
	L	8.125	7.725	0.200	2.13	2.02	51.8	46.9	4.98	2.43	19.3	239
	M	8.125	7.785	0.170	2.13	2.04	51.8	47.6	4.25	2.47	16.5	200
10	K	10.125	9.449	0.338	2.65	2.47	70.5	70.1	10.4	3.65	40.3	332
	L	10.125	9.625	0.250	2.65	2.52	80.5	72.8	7.76	3.79	30.1	241
	M	10.125	9.701	0.212	2.65	2.54	80.5	73.9	6.60	3.84	25.6	202
12	K	12.125	11.315	0.405	3.17	2.96	115.	101.	14.9	5.24	57.8	334
	L	12.125	11.565	0.280	3.17	3.03	115.	105.	10.4	5.45	40.4	225
	M	12.125	11.617	0.254	3.17	3.04	115.	106.	9.47	5.50	36.7	204

Notes for Table 2

[a] Weight per foot is based on tube without couplings.

[b] Working pressure is based on the *American Standard Code for Pressure Piping*, published by the *ASME*, revised November, 1952 for plain end tubing (sweat joints).

$$P = \frac{2st_m}{D - 0.8t_m}$$

where P = allowable pressure, pounds per square inch, gage.
s = 6000 psi allowable fiber stress, pounds per square inch.
t_m = minimum wall thickness, inches.
D = maximum OD, inches.

[c] Types K and L furnished in both hard and soft tempers, Type M in hard only. Standard straight lengths are 20 ft. Standard coils (¼ to 1¼ inch) are 60 ft.

Table 3 .... Table of Availability for Steel Pipe

Material	Specification	Available Sizes (Inclusive) Inches Dia	Allowable Fiber Stress S[a] psi
Butt-weld	A-120	⅛ to 4	6,225
	A-53	⅛ to 4	6,750
Lap-weld	A-120	2 to 12	8,400
	A-53	2 to 24	9,000
Electric-resistance	A-53 Grade A	2 59 24	10,200[b]
	A-53 Grade B	2 to 24	12,750[b]
Seamless	A-53 Grade A	2 to 24	12,000
	A-53 Grade B	2 to 24	15,000
	A-120	2 to 12	10,400

[a] For use in the equation $P = \frac{2S(t_m - c)}{D - 0.8(t_m - c)}$

where P = working pressure, pounds per square inch.
S = fiberstress, pounds per square inch.
t_m = 0.875 x wall thickness, inches.
D = OD, inches.
c = joint factor = 0.05 for 1-in. and smaller size or 0.055 for larger than 1-in. size.

See *American Standard Code for Pressure Piping*, published by ASME, revised November, 1953.

[b] For electric resistance welded pipe for applications where the temperature is below 850 F, and where pipe furnished under this classification is subjected to supplemental tests and/or heat treatments as agreed to by the supplier and the purchaser, and whereby such supplemental tests and/or heat treatments demonstrate the strength characteristics of the weld to be equal to the minimum tensile strength specified for the pipe, the S values equal to the corresponding seamless grades may be used.

pitch elbows and are commercially available. All threaded pipe joints should be made up with a thread paste suitable for the service for which the pipe is to be used.

Table 4 contains pertinent threading data for standard weight, extra strong and double extra strong steel, and wrought-iron pipe. The normal engagement of pipe threads is given in Table 5.

PIPE FITTINGS

Pipe is joined by butt welding one length to another or by the use of fittings. Pipe fittings are made in a variety of forms (the common types are listed in Table 6), and are screwed, flanged, or welded. The type of fitting to be used is determined by the pressure of the fluid being carried, or by the intended use of the pipe line. The materials generally used are steel, cast iron, malleable iron (heat treated cast iron), copper, brass, stainless steel, alloy steel, or bronze. The material used also depends upon the pressure or characteristics of the fluid.

Fittings are designated and sized in accordance with American Standards Association specifications and are identified by their nominal pipe sizes.

In the case of reducing tees, crosses, and Y-branches (laterals), the size of the largest run opening is given first, followed by the size of the opening at the opposite end of the run. Where the fitting is a tee or Y-branch (lateral), the size of the outlet is given last. Where the fitting is a cross, the largest side-outlet opening is the third dimension given followed by the opening opposite. The straight line sketches, Fig. 1, illustrate how the reducing fittings are designated.

Where an external thread is wanted the word *male* follows the size of that opening.

Table 4 .... Threading Data for Pipe

Nominal Size[a]	Pipe OD	Standard Weight Threads per Inch[b]	Standard Weight Coupling OD	Standard Weight Coupling Length	Extra and Double Extra Strong Threads per Inch	Extra and Double Extra Strong Coupling OD	Extra and Double Extra Strong Coupling Length
⅛	0.405	27	0.563	1 3/16	27	0.563	1 1/16
¼	0.540	18	0.719	1 3/16	18	0.719	1⅝
⅜	0.675	18	0.875	1 3/16	18	0.875	1⅜
½	0.840	14	1.063	1 3/16	14	1.063	2⅛
¾	1.050	14	1.313	1⅜	14	1.313	2⅛
1	1.315	11½	1.576	2	11½	1.576	2⅝
1¼	1.660	11½	1.900	2 1/16	11½	2.054	2¾
1½	1.990	11½	2.200	2 1/16	11½	2.200	2¾
2	2.375	11½	2.750	2⅛	11½	2.875	2⅞
2½	2.875	8	3.250	3⅛	8	3.375	4⅛
3	3.500	8	4.000	3¼	8	4.000	4¼
3½	4.000	8	4.625	3⅜	8	4.625	4⅜
4	4.500	8	5.000	3½	8	5.200	4½
5	5.563[c]	8	6.296	3¾	8	6.296	4⅝
6	6.625	8	7.390	4	8	7.390	4⅞
8	8.625	—	—	—	8	9.625	5¼
10	10.750	—	—	—	8	11.750	5¾
12	12.750	—	—	—	8	14.000	6½
14	14.000	—	—	—	8	15.000	6¾
16	16.000	—	—	—	8	17.000	6¾
18	18.000	—	—	—	8	19.000	7⅛
20	20.000	—	—	—	8	21.000	7⅝

[a] All dimensions in inches.

[b] Taper of threads is ¾ in. per ft on diameter on all sizes of pipe, and in couplings of 2½ in. and over. Couplings 2 in. and smaller are straight tapped.

[c] 5.563 for copper and red brass pipe.

Note: The wall thickness and inside diameter shown in Table 1 do not apply to copper and red brass pipe.

Table 5 .... Normal Engagement of Tight-Fit Male and Female Pipe Threads

For American Standard and API Pipe Threads

Pipe Sizes	Engaged Length[a] In.	Pipe Sizes	Engaged Length[a] In.
⅛	¼	2½	1 5/16
¼	5/16	3	1
⅜	3/8	3½	1 1/16
½	½	4	1⅛
¾	9/16	5	1¼
1	1 1/16	6	1 5/16
1¼	1 1/16	8	1 7/16
1½	1 1/16	10	1⅝
2	¾	12	1¾

[a] Dimension shown is the normal tight-fit thread engagement from the leading edge of the female thread back to the leading edge of the male thread. No allowance has been made for threading variations.

Table 6 . . . . Pipe Fittings

90, 60, 45 & 22½° Standard	90 & 45° Flanged	
90 & 45° Street	90° Side outlet	
90 & 45° Male union	90° Railing	
90 & 45° Female union	90° Side outlet railing	
90 & 45° Butt-welding	Base	
90 & 45° Socket-welding	Drop	
Tees		
Standard	Service	Four-way
Flanged	Flanged side outlet	Male union
Female union	Female union outlet	Flanged reducing
Flanged base	Butt-welding	Socket-welding
Railing	Side outlet railing	Drop
Couplings		
Wrought	Cast	Socket-welding
Crosses		
Screwed	Flanged	Butt-welding
Y-Bends		
45° Screwed	45° Flanged	45° Double screwed / True flanged
Return Bends		
Butt-welding	Back outlet	Screwed / Union end
Reducers		
Standard	Eccentric socket-welding	Flanged paper but-welding / Flanged eccentric
Bushings		
Outside hexagon	Eccentric	Face
Caps		
Standard	Socket-welding	Butt-welding
Plugs		
Countersunk	Square-head	Bar
Nipples		
Close	Long	Tank
Unions		
Female	Male & female Flange	Tongue & groove flange
Bushings		
Outside	Eccentric	Face
Caps		
Standard	Socket-welding	Butt-welding
Plugs		
Countersunk	Square head	Bar
Nipples		
Close	Long	Tank
Unions		
Female	Male & female Flange	Tongue & groove flange
Flanges		
Blind	Screwed	Welding neck
Floor	Slip-on	Reducing

Allowable pressures for solder-type fittings are given in Table 7. Allowable pressures for ferrous screwed, flanged, and welding fittings according to ASA specifications are given in Table 8.

Steel welding (butt or socket type) fittings may be used for the same pressure as the pipe, providing wall thickness (or schedule number), and also the material, are the same as the pipe on which they are used.

Screwed fittings include: couplings, elbows (45° or 90°), return bends (close and open), tees, crosses, laterals (Y-branches), *plugs, caps, lock nuts, flanges, bushings,* and reducing fittings (elbows, tees, and couplings). Reducing fittings and bushings may be eccentrically tapped to permit drainage of condensate.

Sometimes nipples are erroneously called fittings. Actually a nipple is a piece of pipe less than 12 in. long that has been threaded on both ends. Pipe 12 in. or longer is regarded as *cut pipe*. Nipples are classified as *close* or full thread, *shoulder, short,* and *long*. A close nipple is about twice the length of the pipe thread since the threads actually meet. A shoulder nipple has a shoulder between the threads and is further classified as *short* or *long*. All shoulder nipples are cut to specific lengths and sold in that manner.

As a general rule, flanged joints are used when it is expected that the line will be disassembled quite often. There are three types of joints for attaching the flange to the pipe; screwed, welded, or lapped. Types of contact surface will be discussed in the next section.

Tables 9 to 15 give data on various types of ferrous fittings.

Fittings for copper tubing are available in soldered, flared, or compression types (see Figure 2). Fittings for copper pipe of IPS dimensions are available in screwed or soldered types. Table 16 *gives size data for screwed fittings* and Tables 17 and 18 give data for solder-type fittings for copper tube and pipe.

Table 7 . . . . Service Pressure Ratings for Wrought-Copper or Bronze Solder Joint Fittings[a]

Type of Solder	Service Temp., F	Maximum Service Pressure, Psig: Water ⅛ to 1 in.[b]	Water 1¼ to 2 in.[b]	Water 2½ to 2 in.[b]	Steam (All)
50-50 Tin-lead (ASTM B32 Alloy Grade 50A)	100	200	175[c]	150[d]	—
	150	150	125[c]	100[d]	—
	200	100	90[c]	75[d]	—
	250	85	75[c]	50[d]	15[e]
95-5 Tin-antimony or 95-5 Lead-tin (ASTM B32 Alloy Grade 5A)	100	500	400	300	—
	150	400	350	275	—
	200	300	250	200	—
	250	200	175	150	15
Solders with melting point 1100 F or above	350	270	190	155	120

Note: Ratings other than specified here may be used upon the recommendation of the manufacturer as to the proper solders that should be used.

[a] Extracted from *American Standard for Wrought Copper and Bronze Solder Joint Fittings*, B16.22-1951.

[b] Standard water tube sizes except 1.3 in., which is ¼ in. OD seamless copper tubing for refrigerator service, etc. (*ASTM* B68).

[c] These pressures may be used for cast-brass fittings (*ASA* B16.18-1950) for sizes 2 in. and smaller.

[d] These pressures may be used for cast-brass fittings (*ASA* B16.18-1950) for sizes 2½ in. and above.

[e] Also allowable for cast-brass fittings (*ASA* B16.18-1950).

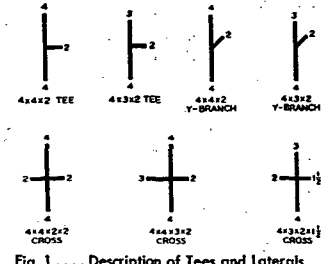


Fig. 1 . . . . Description of Tees and Laterals (ASA B16.3—1951)

Table 8 . . . . ASA Specifications and Pressures for Ferrous Screwed, Flanged, and Welded Fittings

Type of Fitting	Specification	Saturated Steam Pressure Psig[a]
Cast Iron—Flanged	B16.1	125
Cast Iron—Flanged	B16b	250
Cast Iron—Flanged	B16b1	800
Cast Iron—Flanged (refrigeration)	B16.16	300
Cast Iron—Screwed	B16.4	125 and 150
Malleable Iron—Screwed	B16.3	150
Malleable Iron—Screwed	B16.19	300
Brass or Bronze—Screwed	B16.15	125
Brass or Bronze—Screwed	B16.17	250
Steel—Flanged	B16.5	2500

[a] Pressure may reduce with increasing sizes. This pressure is the maximum allowed in this specification for saturated steam. Higher pressures may be allowed with lower temperatures.

The compression-type fitting is generally limited to small sizes of tube, while the flared and soldered types are used in both large and small sizes. An American Standard, ASA A40.2 (1936), has been prepared to standardize dimensions of brass fittings for flared copper tube. Flared tube fittings are widely used in refrigeration work where SAE dimensions and a 45-deg flare render most fittings interchangeable, although for refrigeration use, thread fits and tolerances on thread gages must be maintained within close limits. Brass fittings with SAE dimensions are not interchangeable with the American Standard fittings for water tube.

FLANGE FACINGS AND GASKETS

There are many types of flange facings. The simplest, and the one most commonly used, is the plain face. This type of face is used with fittings shown in Table 10. Other facings, with varying complexity, are used for higher pressures. They include: male and female face (an insert type face), tongue-and-groove (an adaptation of the male and female type), *lapped pipe to flange, lapped pipe to pipe,* and *ring* and groove using neck flanges.

As it is too expensive to machine flange faces sufficiently to provide watertight joints, a gasket is usually inserted between the contact faces of the flanges. The flange bolts then draw the faces up so tight that the gasket deforms into the face imperfections and thereby provides a tight joint.

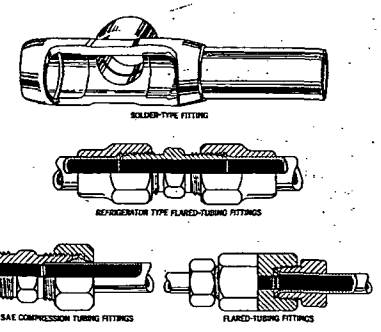
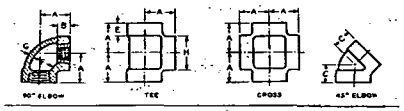


Fig. 2 . . . . Copper or Brass Tube Fittings

Gaskets are made of numerous materials, such as cork, paper, fiber, asbestos, metal (iron, steel, copper, etc.), and combinations of metal and some soft material. The proper gasket material to use depends upon the service of the pipe. The material should not be detrimentally affected by the chemical and thermal conditions of the fluid being carried.

Table 9 . . . . Dimensions of Screwed Cast-Iron 90- and 45-Deg Elbows, Tees, and Crosses (Class 125)

(*Straight Sizes*)



Nominal Pipe Size	Center to End, Elbows, Tees, and Crosses A	Center to End, 45-Deg Elbows C	Length of Thread, Min B	Width of Band, Min E	Inside Diameter of Fitting F Max	F Min	Metal Thickness G	Outside Diameter of Band, Min H
¼	0.81	0.73	0.32	0.38	0.584	0.540	0.110	0.93
⅜	0.95	0.80	0.36	0.44	0.719	0.675	0.120	1.12
½	1.12	0.88	0.43	0.50	0.897	0.840	0.130	1.34
¾	1.31	0.98	0.50	0.56	1.107	1.050	0.155	1.63
1	1.50	1.12	0.58	0.62	1.385	1.315	0.170	1.95
1¼	1.75	1.29	0.67	0.69	1.730	1.660	0.185	2.39
1½	1.94	1.43	0.70	0.75	1.970	1.900	0.200	2.68
2	2.25	1.68	0.75	0.84	2.445	2.375	0.220	3.28
2½	2.70	1.95	0.92	0.94	2.975	2.875	0.240	3.86
3	3.08	2.17	0.98	1.00	3.600	3.500	0.260	4.62
3½	3.42	2.39	1.03	1.06	4.100	4.000	0.280	5.20
4	3.79	2.61	1.08	1.12	4.600	4.500	0.310	5.79
5	4.50	3.05	1.18	1.18	5.663	5.563	0.380	7.05
6	5.13	3.46	1.28	1.28	6.725	6.625	0.430	8.28
8	6.56	4.28	1.47	1.47	8.725	8.625	0.550	10.63
10	8.08*	5.16	1.68	1.68	10.850	10.750	0.690	13.12
12	9.50*	5.97	1.88	1.88	12.850	12.750	0.800	15.47

From *American Standard Dimensions of Cast-Iron Screwed Fittings*, Class 125, *AAS* B16.4-1949. All dimensions given in inches.

* This applies to elbows and tees only.

VALVES

Valves are made with the same end connections as fittings. A brief description of each type follows.

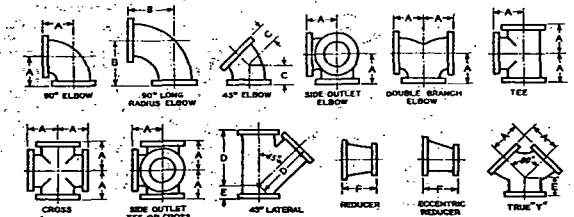
1. *Flanged Ends:* Made in sizes from ½ in. up. Used whenever line fittings are flanged and if frequent removal is contemplated. Flanged valves facilitate installation when large sizes are used.

2. *Screwed Ends:* Most common-type end. Can be used for all pressures. Usually confined to smaller size because of difficulty in making large screwed joints.

3. *Welding Ends:* Available in steel only. Generally employed for high-temperature, high-pressure work. Difficult to disassemble in a welded line. This type includes butt-welding or socket-welding. See Tables 11 and 14 for size of fittings available in each class. The valves are made in the same range.

4. *Brazing Ends:* These ends are available on brass valves. The ends have a special shape to facilitate use of brazing alloys. These valves are used for higher temperatures than solder end valves.

Table 10 . . . Dimensions of Flanged Cast-Iron Elbows, Double Branch Elbows, Tees, Crosses, Laterals, True Y's (Straight Sizes), and Reducers (Class 125)



Nominal Pipe Size	Inside Dia of Fittings	Center to Face 90 Deg Elbow, Tees, Crosses, True Y, and Double Branch Elbow A	Center to Face 90 Deg Long Radius Elbow B	Center to Face 45 Deg Elbow C	Center to Face Lateral D	Short Center to Face True Y and Lateral E	Face to Face Reducer F	Dia of Flange	Thickness of Flange (Min)	Wall Thickness
1	1	3½	5	1¾	5⅝	1¾	....	4¼	7/16	5/16
1¼	1¼	3¾	5½	2	6¼	1¾	....	4⅝	½	5/16
1½	1½	4	6	2¼	7	2	....	5	9/16	5/16
2	2	4½	6½	2½	8	2½	5	6	5/8	5/16
2½	2½	5	7	3	9½	2½	5½	7	11/16	5/16
3	3	5½	7¾	3	10	3	6	7½	¾	5/8
3½	3½	6	8½	3½	11½	3	6½	8½	13/16	7/16
4	4	6½	9	4	12	3	7	9	15/16	½
5	5	7½	10¼	4½	13½	3½	8	10	15/16	½
6	6	8	11½	5	14½	3½	9	11	1	9/16
8	8	9	14	5½	17½	4½	11	13½	1⅛	5/8
10	10	11	16½	6½	20½	5	12	16	1 5/16	¾
12	12	12	19	7½	24½	5½	14	19	1¼	13/16
14 OD	14	14	21½	7½	27	6	16	21	1⅜	⅞
16 OD	16	15	24	8	30	6½	18	23½	1 7/16	1
18 OD	18	16½	26½	8½	32	7	19	25	1 9/16	1 1/16
20 OD	20	18	29	9½	35	8	20	27½	1 11/16	1⅛
24 OD	24	22	34	11	40½	9	24	32	1⅞	1¼
30 OD	30	25	41½	15	49	10	30	38¾	2⅛	1 7/16
36 OD	36	28*	49	18	....	....	36	46	2⅜	1⅝
42 OD	42	31*	56½	21	....	....	42	53	2⅝	1 13/16
48 OD	48	34*	64	24	....	....	48	59½	2¾	2

From *American Standard for Cast Iron Pipe Flanges and Flanged Fittings*, Class 125, *ASA* B16.1-1948. All dimensions given are in inches.
* Does not apply to true Y's or double-flanged elbows.

5. *Solder Ends:* For use with copper tube in plumbing, heating, and air conditioning. Should not be used at high temperatures.

6. *Flared Ends:* For use with light-wall tubing, either metal or plastic, and for sizes 2 in. and below. A flaring tool flares the tube end, and a ring nut pulls the flared tubing up against a seat on the valve. This makes a lap joint.

7. *Hub Ends:* Generally used for water supply and sewage piping. This type joint is used with cast-iron pipe. The joint is a socket type made by inserting the pipe in the hub, then caulking with oakum. After caulking, the joint is sealed with molten lead.

Valves are made of brass, iron, steel, and to some extent malleable iron. Brass should not be used for work requiring temperatures above 550 F. The brass used in valves is similar to that used in brass pipe and contains some tin and lead as well as copper and zinc.

Cast iron is generally provided in some alloyed state, especially in the larger sizes. Cast iron alone is not suitable for service (above 450 F).

Malleable iron, which is a cast iron heat-treated to provide some ductility, is used to some extent, primarily in place of cast iron.

Steel is used either in the cast or forged state. For corrosive service, alloy and stainless steel are used. Steel is used for high temperature, high-pressure work. Steel is the only valve material that can stand appreciable expansion stresses.

The characteristics of valves restrict their use. The five main uses for valves are listed below:

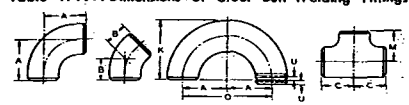
1. *Stopping flow:* The most general use for valves. *Gate* valves are generally used for this purpose. The gate valve operates by a disc moving at right angles to the path of the fluid. The fluid flows straight through the valve, hence, a gate valve offers the minimum resistance when it is fully open. The gate is moved by a stem screw attached to the handle. Gate valves are poor regulators because at nearly closed position (where best regulation is obtained) the fluid velocity either causes vibration and noise, or else tends to erode the disc.

2. *Throttling flow:* This is best accomplished by *globe* or *angle* valves. These valves cause the fluid to change direction, and consequently, offer considerable fluid resistance even when in the open position. Both the globe and angle valves operate by closing a plug into a fitted seat. The angle valve is similar to the globe valve except that the outlet face is at an angle of 90 deg from the inlet face. An angle valve takes the place of a globe valve and an elbow.

3. *Checking flow:* Preventing back flow is called checking, hence, a *check* valve is used. The basic principle of all check valves is that a reversal of flow will close the valve. Check valves are either *swing check* or *lift check*. With a swing check the fluid presses on a hinged flap. If the flow stops or reverses, the flap swings down and seats itself to close the valve. A lift check operates similarly except that the flap is replaced by a plug that is lifted—through guides—by the force of the fluid flow. The fluid changes direction of flow, as in a globe valve, and transmits a vertical thrust. If the flow stops, gravity pulls the plug downward and closes the valve. A reversal of flow will close the valve.

Two other types of valves are the relief valve and the pressure regulating valve. Relief valves are usually spring loaded in order that some predetermined pressure can force the valve to open and relieve the pressure. Pressure regulators are used to reduce a high incoming pressure to a required service pressure. These valves are designed to eliminate a fluctuation in the incoming pressure and thereby maintain a steady, lower service pressure (see Chapter 26).

Table 11 . . . Dimensions of Steel Butt-Welding Fittings



Nominal Pipe Size	Long Radius Elbows: Outside Diameter at Bevel	Long Radius Elbows: Center-to-End 90-Deg Elbows A	Long Radius Elbows: Center-to-End 45-Deg Elbows B	180 Deg Returns: Outside Diameter at Bevel	180 Deg Returns: Center-to-Center O	180 Deg Returns: Back to Face K	Straight Tees: Outside Diameter at Bevel	Straight Tees: Center-to-End Run C	Straight Tees: Center-to-End Outlet M
1	1.315	1½	7/8	1.315	3	2 5/16	1.315	1½	1½
1¼	1.660	1⅞	1	1.660	3¾	2¾	1.660	1⅞	1⅞
1½	1.900	2¼	1⅛	1.900	4½	3¼	1.900	2¼	2¼
2	2.375	3	1⅜	2.375	6	4 5/16	2.375	2½	2½
2½	2.875	3¾	1¾	2.875	7½	5 5/16	2.875	3	3
3	3.500	4½	2	3.500	9	6¼	3.500	3½	3½
3½	4.000	5¼	2¼	4.000	10½	7¼	4.000	3¾	3¾
4	4.500	6	2½	4.500	12	8¼	4.500	4½	4½
5	5.563	7½	3½	5.563	15	10 5/16	5.563	4⅞	4⅞
6	6.625	9	3¾	6.625	18	12 5/16	6.625	5⅝	5⅝
8	8.625	12	5	8.625	24	16 5/16	8.625	7	7
10	10.750	15	6¼	10.750	30	20⅜	10.750	8½	8½
12	12.750	18	7½	12.750	36	24⅜	12.750	10	10
14	14.000	21	8¾	14.000	42	28	14.000	11	
16	16.000	24	10	16.000	48	32	16.000	12	Not
18	18.000	27	11¼	18.000	54	36	18.000	13½	stand-
20	20.000	30	12½	20.000	60	40	20.000	15	ard
24	24.000	36	15	24.000	72	48	24.000	17	

From *American Standard for Steel Butt-Welding Fittings*, *ASA* B16.9-1951. All dimensions are in inches. Dimension A is equal to ½ of dimension O.

WELDING

Erection of piping in heating and ventilating installations by means of fusion welding has been commonly accepted in the past few years as an alternate method to the screwed and flanged joint. Since the question of economy of welding as against the use of screwed and flanged fittings is dependent on the individual job, the use of welding is generally recommended on the basis of: greatly reduced cost of maintenance and repair; reduced weight resulting from the use of a lighter-weight pipe; or increased economy in pipe insulation, hangers, and supports rather than any economy that might be effected in actual erection by welding on low to medium pressure heating jobs.

Fusion welding, commonly used in erection of piping, is defined as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Fusion welding embraces gas welding and electric arc welding, both of which are commonly used to produce acceptable welds. Welding processes and procedure are described in various publications.[8]

Welding application requires the same basic knowledge of design as do the other types of assembly, but, in addition, requires a generous knowledge of welding principles and particularly the welding qualities of metal and its reaction to extremely high temperatures. The ability to select and use the proper welding rods is important. This requirement applies equally to employer and employee, with the employer accepting all of the responsibility. Thus the employer should select his welding mechanics with good judgment, provide them with first-class equipment and tools, arrange for their training and use of acceptable workmanship standards, and at regular intervals subject their work to prescribed tests.

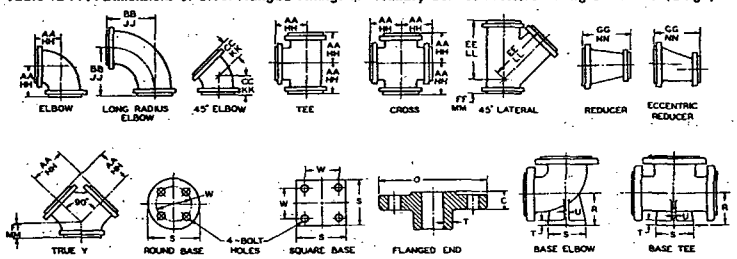
Rules for fusion welding of pipe joints, the qualification of welding operators, welding procedures, and testing are contained in the *Standard Manual on Pipe Welding*[8] and in publications of other groups.[7, 8, 9] In general, the wall thickness and chemical analysis of the pipe are the governing factors, not the working pressure. There are a number of safety codes which govern the installation of welded piping in many cities and states.

A complete line of manufactured steel-welding fittings is now available, and a dimensional standard has been prepared under the procedure of the American Standards Association, (see Table 11) to unify heretofore divergent dimensions for the same type welding fittings produced by different manufacturers. Standard dimensions for steel butt-welding elbows, tees, caps, and lapped-joint stub ends are given in Table 11. Dimensions for eccentric and concentric reducers are not shown in Table 11, but are included in the standard.

Steel butt-welding fittings have beveled ends. For wall thicknesses less than 5/16 in., the bevel may be either 37½ deg. or cut square. For wall thicknesses of 5/16 in. to and including ¾ in. the bevel is 37½ deg. For walls larger than ¾ in. but not greater than 1¾ in., the bevel is U-shaped with a slope of 20 deg above the 5/16 in. radius fillet. For walls greater than 1¾ in., the welding ends are prepared as agreed between the manufacturer and the purchaser.

Table 19 contains data on the dimensions of flanged and welding end valves. Dimensions of other type valves can be obtained by writing to the Manufacturers Standardization Society of the Valve and Fitting Industry.

Table 12 .... Dimensions of Steel Flanged Fittings for Primary Service Pressure Rating of 150 Psi (Gage)



1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
			1/16 in. Raised Face (Flange Edge)						Ring Joint									Base Drilling	
Nominal* Pipe Size	Outside Diameter of Flange	Thickness of Flange, Min	Center to Contact Surface of Raised Face, Elbow, Tee, Cross, and True Y	Center to Contact Surface of Raised Face, Long Radius Ell	Center to Contact Surface of Raised Face 45° Ell	Long Center to Contact Surface of Raised Face, Lateral	Short Center to Contact Surface of Raised Face, Lateral, and True Y	Center to Contact Surface of Raised Face, Reducer	Center to End, Elbow, Tee, Cross, and True Y	Center to End, Long Radius Ell	Center to End, 45-Deg Elbow	Long Center to End, Lateral	Short Center to End, Lateral, and True Y	Center to Base	Dia. of Round Base or Width of Square Base	Thickness of Base	Thickness of Ribs	Bolt Circle or Bolt Spacing	Dia. of Drilled Holes
	O	C	AA	BB	CC	EE	FF	GG	HH	JJ	KK	LL	MM	R	S	T	U	W	
1	4¼	7/16	3½	5	1¾	5¾	1¾	4½	3¾	5¼	2	6	2						
1¼	4⅝	½	3¾	5½	2	6¼	1¾	4½	4	5¾	2¼	6½	2						
1½	5	9/16	4	6	2¼	7	2	4½	4¼	6¼	2½	7¼	2¼						
2	6	5/8	4½	6½	2½	8	2½	5	4¾	6¾	2¾	8¼	2¾	4⅛	4⅝	⅛	½	3½	⅝
2½	7	11/16	5	7	3	9½	2¾	5½	5¼	7¼	3¼	9¾	2¾	4½	4⅝	⅛	½	3½	⅝
3	7½	¾	5½	7¾	3	10	3	6	5¾	8	3¼	10¼	3¼	4⅞	5	9/16	½	3⅞	⅝
3½	8½	13/16	6	8½	3½	11½	3	6½	6¼	8¾	3¾	11¾	3¼	5¼	5	9/16	½	3⅞	⅝
4	9	15/16	6½	9	4	12	3	7	6¾	9¼	4¼	12¼	3¼	5½	6	⅝	½	4¾	¾
5	10	15/16	7½	10¼	4¼	13½	3½	8	7¾	10½	4¾	13¾	3¾	6¼	7	11/16	⅝	5½	¾
6	11	1	8	11½	5	14½	3½	9	8¾	11¾	5¼	14¾	3¾	7	7	11/16	⅝	5½	¾
8	13½	1⅛	9	14	5½	17½	4½	11	9¼	14¼	5¾	17¾	4¾	8⅜	9	15/16	⅞	7¼	¾
10	16	1⅜	11	16½	6¼	20½	5	12	11¼	16¾	6¾	20¾	5¼	9¾	9	15/16	⅞	7½	¾
12	19	1⅜	12	19	7½	24½	5½	14	12¼	19¼	7¾	24¾	5¾	11¼	11	1	1	9½	⅞
14	21	1⅜	14	21½	7½	27	6	16	14¼	21¾	7¾	27¼	6¼	12½	11	1	1	9½	⅞
16	23½	1⅝	15	24	8	30	6½	18	15¼	24¼	8¼	30¼	6¾	13¾	11	1	1	9½	⅞
18	25	1⅝	16½	26½	8½	32	7	19	16¾	26¾	8¾	32¼	7¼	15	13¼	1⅛	1⅛	11¾	⅞
20	27½	1 11/16	18	29	9½	35	8	20	18¼	29¼	9¾	35¼	8¼	16	13½	1⅛	1⅛	11¾	⅞
24	32	1⅞	22	34	11	40½	9	24	22¼	34¼	11¼	40¾	9¼	18½	13½	1⅛	1⅛	11¾	⅞

From *American Standard for Steel Pipe Flanges and Flanged Fittings*, 150, 300, 400, 600, 900, 1500, and 2500 Lb. *ASA* B16.5-1953. All dimensions given in inches.
* Actual diameter is same for sizes 1 to 12 in. and is ¾ in. less than nominal for sizes 14 to 24 in.

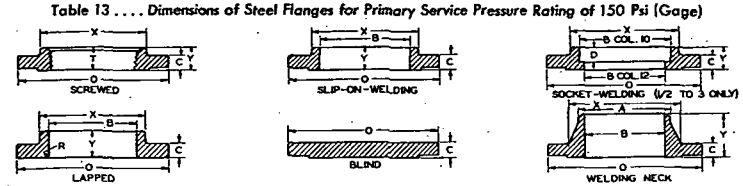
Flange fittings may be welded from sizes ½ to 24 in. inclusive (see Table 13).

Socket-welding fittings also are commercially available. These fittings have a machined recess for inserting the pipe which is attached by a fillet weld between the pipe wall and socket end. Use of socket-welding fittings generally is restricted to nominal pipe sizes 3 in. and smaller in which range commercial fittings are available. This type of fitting has gained rapid acceptance owing to its ease of installation, low cost, and ability to make a pressure-tight joint without weakening the pipe, as is the case with threading. Dimensions for socket-welding fittings, in accordance with ASA Standard B16.11-1946, are given in Table 14.

EXPANSION AND FLEXIBILITY

Changes in temperature cause a change in dimensions of any matter. The metals that are used for pipe have the same characteristics in that they expand with increasing temperature. If the metal is constrained so that it cannot expand, then an internal compressive stress is set up in the material.

Table 13 .... Dimensions of Steel Flanges for Primary Service Pressure Rating of 150 Psi (Gage)



1	2	3	4	5	6	7	8	9	10	11	12	13	14
					Length Through Hub Y				Bore B				
Nominal Pipe Size	Outside Diameter of Flange	Thickness of Flange, Min	Diameter of Hub	Hub Diameter Beginning of Chamfer Welding Neck	Screwed slip-on socket welding	Lapped	Welding neck	Thread Length Screwed	Slip-on socket welding, min	Lapped, min	Welding neck socket welding	Corner Radius of Bore of Lapped Flange and Pipe	Depth of Socket
	O	C	X	A				T				R	D
½	3½	7/16	1 3/16	0.84	5/8	5/8	17/8	5/8	0.88	0.90	0.62	1/8	3/8
¾	3⅞	½	1½	1.05	5/8	5/8	2 1/16	5/8	1.09	1.11	0.82	1/8	7/16
1	4¼	9/16	1 13/16	1.32	1 1/16	1 1/16	2 3/16	1 1/16	1.36	1.38	1.05	1/8	1/2
1¼	4⅝	5/8	2 5/16	1.66	1 3/16	1 3/16	2¼	1 3/16	1.70	1.72	1.38	5/32	9/16
1½	5	1 1/16	2 9/16	1.90	7/8	7/8	2 5/16	7/8	1.95	1.97	1.61	1/4	5/8
2	6	¾	3 1/16	2.38	1	1	2½	1	2.44	2.46	2.07	5/16	11/16
2½	7	7/8	3 5/16	2.88	1⅛	1⅛	2¾	1⅛	2.94	2.97	2.47	5/16	¾
3	7½	15/16	4¼	3.50	1 5/16	1 5/16	2¾	1 5/16	3.57	3.60	3.07	3/8	13/16
3½	8½	15/16	4 13/16	4.00	1¼	1¼	2 13/16	1¼	4.07	4.10	3.55	3/8	
4	9	15/16	5 5/16	4.50	1 5/16	1 5/16	3	1 5/16	4.57	4.60	4.03	7/16	
5	10	1 3/16	6 7/16	5.56	1 3/16	1 7/16	3½	1 3/16	5.66	5.69	5.05	7/16	
6	11	1	7 5/16	6.63	1 5/16	1 5/16	3½	1 5/16	6.72	6.75	6.07	1/2	
8	13½	1⅛	9 1/16	8.63	1¾	1¾	4	1¾	8.72	8.75	7.98	1/2	
10	16	1 3/16	12	10.75	1 13/16	1 13/16	4	1 13/16	10.88	10.92	10.02	1/2	
12	19	1¼	14⅜	12.75	2 3/16	2 3/16	4½	2 3/16	12.88	12.92	12.00	1/2	
14	21	1⅜	15¾	14.00	2¼	3⅛	5	2¼	14.14	14.18	To be	1/2	
16	23½	1 7/16	18	16.00	2½	3 5/16	5	2½	16.16	16.19	speci-	1/2	
18	25	1 5/16	19⅝	18.00	2 11/16	3 13/16	5½	2 11/16	18.18	18.20	fied by	1/2	
20	27½	1 11/16	22	20.00	2⅞	4 1/16	5 11/16	2⅞	20.20	20.25	pur-	1/2	
24	32	1⅞	26⅛	24.00	3¼	4⅜	6	3¼	24.25	24.25	chaser	1/2	

From *American Standard for Steel Pipe Flanges and Flanged Fittings*, *ASA* B16.5-1953. All dimensions given in inches.

This stress can be calculated, just as any other stress, by using Hooke's Law.

$$F = \frac{eEA}{L} \qquad (1)$$

where

F = Force of constraint, pounds.
e = Deformation, inches.
E = Modulus of elasticity, pounds per square inch.
A = Area of metal, square inches.
L = Length of metal, inches.

This equation is valuable in determining the internal stress of the pipe and also the constraining force on the anchors. This equation assumes that the stress is less than the yield strength of the metal. In other words, the metal will return to its original dimension when the temperature returns to its original level.

Materials such as cast iron are not *ductile*, hence the *yield strength* is approximately equal to the ultimate strength. These materials have excellent compressive strength but poor tensile strength. For this reason, it is difficult to bend these materials appreciably. Cast iron, for example, could be put in tension and, from Table 20, it is seen that for a temperature *change of 100 deg, a compressive stress of 7140 psi would be* developed. If the pipe were straight, this stress would be handled easily. If, however, the stress caused a bending moment, then fracture might occur.

The values of stress found in Table 20 were determined by using Equation 1 and substituting Equation 2 for e

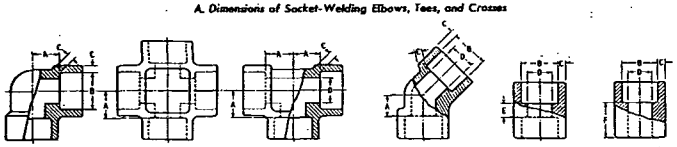
$$e = \alpha L \Delta t \qquad (2)$$

where

α = coefficient of linear expansion, inches per inch.
Δt = temperature change, Fahrenheit degrees.

Table 14 . . . . Dimensions of Socket-Welding Fittings

A. Dimensions of Socket-Welding Elbows, Tees, and Crosses



Nominal Pipe Size	Depth of Socket, Min	Center to Bottom of Socket: Sched 40 and 80	Center to Bottom of Socket: Sched 160	Bore Diameter of Socket, Min	Socket Wall Thickness, Min: Sched 40	Socket Wall Thickness, Min: Sched 80	Socket Wall Thickness, Min: Sched 160	Bore Diameter of Fitting: Sched 40	Bore Diameter of Fitting: Sched 80	Bore Diameter of Fitting: Sched 160
		A		B	C			D		
⅛	⅜	7/16	....	0.420	0.125	0.125	....	0.269	0.215	....
¼	⅜	7/16	....	0.555	0.125	0.149	....	0.364	0.302	....
⅜	⅜	13/32	....	0.690	0.125	0.158	....	0.493	0.423	....
½	⅜	⅜	¾	0.855	0.136	0.184	0.234	0.622	0.546	0.466
¾	½	¾	⅞	1.065	0.141	0.193	0.273	0.824	0.742	0.614
1	½	⅞	1 1/16	1.330	0.166	0.224	0.313	1.049	0.957	0.815
1¼	½	1 1/16	1¼	1.675	0.175	0.239	0.313	1.380	1.278	1.160
1½	½	1¼	1½	1.915	0.181	0.250	0.351	1.610	1.500	1.338
2	⅝	1½	1⅝	2.406	0.193	0.273	0.429	2.067	1.939	1.689
2½	⅝	1⅝	2¼	2.906	0.254	0.345	0.469	2.469	2.323	2.125
3	⅝	2¼	2½	3.535	0.270	0.375	0.546	3.068	2.900	2.626

B. Dimensions of Socket-Welding 45 Deg Elbows, Couplings, and Half Couplings

Nominal Pipe Size	Depth of Socket, Min	Center to Bottom of Socket for 45 Deg Ells: Sched 40 and 80	Center to Bottom of Socket for 45 Deg Ells: Sched 160	Couplings Distance Between Bottoms of Sockets	Half Couplings, Bottom of Socket to Opposite Face	Bore Diameter of Socket, Min	Socket Wall Thickness, Min: Sched 40	Socket Wall Thickness, Min: Sched 80	Socket Wall Thickness, Min: Sched 160	Bore Diameter of Fitting: Sched 40	Bore Diameter of Fitting: Sched 80	Bore Diameter of Fitting: Sched 160
		A		E	F	B	C			D		
⅛	⅜	5/16	....	¼	⅝	0.420	0.125	0.125	....	0.269	0.215	....
¼	⅜	5/16	....	¼	⅝	0.555	0.125	0.149	....	0.364	0.302	....
⅜	⅜	5/16	....	¼	1 1/16	0.690	0.125	0.158	....	0.493	0.423	....
½	⅜	7/16	½	⅜	⅞	0.855	0.136	0.184	0.234	0.622	0.546	0.466
¾	½	½	9/16	⅜	1 5/16	1.065	0.141	0.193	0.273	0.824	0.742	0.614
1	½	9/16	1 1/16	½	1⅛	1.330	0.166	0.224	0.313	1.049	0.957	0.815
1¼	½	1 1/16	1 3/16	½	1 3/16	1.675	0.175	0.239	0.313	1.380	1.278	1.160
1½	½	1 3/16	1	½	1¼	1.915	0.181	0.250	0.351	1.610	1.500	1.338
2	⅝	1	1⅛	¾	1⅝	2.406	0.193	0.273	0.429	2.076	1.939	1.689
2½	⅝	1⅛	1¼	¾	1 11/16	2.906	0.254	0.345	0.469	2.469	2.323	2.125
3	⅝	1¼	1⅜	¾	1¾	3.535	0.270	0.375	0.546	3.068	2.900	2.626

From *American Standard for Steel Socket-Welding Fittings, ASA* B16.11-1946. All dimensions are in inches.

This gives Equation 3

$$F = \alpha E A \Delta t. \qquad (3)$$

Values for α and E are given in Table 21 for various pipe materials, to facilitate determination of the stress in an actual pipe. The following example illustrates the computation of stress.

Example 1: What is the force set up by a 1-in. Schedule 40 steel pipe if the thermal expansion from a 100 deg temperature increase is fully constrained?

Answer: From Table 1, determine that for a 1-in. Schedule 40 steel pipe, the metal area = 0.494 sq in. Then, from Table 21, αE = 195 psi, and from Equation 3

$$F = 195 \times 0.494 \times 100$$

$$= 9630 \text{ lb.}$$

It is interesting to note that length does not enter into the determination of the constraining force. This is so because the stress is a *unit length* function and so is the expansion; therefore, they cancel each other. In other words, if a weight stretches a 10-foot wire 0.10 in., then that *same* weight would stretch a 100-foot wire one inch. In both cases the elongation was 1 part in 1200. The realization that thermal *stress* is independent of length is important, but often overlooked. Nonetheless, that concept explains the feasibility of utilizing the inherent flexibility of the pipe to take care of expansion.

Regardless of the length of pipe between anchors, it is possible to take care of the expansion by putting the force of expansion into the pipe as internal stress.

If the pipe is not straight, then the force of expansion will cause a bending moment. The pipe will then be under a combined stress. The stresses will be longitudinal and transverse, and to a small extent radial. All of these stresses tend to fracture the pipe, and therefore must be added to make up what is called *allowable combined stress*. The *Code for Pressure Piping*, ASA B31.1, published by the ASME, sets up very definite limits for the allowable combined stresses for various pipe materials. For temperatures below 400 F and for piping that is for neither district heating nor power, any standard pipe material may be used. For high-temperature or high-pressure work, some text on expansion and flexibility should be consulted. It is beyond the scope of this chapter to develop the theory of expansion caused by high temperatures.

For the simpler problems encountered for temperatures 400 F and less, which the heating engineer is more apt to encounter, there are four methods of allowing for expansion.

The first method is to use *packless* expansion joints. These joints include types of bellows expansion joints, rubber, corrugated copper and other metals.

Rubber-type joints are generally used in vacuum or low-pressure steam lines. Maximum temperature is generally 180 F. If oil is present in the line, rubber-type joints will be attacked. Travel of these joints is up to 1 inch. The travel to be expected can be calculated by using Equation 2.

The second type of expansion joint is called a *slip* joint. This joint allows for expansion by a sliding of a female member over a male member. The joint is kept tight by means of packing. The packing determines the limit of the temperature to which the joint may be subjected. The main disadvantage to this joint is that it must be continually inspected for a deterioration of the packing. By using double slip joints, the travel may go to several feet, but generally a single joint allows about a foot of expansion. It is common practice to use slip joints up to 250 psi.

The most common method of allowing for expansion in heating systems is to use the swivel joint. This is the third method. The swivel joint was first used with screwed fittings, but it is also used now with welded fittings. With welded elbows, the swivel introduces torsional stress in the elbow and in the swing piece. This type of joint is adequate for taking up the expansion in a header, and preventing fracture of the riser or heating element.

The fourth method is to allow the flexibility of the pipe to absorb the stress of expansion. A technique used to reduce the stress in the expanded pipe is to *cold spring* it before hooking it up. Cold springing can be used to absorb about one-half the stress. The technique is to install the pipe with an opposite stress. For example, if the pipe were to have a compression of 2000 psi, then the pipe would be installed with 1000 psi of tension. For refrigeration, the pipe would be cold sprung in compression.

For conditions not requiring rigorous analysis, it is permissible to use expansion bends designed by means of Equation 4

$$L = 6.16\sqrt{D\Delta} \qquad (4)$$

where

L = length of pipe, feet.

D = OD of pipe, inches.

Δ = deformation, inches (see Equation 2) fiber stress ≤ 16,000 pounds per square inch.

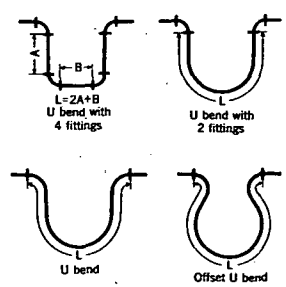
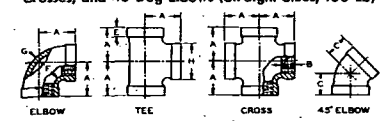


Fig. 3 . . . . Measurement of L on Various Pipe Bends

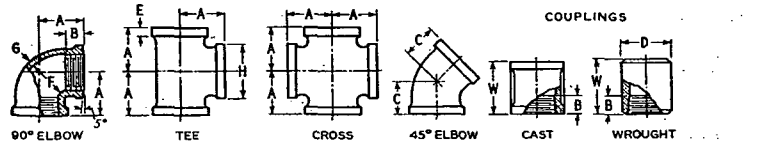
Table 15 . . . . Dimensions of Malleable 90-Deg Elbows, Tees, Crosses, and 45-Deg Elbows (Straight Sizes, 150 Lb)



Nominal Pipe Size	Center to End, Elbows, Tees, and Crosses	Center to End, 45-Deg Elbows	Length of Thread, Min	Width of Band, Min	Inside Diameter of Fitting: Min	Inside Diameter of Fitting: Max	Metal Thickness	Outside Diameter of Band, Min
	A	C	B	E	F		G	H
⅛	0.69	....	0.25	0.200	0.405	0.435	0.090	0.693
¼	0.81	0.73	0.32	0.215	0.540	0.584	0.095	0.844
⅜	0.95	0.80	0.36	0.230	0.675	0.719	0.100	1.015
½	1.12	0.88	0.43	0.249	0.840	0.897	0.105	1.197
¾	1.31	0.98	0.50	0.273	1.050	1.107	0.120	1.458
1	1.50	1.12	0.58	0.302	1.315	1.385	0.134	1.771
1¼	1.75	1.29	0.67	0.341	1.660	1.730	0.145	2.153
1½	1.94	1.43	0.70	0.368	1.900	1.970	0.155	2.427
2	2.25	1.68	0.75	0.422	2.375	2.445	0.173	2.963
2½	2.70	1.95	0.92	0.478	2.875	2.975	0.210	3.589
3	3.08	2.17	0.98	0.548	3.500	3.600	0.231	4.285
3½	3.42	2.39	1.03	0.604	4.000	4.100	0.248	4.843
4	3.79	2.61	1.08	0.661	4.500	4.600	0.265	5.401
5	4.50	3.05	1.18	0.780	5.563	5.663	0.300	6.583
6	5.13	3.46	1.28	0.900	6.625	6.725	0.336	7.767

From *American Standard for Malleable-Iron Screwed Fittings*, 150 Lb. *ASA* B16.3-1951. All dimensions given in inches.

Table 16 . . . . Dimensions of Brass or Bronze Screwed 90-Deg Elbows, Tees, Crosses, 45-Deg Elbows and Couplings (Straight Sizes 125 Lb)



Nominal Pipe Size	Center to End, Elbows, Tees, Crosses	Length of Thread, Min	Center to End 45-Deg Elbows	Diameter Wrought Coupling	Band Length, Min	Inside Diameter of Cast Fitting		Metal Thickness, Min	Band Diameter, Min	End to End Straight Coupling	
						Min	Max			Cast	Wrought
	A	B	C	D	E	F		G	H	W	
⅛	0.54	0.25	0.42	9/16	0.14	0.41	0.44	0.08	0.67	0.80	0.83
¼	0.71	0.32	0.56	11/16	0.16	0.54	0.58	0.08	0.81	0.97	1.03
⅜	0.82	0.36	0.63	23/32	0.17	0.68	0.72	0.09	1.00	1.05	1.11
½	1.01	0.43	0.78	1 1/16	0.19	0.84	0.90	0.09	1.17	1.29	1.36
¾	1.18	0.50	0.89	1 5/16	0.23	1.05	1.11	0.10	1.42	1.43	1.50
1	1.43	0.58	1.06	....	0.27	1.32	1.39	0.11	1.72	1.68	....
1¼	1.69	0.67	1.22	....	0.31	1.66	1.73	0.12	2.10	1.86	....
1½	1.84	0.70	1.30	....	0.34	1.90	1.97	0.13	2.38	1.92	....
2	2.12	0.75	1.45	....	0.41	2.38	2.45	0.15	2.92	2.20	....
2½	2.70	0.92	1.95	....	0.48	2.88	2.98	0.17	3.49	2.88	....
3	3.08	0.98	2.17	....	0.55	3.50	3.60	0.19	4.20	3.18	....
4	3.79	1.08	2.61	....	0.66	4.50	4.60	0.22	5.31	3.69	....

From *American Standard for Brass or Bronze Screwed Fittings*, 125 Lb. *ASA* B16.15-1947. All dimensions given in inches.

Equation 4 can be used for bends with two fittings, regular U-bend, and offset U-bend, as shown in Fig. 3. It gives the length of pipe, L, that must be used in the U-bend to take up the deformation, Δ. When welding fittings are used, the radii should be six times the outside diameter of the pipe. One other restriction is that the pipe should not be heavier than extra strong.

Manufacturers of pipe, or pipe fittings, often publish excellent treatises on the simplification of expansion problems. It is suggested that the engineer obtain such publications and also a text or handbook[a] on the subject before attempting to solve a problem on flexibility.

HANGERS AND SUPPORTS

In the preceding section, Equation 3 was developed to give the force of total constraint. This equation is appropriate regardless of the sign of Δt, positive or negative. It did not, however, allow for any bending moment but was for straight compression or tension.

If the compressive force is great enough, or if the line is long enough, a slight eccentricity of the load application will cause a *buckling* of the line. This buckling is identical with the action of a column under excessive compression. In fact, a pipe line in compression is identical to a column. Column criteria can therefore be used for selecting the action of supports to resist lateral thrust.

A common equation for determining safe column loads is Euler's formula:

$$S = \pi^2 E \left(\frac{k}{l}\right)^2 \qquad \text{(for cast iron see Equation 8)} \qquad (5)$$

where

S = allowable column load, pounds per square inch.
k = radius of gyration, inches.
l = column length, inches.

$$k^2 = \frac{I}{A} = \frac{(OD)^2 + (ID)^2}{16}$$

Equation 5 is sometimes modified by a correction factor C to the following:

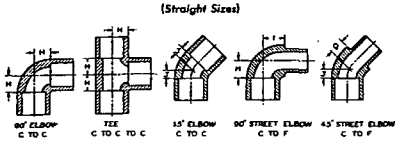
$$S = C\pi^2 E \left(\frac{k}{l}\right) \qquad (6)$$

where

C = correction factor depending on end loading in accordance with the following table:

End Conditions of Column		Value of C
One End	Other End	
Fixed	Fixed	4.00
Fixed	Free	0.25
Fixed	Pin	2.25
Pin	Pin	1.00

Table 17 . . . . Dimensions of Cast-Brass Solder-Joint Elbows, Tees, and 45-Deg Elbows
(Straight Sizes)



Standard Water Tube Size	Laying Length* Tee and Elbow	Center to External Shoulder 90-Deg Street Elbow	Laying Length 45-Deg Elbow	Center to External Shoulder 45-Deg Street Elbow
	H	I	J	Q
¼	¼	⅜	....	....
⅜	5/16	7/16	5/16	5/16
½	7/16	9/16	5/16	5/16
¾	9/16	11/16	¼	⅜
1	¾	⅞	5/16	7/16
1¼	⅞	1	7/16	9/16
1½	1	1⅛	½	⅝
2	1¼	1⅜	9/16	¾
2½	1½	1⅝	⅝	....
3	1¾	1⅞	¾	....
3½	2	....	⅞	....
4	2¼	2⅜	1 5/16	....
5	3⅛	....	1 11/16	....
6	3⅝	....	1⅝	....
8	4⅝	....	2⅛	....

From *American Standard for Cast-Brass Solder-Joint Fittings*, *ASA* B16.18-1950. All dimensions given in inches.
* Not standardized for wrought fittings. Consult manufacturer for dimensions.

From Equation 6,

$$l = \pi k \sqrt{\frac{CE}{S}} \qquad (7)$$

This equation is not used for column construction but is sufficiently accurate for the work of a heating engineer.

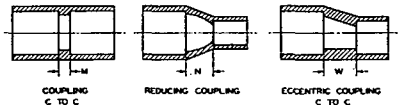
The equation is modified for cast iron because of its poor tensile strength. It is generally acceptable to use one-half the value of S from Equation 5 for cast iron, so the safe column load S_1 for cast iron is

$$S_1 = \frac{\pi^2 E}{2} \left(\frac{k}{l}\right)^2 \qquad \text{for cast iron.} \qquad (8)$$

It is permissible to allow the l/k ratio for steel to go as high as 120, but for cast iron the desirable maximum is taken as 70. It is not to be understood that a fracture will occur at l/k ratios in excess of those given, rather it is to be understood that lateral support should be given for those ratios and for those magnitudes of stress. If an expansion joint is used, this will, of course, relieve the compressive stress.

Example 2: For Example 1 given in the preceding section, the constraining force for a 1-in. pipe was 9630 lb. What is a required spacing for lateral support for this pipe?

Table 18 . . . . Dimensions of Cast-Brass Solder-Joint Couplings
(Straight, Reducing, and Eccentric Sizes)



STRAIGHT		REDUCING		ECCENTRIC	
Standard Water Tube Size	Laying Length* M	Standard Water Tube Size	Laying Length* N	Standard Water Tube Size	Laying Length* W
⅛	1/16	¾ x ½	5/16	¾ x ½	⅝
⅜	1/16	1 x ¾	⅜	1 x ¾	1 1/16
½	⅛	1¼ x 1	⅜	1¼ x 1	¾
¾	⅛	1¼ x ¾	⅜	1¼ x ¾	¾
1	⅛	1½ x 1¼	⅜	1½ x 1¼	1 1/16
1¼	⅛	1½ x 1	⅜	1½ x 1	1 1/16
1½	⅛	1½ x ¾	7/16	2 x 1½	1⅛
2	3/16	2 x 1½	½	2 x 1¾	1 5/16
2½	3/16	2 x 1¼	½	2½ x 2	1 3/16
3	3/16	2 x 1	½	3 x 2½	1¼
3½	¼	2 x ¾	½	3 x 2	1 5/16
4	¼	2½ x 2	9/16	4 x 3	2
5	¼	2½ x 1½	9/16	....	....
6	¼	2½ x 1¼	⅝	....	....
8	5/16	2½ x 1	⅝	....	....
....	....	3 x 2½	⅝	....	....
....	....	3 x 2	⅝	....	....
....	....	4 x 3	1 1/16	....	....
....	....	4 x 2½	1⅛	....	....
....	....	4 x 2	1 5/16	....	....
....	....	6 x 4	1 5/16	....	....
....	....	8 x 6	1⅜	....	....

From *American Standard for Cast-Brass Solder-Joint Fittings*, *ASA* B16.18-1950. All dimensions given in inches.
* Not standardized for wrought fittings. Consult manufacturer for dimensions.

Answer: Using Euler's formula, Equation 5 gives

$$S = \pi^2 E \left(\frac{k}{l}\right)^2$$

so

$$l = \pi k \sqrt{E/S}$$

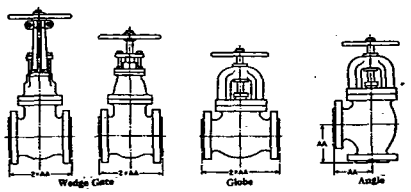
For a 1 in. Schedule 40 steel pipe

k = 0.4205 in.
E = 30,000,000 psi.
S = 9630 ÷ 0.494 = 19500 psi.
l = 51.8 in.

Therefore, lateral supports on 4-ft centers are adequate.

The other type of load requiring some support is the bending stress due to the weight of the pipe, its insulation, and of

Table 19 .... Dimensions of Flanged Cast-Iron and Steel Valves



	Nominal Pipe Size	Contact Surface to Contact Surface Dimensions, (2 × AA)									
		Cast Iron				Steel					
		125	175[a,b]	250[a]	800[a] Hydraulic	150[a]	300[a]	400[a]	600[a]	900[a]	1500[a]
Wedge Gate Valves	1	....	....	....	....	....	....	*8½	8½	*10	10
	1¼	....	....	....	....	....	....	*9	9	*11	11
	1½	....	....	....	....	....	7½	*9½	9½	*12	12
	2	7	7½	8½	11½	7	8½	*11½	11½	*14½	14½
	2½	7½	8	9½	13	7½	9½	*13	13	*16½	16½
	3	8	9½	11½	14	8	11½	*14	14	15	18½
	3½	8½	10	11⅞	....	8½	11⅞	....	....	....	....
	4	9	10½	12	17	9	12	16	17	18	21½
	5	10	11½	15	....	10	15	18	20	22	26½
	6	10½	13	15⅞	22	10½	15⅞	19½	22	24	27¾
	8	11½	14½	16½	26	11½	16½	23½	26	29	32¾
	10	13	16¾	18	31	13	18	26½	31	33	39
	12	14	17½	19¾	33	14	19¾	30	33	38	44½
	14 OD	15	....	22½	....	15	30	32½	35	40½	....
	16 OD	16	....	24	....	16	33	35½	39	44½	....
	18 OD	17	....	26	....	17	36	38½	43	48	....
	20 OD	18	....	28	....	18	39	41½	47	52	....
	24 OD	20	....	31	....	20	45	48½	55	61	....
Globe & Angle Valves	½	....	....	....	....	....	....	....	....	....	....
	¾	....	....	....	....	....	....	7½	7½	....	....
	1	....	....	....	....	....	....	8½	8½	10	10
	1¼	....	....	....	....	....	....	9	9	11	11
	1½	....	....	....	....	....	....	9½	9½	12	12
	2	8	....	10½	....	8	10½	11½	11½	14½	14½
	2½	8½	....	11½	....	8½	11½	13	13	16½	16½
	3	9½	....	12½	....	9½	12½	14	14	15	18½
	3½	10½	....	13¼	....	10½	13¼	....	....	....	....
	4	11½	....	14	....	11½	14	16	17	18	21½
	5	13	....	15¾	....	14	15¾	18	20	22	26½
	6	14	....	17½	....	16	17½	19½	22	24	27¾
	8	19½	....	21	....	19½	22	23½	26	29	32¾

Notes for Table 19

From American Standard for Face-to-Face Dimensions of Ferrous Flanged and Welding End Valves, ASA B16.10-1939.

All dimensions given in inches.

Where dimensions are not given, the sizes either are not made or there is insufficient demand to warrant the expense of unification.

Female and groove joint facings have bottom of groove in same plane as *flange edge*, and center to contact surface dimensions for these facings are reduced by the amount of the raised face.

[a] These are pressure designations which refer to the primary service ratings in lb per sq in. of the connecting end flanges.

[b] The connecting end flanges of 175 lb valves are the same as those on 250 lb valves.

[c] The contact surface to contact surface dimensions and connecting end flanges for 600 lb valves 3 in. size and smaller, are identical with those of 800 lb valves; and the contact surface to contact surface dimensions and connecting end flanges for 900 lb valves 2½ in. size and smaller, are identical with those of 1500 lb valves.

Table 20 .... Thermal Stresses Due to Total Constraint

Temperature[a] Change, F Deg	Thermal Stresses, Psi					
	Steel	Wrought Iron	Cast Iron	Brass or Bronze	Copper	Aluminum
20	3900	3800	1428	2760	2980	2700
40	7800	7600	2856	5520	5960	5400
60	11700	11400	4284	8280	8940	8100
80	15600	15200	5712	11040	11920	10800
100	19500	19000	7140	13800	14900	13500

[a] For temperatures between 32 and 400 F.

Table 21 .... Physical Properties for Determining Thermal Stress

(Temperatures between 32 and 400 F)

	Steel	Wrought Iron	Cast Iron	Brass or Bronze	Copper	Aluminum
Coefficient of Linear Expansion, $a \times 10^6$	6.5	6.8	5.95	9.85	9.3	13.5
Modulus of Elasticity, E, psi $\times 10^{-6}$	30	28	12	14	16	10
Product of aE, psi	195	190	71.4	138	149	135

Example: The coefficient of expansion for steel = 0.0000065 in./in. and the modulus of elasticity = 30,000,000 psi. Therefore aE = 195 psi.

Table 22 .... Equations for Bending Stress and Deflection between Supports

Type of Support	Bending Stress, Psi S	Deflection, Inches Y
Single span (free ends)	$S = \frac{0.75\,WL^3D}{I}$	$Y = \frac{22.5\,WL^4}{EI}$
Continuous line	$S = \frac{0.5\,WL^3D}{I}$	$Y = \frac{4.5\,WL^4}{EI}$

W = Total weight (pipe, fluid, etc.) pounds per foot.
D = OD of pipe, inches.
L = Length of span, feet.
E = Modulus of elasticity, pounds per square inch.
I = Moment of inertia, (inches)^4.

$I = \frac{\pi}{64}\left[(\text{OD})^4 - (\text{ID})^4\right]$, OD, and ID in inches.

Table 23 .... Application of Pipe, Tube, and Fittings for Heating and Air Conditioning

Application	Materials Commonly Used
Chilled water	Type K or L hard copper or black steel, Schedule 40
Cooling tower	Type K or L hard copper, wrought iron, or black steel, Schedule 40
Underground water 2-in. and smaller 2½-in. and larger	Type K, soft copper or galvanized steel, Schedule 40 (for corrosive soils, a coated steel pipe should be used) Type K, soft copper, cast iron, Class 150 or galvanized steel, Schedule 40 (for corrosive soils, a coated steel pipe should be used)
City water—inside building	Type K or L, hard copper or galvanized steel, Schedule 40
Underground rain water conductors Small installations Under paved areas Large installations 8-in. and smaller 10 to 24-in. 30-in. and larger 5 feet outside bldg. and under paving	 Vitrified clay or cast-iron soil Cast-iron soil Cast-iron soil XH or vitrified clay Vitrified clay or concrete Concrete Cast-iron soil XH
Interior rain water conductors Small installations Large installations	--- Black or galvanized wrought iron or steel Schedule 40 Cast-iron soil XH, black or galvanized wrought iron or steel, Schedule 40
Panel heating[a]	Type K or L soft copper. Black steel or wrought iron, Schedule 40
Hot water heating[b]	Type K, hard copper or black steel, Schedule 40
Refrigerant[c]	Type K or L hard copper or black steel, Schedule 40
Steam heating Supply 125 psi and less 125 to 250 psi Return pipe All sizes	 Black steel, Schedule 40, standard fittings Black steel, Schedule 40. For fittings see Table 8 Black wrought iron or steel, Schedule 40. Standard fittings
Receiver vent All sizes	Galvanized wrought iron or steel, Schedule 40. Standard fittings
Sanitary[d] Small installations Waste: 1½-in. and smaller 2-in. and larger Vents: 2½-in. and smaller 3-in. and larger Large installations Waste: 2-in. and larger Vents: 2-in. and larger	 Galvanized wrought iron or steel, Schedule 40 Cast iron Galvanized wrought iron or steel, Schedule 40 Cast iron Cast iron Cast iron
Gas (all kinds) All sizes	Black steel, Schedule 40. Malleable iron fittings

[a] Panel heating copper fittings should be soldered with 95-5 solder. Steel fittings should be welded.
[b] All underground feed and return lines buried in slab, should be equipped with fittings as outlined for panel heating.
[c] Refrigerant fittings on copper pipe must be wrought copper. All welded fittings should be long turn.
[d] House sewer to main should be cast iron XH.

the fluid being carried. There is also the bending moment that may be caused by the flexure of the pipe. For this chapter, however, the flexure loading will not be treated for the reason of complexity, as previously stated for the problem of expansion.

Since hangers should always be designed to take the bending moment (including buckling) from the pipe, it is necessary to design the hangers or supports for the maximum bending stress. Even with perfect support spacing there will be some sagging; therefore, the supports should be on a gradient if it is necessary to prevent pockets. Such a condition arises for condensate piping.

Typical bending and deflection equations are given in Table 22 in order that supports can be properly sized, and if necessary, pitched. These equations are typical beam equations. If more complex loadings exist, proper beam equations can be applied. Such equations are found in standard texts on mechanics of materials.

By application of the equations given in this section and the section on Expansion and Flexibility, it is possible to determine the proper location and type of support for straight lines. The types are anchors and hangers.

Anchors are used to guide the expansion (sliding anchors) or to fix the pipe. They are needed to take the stress out of the line and translate the stresses to the foundations. In that way all stress can be taken out of the line before it can be transferred to such equipment as boilers or pumps. By means of solid anchors, the anchors can be used to divide the piping into isolated sections that cannot transmit the stresses to other sections.

Hangers are used to support the piping and to take out the bending stresses. They are of the spring, roller, or rod type. Such types generally allow the free movement of the pipe and yet maintain full support.

The *Code for Pressure Piping*, ASA B31.1, has strict requirements for the materials used in hangers and supports. The structural properties have a safety factor of five, and consequently there is little chance of failure due to a sudden change of load.

There is a type of support for almost every conceivable

requirement. Complete information can best be obtained from a hanger manufacturer.

APPLICATION OF PIPE, TUBE, AND FITTINGS

In the application of pipe, tube, and fittings it is necessary to comply with regulations contained in local codes and ordinances. Table 23 is offered as a guide to current practice in heating and air conditioning. It is based on practice in a number of consulting engineers' offices and information extracted from Reference 7. It should be borne in mind that selection of material to be used will also be influenced by the familiarity of local artisans with methods of using different materials.

REFERENCES

[1] *American Standard Code for Pressure Piping* (American Standards Association, ASA B-31.1, 1942).

[2] *API Specification 5L for Line Pipe* (American Petroleum Institute).

[3] *Standard Manual on Pipe Welding* (Heating, Piping and Air Conditioning Contractors National Association, 1951, 2nd ed.). *Welding Handbook* (American Welding Society, 1942).

[4] *ASME Power Boiler Code* (American Society of Mechanical Engineers).

[5] *Marine Engineering Regulations of the Coast Guard* (American Bureau of Shipping). Welding (*General Specifications for Inspection of Material*, Appendix VII, U. S. Navy). General—for vessels of the U. S. Navy (*Specifications for Welding*, Appendix 5, Part 1, Bureau of Ships, April 1940).

[6] Sabin Crocker: *Piping Handbook* (McGraw-Hill Book Co., New York). E. A. Wert, S. Smith, and E. T. Cope: *A Manual for the Design of Piping for Flexibility by the Use of Graphs* (The Detroit Edison Company).

[7] C. S. Beard: Control valves and positioners (*Industry and Power*, March 1953, p. 67).

CHAPTER 32

PIPE AND INDUSTRIAL INSULATION

Industrial Insulation: Uses, Properties, Temperature Range of Use, Temperature Limitations, Forms; Bare Surface Heat Loss; Conductivities; Heat Flow Calculations; Thickness for Application; Economical Thickness; Low Temperature Insulation; Insulation to Prevent Freezing; Application; Underground Insulations

THE PAST 50 years have seen an enormous increase in the use of thermal insulation by industry. Where once it was not uncommon to see hot water and steam lines, and even boiler surfaces, uninsulated, it is now rare indeed to see any surface operating more than a little above or below ambient temperature without insulation. Higher costs of fuel, increased efficiency of industrial operations, and the necessity for close control of the temperature of industrial processes are among the factors that have caused this greatly increased usage of thermal insulation.

USES OF INDUSTRIAL INSULATION

Thermal insulation serves five principal purposes: (1) to conserve heat or some other form of energy, for example, electrical energy to operate a refrigeration compressor; (2) to increase the comfort of living spaces; (3) to facilitate control of the temperature of a process; (4) to reduce the temperature of the shell of a pressure vessel; and (5) to control the external temperature of the insulated space in order to avoid danger to personnel, to protect surrounding structural members from damage by high temperature, to reduce the temperature of working spaces, and, in case of equipment operating below ambient temperatures, to prevent sweating or icing at the warmer surface. Depending on the application, one or another purpose may govern the choice of insulation; usually two or more are concurrently of importance.

PROPERTIES OF INSULATIONS

Low thermal conductivity is the property that distinguishes a thermal insulation. Thermal conductivity is defined as *the rate of heat flow through unit area of a homogeneous substance under the influence of unit temperature gradient in the direction perpendicular to the area.* Many units have been used to express thermal conductivity but the most common engineering unit is Btu (hour) (square foot) (Fahrenheit degree per inch of thickness). The usual symbol for thermal conductivity is k.

Reflective insulations, such as aluminum and steel coated with a reflecting coating, depend not upon low thermal conductivity for their resistance to heat flow, but upon the low thermal emissivity of their surfaces and subdivision of the air space. To be effective in retarding heat flow by radiation, the reflective surface must face an air space.

Thermal conductivity values of the most commonly used industrial and pipe insulations are shown in Table 1. These values have been selected as typical and useful for engineering calculations, and may or may not apply to a particular material or application. For accurate values for a particular insulation the user should consult the manufacturer or obtain the results of unbiased tests.

Other properties of thermal insulations that are of more or less importance, depending upon the use, are strength, hardness, density, compressibility, specific heat, resistance to high or low temperature, and thermal coefficient of expansion. The final choice of an insulating material for a given purpose usually involves a compromise with regard to several desirable properties. For example, an insulating firebrick must have considerable strength and must be resistant to the temperature to which it is to be exposed. The desirable properties of low conductivity and light weight must be sacrificed to some extent to achieve these necessary properties of strength and temperature resistance.

TEMPERATURE RANGE OF USE OF INDUSTRIAL INSULATION

Since thermal insulation reduces the exchange of heat energy between an insulated surface and its surroundings, it can be useful at any temperature from the lowest to the highest. Actually insulation is used in industrial processes operating at temperatures from −400 F to +3000 F and higher.

At the low end of the scale are processes involving liquefied gases, usually in vessels or piping, at temperatures from the lowest up to about −40 F. From −40 to normal atmospheric temperatures large amounts of low temperature insulation are used for enclosing freezing tunnels, cold storage rooms, and a great variety of food and industrial processes requiring low temperatures.

At normal and near-normal temperatures insulation in building structures is required for economical operation of heating and air-conditioning systems. Insulation for heating and cooling equipment and for the distribution ducts, piping, etc., is, however, usually considered as industrial. The range of temperatures is from that of chilled water or about 32 F up to steam at 100–125 psig (about 350 F).

Drying and baking ovens, small and large boilers, distillation equipment in oil refineries, and high-pressure and superheated steam piping call for insulation suitable up to 1200 F. Petroleum cracking units, heat treating furnaces, and ceramic kilns may reach 2200–2400 F. Still higher temperatures with surfaces to be insulated up to 3000 F are found on melting furnaces for glass, iron, steel, and metals of all kinds as well as furnace enclosures for combustion spaces for many kinds of heating units.

Above 3000 F the variety of materials physically suited for walls and structural parts decreases rapidly with increasing

Table 1 . . . . Thermal Conductivity (k) of Industrial and Pipe Insulations

(For Mean Temperatures Indicated)

Expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference per in.)

Form	Material (Composition)	Accepted Max. Temp. for Use,[a] F	Density Lb/Cu Ft	Typical Conductivity k at Mean Temp. Fahr. 40	70	100	200	300	500	700	900
BLANKETS	MINERAL WOOL (*Rock, Slag, or Glass*)										
	Metal Reinforced	1200	6–15			0.29	0.35	0.42	0.56		
	Felt-Flexible Type	450	0.5–3	0.23	0.25	0.26	0.34	0.45			
	Felt-Semi-Rigid Type	450	2–8	0.24	0.25	0.27	0.35	0.44			
	VEGETABLE & ANIMAL FIBER										
	Hair Felt or Hair Felt plus Jute	180	10	0.27	0.28	0.30					
BLOCKS AND BOARDS	ASBESTOS										
	Molded Amosite & Binder	1200	18			0.37	0.40	0.47	0.56	0.66	0.76
	Laminated Asbestos Paper	700	30			0.40	0.45	0.50	0.60		
	Corrugated & Laminated Asbestos Paper										
	4 ply	300	11–13		0.54	0.57	0.68	0.80			
	6 ply	300	15–17		0.49	0.51	0.59	0.69			
	8 ply	300	18–20		0.47	0.49	0.57	0.65			
	CALCIUM SILICATE	1200	11			0.33	0.38	0.43	0.53	0.64	0.75
	CELLULAR GLASS	800	9	0.37	0.39	0.41	0.48	0.55			
	CORKBOARD (*Without Added Binder*)	200	6.5–8	0.26	0.27	0.28					
	DIATOMACEOUS SILICA	1500	22						0.60	0.64	0.68
		1900	25						0.70	0.75	0.80
	85% MAGNESIA										
	Mag. Carb. & Asbestos	600	11–14			0.35	0.38	0.42	0.46		
	MINERAL WOOL (*Rock, Slag or Glass*)										
	Low Temp. (Asphalt or Resin Bonded)	200	6–18	0.28	0.29	0.30					
	High Temp. (Resin Bonded)	600	6–10			0.28	0.35	0.43			
	(With Inorganic Binder)	1600	16–24			0.34	0.39	0.44	0.54	0.64	
	PLASTICS (Foamed)	175	1.6	0.26	0.28	0.30					
	RUBBER (Foamed)	150	5	0.23	0.24	0.25					
PIPE INSULATION	ASBESTOS										
	Molded Amosite & Binder	1200	16			0.33	0.38	0.43	0.53		
	Laminated Asbestos Paper	700	30			0.40	0.45	0.50	0.60		
	Corrugated & Laminated Asbestos Paper										
	4 Ply per in.	300	11–13		0.54	0.57	0.62	0.80			
	6 Ply per in.	300	15–17		0.49	0.51	0.59	0.69			
	8 Ply per in.	300	18–20		0.47	0.49	0.57	0.65			
	CALCIUM SILICATE										
	Calc. Sil. & Asbestos	1200	11			0.36	0.40	0.44	0.55		
	CELLULAR GLASS	800	9	0.37	0.39	0.41	0.48	0.55			
	CORK (*Without Added Binder*)	200	7–10	0.27	0.28	0.29	0.30				
	DIATOMACEOUS SILICA	1500	22						0.64	0.66	0.71
		1900	25						0.70	0.75	0.80
	85% MAGNESIA										
	Mag. Carb. & Asbestos	600	11–14			0.39	0.42	0.45	0.51		
	MINERAL WOOL (*Rock, Slag or Glass*)										
	Low Temp. (Asphalt or Resin Bonded)	200	15	0.28	0.30	0.33	0.39				
	Low Temp. (Fine Fiber Resin Bonded)	450	3	0.22	0.23	0.24	0.27	0.31			
	High Temp. Blanket-Type (Metal Reinforced)	1200	6–15			0.29	0.36	0.42	0.56		
	PLASTICS (Foamed)	175	1.6	0.26	0.28	0.31					
	RUBBER (Foamed)	150	5	0.23	0.24	0.25					
	VEGETABLE & ANIMAL FIBER										
	Wool Felt	180	20	0.29	0.31	0.33					
	Hair Felt or Hair Felt plus Jute	180	10	0.27	0.28	0.30					
INSULATING CEMENT	85% MAGNESIA										
	Mag. Carb. & Asbestos	600	18			0.46	0.52	0.58			
	MINERAL WOOL (*Rock, Slag or Glass*)										
	With Colloidal Clay Binder	1800	24–30			0.49		0.61	0.73	0.83	

[a] These temperatures are generally accepted as maximum. When operating temperature approaches these limits the Manufacturer's recommendations should be followed.

temperature. Insulation for conservation of heat energy must, therefore, often be applied so as to permit enough heat to escape to retain mechanical strength.

TEMPERATURE LIMITATIONS OF INSULATIONS

Probably the greatest volume of low temperature insulation is used for temperatures from minus 100 F to atmospheric. The effect of temperature alone on physical integrity is not ordinarily very important for most types in the low temperature field. There are more definite limitations for materials in the high temperature field where decomposition, excessive linear shrinkage, softening, or some other effect of temperature alone will put a maximum limit on the temperature for which a material is suited. As extremes in temperature are approached, both high and low, the selection of a material for a specific service becomes more critical and must be based on experience and factual performance data.

FORMS OF INSULATION

The physical form of industrial insulations varies from powders and loose fills which might be called *formless*, through materials with varying degrees of compressibility and flexibility, to completely rigid shapes often referred to as preformed insulation. After deciding that a material has the necessary insulating value, physical properties, and temperature resistance, the choice of the form desired will usually be governed by considerations of material and application cost. The application method and cost in turn will be greatly affected by the nature of the surface to be covered.

Loose fills such as powders and granules must be supported in place by retaining walls, coverings, etc. for application on vertical surfaces and on the underside of horizontal surfaces. Insulating cement is a loose material which when mixed with water to obtain plasticity and adhesion, may be trowelled on a surface and dried in place to serve as insulation. Both loose-fill and insulating cement are especially suited for covering uneven and irregular surfaces.

Flexible and nonrigid insulations are generally grouped as blanket insulation, and are available in many types and varieties, both organic and inorganic, with and without binders, and with reinforcing coverings on one or both sides. Only inherently fibrous materials are generally suited for blanket insulation. Such materials are wood, cotton, and hair in the organic field, and asbestos and mineral wool in the inorganic field. Coverings and facings such as paper, aluminum, asbestos sheet, wire mesh, metal lath, may be used as reinforcing, vapor barriers, reflective surfaces, or surface finishes in general. Thicknesses and shapes may be of any dimension that can be handled although standard sizes are generally used. Some blanket insulations are prepared in roll form for economical shipping.

Rigid materials are preformed during manufacture to standard lengths, widths, and thicknesses of a size that can be handled readily. In the case of insulation for pipes and cylindrical surfaces the radius of curvature is varied to suit all standard sizes of pipe and tubing. Some materials are offered in shapes curved to fit greater radii up to several feet or more. Material in rectangular dimensions is usually called block, board, or sheets, the name varying with custom for particular uses. Most common sizes of block insulation are 36 in. long, 6 in. or 12 in. wide, and in thicknesses ranging from 1 to 4 in. *Boards* in the building insulation field range up to 12 ft long by 4 ft wide with the thickness usually ½ to 1 in. but ranging up to 1½ in. Smaller *boards*, or sheets, are used on air-conditioning ducts, equipment, and in cold storage.

BARE SURFACE HEAT LOSSES—FLAT SURFACES AND PIPE

Heat losses from or to uninsulated surfaces of pipes, ducts, vessels, or furnaces may be of considerable magnitude if the temperature of the surrounding medium differs appreciably from that of the surface. Losses are increased by motion of the surrounding air or by contact of the hot or cold surface with bodies of high conductivity. Good design, therefore, must include consideration of such heat losses and provision for adequate insulation wherever needed.

The basic principles of heat loss from surfaces are discussed in Chapter 5. In that chapter radiation and convection are treated separately.

Heat losses from horizontal bare steel pipes, based on tests at Mellon Institute and calculated from the fundamental radiation and convection equations (Chapter 5), are given in Table 2. This table also gives the heat losses or surface conductances for flat vertical and horizontal surfaces for surface temperatures up to 1080 F with the surrounding air at 80 F. The surface per linear foot of pipe is given in the second column of Table 2.

Heat losses from tarnished copper pipe and tube are given in Table 3. The surface per linear foot of tube is given in Table 4. Table 3, Section A, of Chapter 9 also gives the surface conductances for flat surfaces of different emissivities and orientations in contact with still air. Table 5 gives the area in square feet of flanges and fittings for various standard pipe sizes. These tables can be used to advantage in estimating the amount of insulation required.

Examples 1 and *2* show how the annual heat loss from uncovered pipe and its dollar value may be computed from the data in Table 2.

Example 1: Compute the total annual heat loss from 165 ft of 2 in. bare pipe in service 4000 hr per year. The pipe is carrying steam at 10 psi pressure and is exposed to an average air temperature of 80 F.

Solution: The pipe temperature is taken as the steam temperature, which is 239.4 F, obtained by interpolation from Steam Tables. The temperature difference between the pipe and air = 239.4 − 80 = 159.4 F. By interpolation in Table 2 between temperature differences of 150 and 200 F, the heat loss from a 2-in. pipe at a temperature difference of 159.4 F is found to be 2.615 Btu per (hr) (sq ft) (F deg). The total annual heat loss from the entire line = 2.615 × 159.4 × 0.622 (linear ft factor) × 165 (linear ft) × 4000 (hr) = 171,100 Mb. (Mb = 1000 Btu.)

Example 2: Coal costing $11.50 per ton and having a calorific value of 13,000 Btu per pound is being burned in the furnace supplying steam to the pipe line given in the previous example. If the system is operating at an overall efficiency of 55 percent, determine the monetary value of the annual heat loss from the line.

Solution: The cost of heat per 1000 Mb supplied to the system = 1,000,000 × 11.5 (dollars) ÷ [13,000 (Btu) × 2000 (lb) × 0.55 (efficiency)] = $0.804. The total cost of heat lost per year = 0.804 × 171.1 (thousand Mb) = $137.56.

CONDUCTIVITY OF INDUSTRIAL INSULATIONS

The conductivities of various materials used for insulating steam and hot water systems are given in Table 1. They are given as functions of the mean temperatures or the arithmetic mean of the inner and outer surface temperatures of the insulations.

HEAT FLOW CALCULATIONS

The heat losses through 1, 1½, and 2-in. thick pipe insulation on various size pipes for various temperature differences

Table 2 .... Heat Losses from Horizontal Bare Steel Pipes and Flat Surfaces

In Btu per (Sq Ft of Pipe Surface) (Hour) (F Deg Temperature Difference Between Pipe and Air)

Pipe Size Inches	Linear Foot Factor[a]	Temperature Difference F Deg Between Pipe Surface and Surrounding Air. Air at 80 F.																		
		50	100	150	200	250	300	350	400	450	500	550	600	650	700	750	800	850	900	950	1000
¼	0.220	2.12	2.48	2.80	3.10	3.42	3.74	4.07	4.47	4.86	5.28	5.72	6.19	6.69	7.22	7.79	8.39	9.03	9.70	10.42	11.18
¾	0.275	2.08	2.43	2.74	3.04	3.35	3.67	4.00	4.40	4.79	5.21	5.65	6.12	6.61	7.15	7.71	8.31	8.95	9.62	10.34	11.09
1	0.344	2.04	2.38	2.69	2.99	3.30	3.61	3.94	4.33	4.72	5.14	5.58	6.05	6.54	7.07	7.64	8.23	8.87	9.55	10.26	11.02
1¼	0.435	2.00	2.34	2.64	2.93	3.24	3.55	3.88	4.27	4.66	5.07	5.51	5.97	6.47	7.00	7.56	8.16	8.79	9.47	10.18	10.94
1½	0.497	1.98	2.31	2.61	2.90	3.20	3.52	3.84	4.23	4.62	5.03	5.47	5.93	6.43	6.96	7.52	8.12	8.75	9.43	10.14	10.89
2	0.622	1.95	2.27	2.56	2.85	3.15	3.46	3.78	4.17	4.56	4.97	5.41	5.87	6.37	6.89	7.45	8.05	8.68	9.36	10.07	10.82
2½	0.753	1.92	2.23	2.52	2.81	3.11	3.42	3.74	4.12	4.51	4.92	5.36	5.82	6.31	6.84	7.40	7.99	8.63	9.30	10.01	10.77
3	0.916	1.89	2.20	2.49	2.77	3.07	3.37	3.69	4.08	4.46	4.87	5.31	5.77	6.26	6.79	7.35	7.94	8.57	9.25	9.96	10.71
3½	1.047	1.87	2.18	2.46	2.74	3.04	3.34	3.66	4.05	4.43	4.84	5.27	5.73	6.23	6.75	7.31	7.91	8.54	9.21	9.92	10.67
4	1.178	1.85	2.16	2.44	2.72	3.01	3.32	3.64	4.02	4.40	4.81	5.25	5.71	6.20	6.72	7.28	7.87	8.51	9.18	9.89	10.64
4½	1.309	1.84	2.14	2.42	2.70	2.99	3.30	3.61	4.00	4.38	4.79	5.22	5.68	6.17	6.69	7.25	7.85	8.48	9.15	9.86	10.61
5	1.456	1.83	2.13	2.40	2.68	2.97	3.28	3.59	3.97	4.35	4.76	5.20	5.65	6.15	6.68	7.23	7.82	8.45	9.12	9.83	10.58
6	1.734	1.80	2.10	2.37	2.65	2.94	3.24	3.55	3.94	4.32	4.72	5.16	5.61	6.10	6.63	7.19	7.78	8.41	9.08	9.79	10.54
7	1.996	1.79	2.08	2.35	2.63	2.91	3.21	3.53	3.91	4.29	4.69	5.13	5.58	6.07	6.60	7.15	7.75	8.38	9.05	9.76	10.51
8	2.258	1.77	2.06	2.33	2.60	2.89	3.19	3.50	3.88	4.26	4.67	5.10	5.56	6.05	6.57	7.12	7.72	8.35	9.02	9.73	10.48
9	2.520	1.76	2.05	2.31	2.59	2.87	3.17	3.48	3.86	4.24	4.65	5.08	5.53	6.02	6.54	7.10	7.69	8.32	8.99	9.70	10.45
10	2.814	1.75	2.03	2.30	2.57	2.85	3.15	3.46	3.84	4.22	4.62	5.05	5.51	6.00	6.52	7.08	7.67	8.30	8.97	9.68	10.43
12	3.338	1.73	2.01	2.27	2.54	2.83	3.12	3.43	3.81	4.19	4.59	5.02	5.48	5.96	6.48	7.04	7.63	8.26	8.93	9.64	10.39
14	3.665	1.72	2.00	2.26	2.53	2.81	3.11	3.41	3.79	4.17	4.57	5.00	5.46	5.94	6.47	7.02	7.61	8.24	8.91	9.62	10.37
16	4.189	1.70	1.98	2.24	2.51	2.79	3.08	3.39	3.77	4.14	4.55	4.98	5.43	5.92	6.44	6.99	7.59	8.21	8.88	9.59	10.34
18	4.717	1.69	1.96	2.22	2.49	2.77	3.07	3.37	3.75	4.12	4.53	4.96	5.41	5.90	6.42	6.97	7.56	8.19	8.86	9.57	10.32
20	5.236	1.68	1.95	2.21	2.47	2.75	3.05	3.36	3.73	4.11	4.51	4.94	5.39	5.88	6.40	6.95	7.54	8.17	8.84	9.55	10.29
24	6.283	1.66	1.93	2.19	2.45	2.73	3.02	3.33	3.70	4.07	4.48	4.90	5.36	5.84	6.36	6.92	7.51	8.14	8.80	9.51	10.26
Vertical Surface		1.84	2.14	2.42	2.70	3.00	3.30	3.62	4.00	4.38	4.79	5.22	5.68	6.17	6.70	7.26	7.85	8.48	9.15	9.86	10.62
Horizontal Surface Facing Upward		2.03	2.37	2.67	2.97	3.28	3.59	3.92	4.31	4.70	5.12	5.56	6.02	6.52	7.05	7.61	8.21	8.85	9.52	10.24	10.99
Horizontal Surface Facing Downward		1.61	1.86	2.11	2.36	2.64	2.93	3.23	3.60	3.97	4.37	4.80	5.25	5.73	6.25	6.80	7.39	8.02	8.69	9.39	10.14

Values are for Flat Surfaces four square feet or more in area.

[a] To secure losses per linear foot, multiply sq ft losses in table by this factor. The losses per sq ft of pipe surface for pipes larger than 24 in. can be considered the same as the losses for the 24-in. pipe.

between the pipe and the surrounding atmosphere up to 525 F, are shown in Figs. 1, 2, and 3.

The actual thickness of many molded pipe coverings is not exactly 1 in. However, the loss through any given thickness of insulation can be obtained by interpolation.

The heat losses through 1, 1½, and 2-in. thick block, blanket, or cement insulation when applied to a flat vertical surface is given in Fig. 4.

The losses through any of the insulations given in Table 1 can be obtained by multiplying the losses obtained from Figs. 1, 2, 3, or 4 by the factors given in Table 6.

Pipes operating at high temperatures are frequently insulated to the best advantage by combining a high temperature insulation near the pipe with a moderate or low temperature insulation around it as an outer layer. By this method an efficient material may be used for each of the two temperature ranges encountered. In calculating the heat loss through such a combination the mean temperature of each layer must be determined along with the thickness of each. This is readily done in two or three calculations performed as a series of approximations, in which assumptions of thickness and mean temperature are adjusted as indicated in the discussion which follows.

In the case of a single thickness of pipe covering, the quantity of heat transferred per square foot of outer surface of the insulation is given by the equation:

$$q_o = \frac{k(t_1 - t_2)}{r_1 \log_e \frac{r_2}{r_1}} \qquad (1)$$

where

q_o = Btu per (hour) (square foot of outer surface of insulation).
r_1 = outer radius of pipe or inner radius of insulation, inches.
r_2 = outer radius of insulation, inches.
k = thermal conductivity of insulation, Btu per (hour) (square foot) (Fahrenheit degree per inch).
t_1 = temperature of inner surface of insulation, Fahrenheit.
t_2 = temperature of outer surface of insulation, Fahrenheit.

Table 3 .... Heat Loss from Horizontal Tarnished Copper Pipe[a]

Expressed in Btu per (hour) (linear foot) (Fahrenheit degree difference between the pipe and surrounding still air at 70 F)

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 psig)	297.7 F (50 psig)	337.9 F (100 psig)
	Temperature Difference—F Deg						
	50	80	110	140	157.1	227.7	267.9
½	0.250	0.287	0.300	0.321	0.433	0.500	0.530
¾	0.340	0.381	0.409	0.429	0.533	0.543	0.654
1	0.440	0.475	0.509	0.536	0.636	0.746	0.803
1¼	0.500	0.559	0.618	0.622	0.764	0.878	0.934
1½	0.580	0.656	0.710	0.750	0.904	1.053	1.120
2	0.730	0.825	0.890	0.957	1.101	1.273	1.364
2½	0.880	1.000	1.091	1.143	1.305	1.490	1.605
3	1.040	1.175	1.272	1.343	1.560	1.800	1.940
3½	1.180	1.350	1.454	1.535	1.750	2.020	2.170
4	1.460	1.500	1.635	1.715	1.941	2.240	2.430
4½	.....	.....	.....	.....	2.131	2.465	2.650
5	1.600	1.812	1.980	2.071	2.387	2.770	2.990
6	1.840	2.125	2.270	2.430	2.740	3.210	3.440
8	2.400	2.685	2.910	3.110	3.310	4.050	4.370

It is convenient to work from the outer surface of the insulation, since the loss through the covering must be determined from the outer surface loss by means of surface loss curves such as given in Fig. 5. The curves were plotted from tests conducted at the Mellon Institute.

After the true heat loss is obtained, the loss per square foot of pipe surface can be calculated from the relationship:

$$q_i = q_o(r_2/r_1)$$

where

q_i = Btu per (hour) (square foot outer surface of pipe).

Table 4 .... External Surface per Linear Foot of Copper Tubing

Outside diameter ⅛ in. greater than nominal size

Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)
⅛	0.164	2	0.556	5	1.342
¾	0.229	2½	0.687	6	1.604
1	0.295	3	0.818	8	2.128
1¼	0.360	3½	0.949	...	.....
1½	0.426	4	1.080	...	.....

The heat loss through two or more thicknesses of insulation applied to a pipe can be calculated by means of the equation:

$$q_o = \frac{t_1 - t_2}{\frac{r_2}{r_1 \log_e \frac{r_2}{r_1}}{k_1} + \frac{r_3}{r_2 \log_e \frac{r_3}{r_2}}{k_2} + \cdots} \qquad (2)$$

where

r_3 = outer radius of second layer of insulation, inches.
r_s = outer radius of last layer of insulation, inches.

The method of solving Equation 2, which is the most difficult of the two, is given in *Example 3*.

Example 3: Compute the heat loss per linear foot of pipe surface per hour from a 6-in. pipe, insulated with a 3-in. thickness of 1500 F diatomaceous earth, and a nominal 2-in. thickness of 85 percent magnesia. The pipe is operating at a temperature of 1200 F and is exposed to a room temperature of 80 F.

Solution: In figuring the heat loss from Equation 2, it is necessary to first make an assumption for the outer surface temperature t_2 and the temperature between the diatomaceous earth and 85 percent magnesia insulation, so that the mean temperature of each material can be obtained and the thermal conductivity corresponding to the mean temperature of each material substituted in the equation. First assume an outer surface temperature of 135 F and a temperature of 570 F between the two materials corresponding to a mean temperature of (1200 + 570) ÷ 2 or 885 F for the diatomaceous earth and (570 + 135) ÷ 2 or 352.5 F for the 85 percent magnesia insulation. The conduc-

Table 5 .... Area of Flanged Fittings, Square Feet[a]

Nominal Pipe Size (Inches)	Flanged Coupling		90 Deg Ell		Long Radius Ell		Tee		Cross	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.848	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2.68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
4½	1.474	2.04	3.95	5.02	4.43	5.46	6.07	7.72	7.87	10.07
5	1.622	2.18	4.44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15.41	20.41	19.58	26.26
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11

[a] Including areas of accompanying flanges bolted to the fitting.

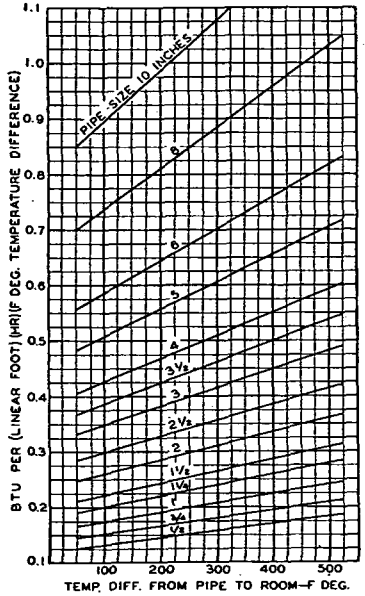


Fig. 1 .... Heat Loss through 1 In. Thick Pipe Insulation
(Use with Table 6 for Various Insulations)

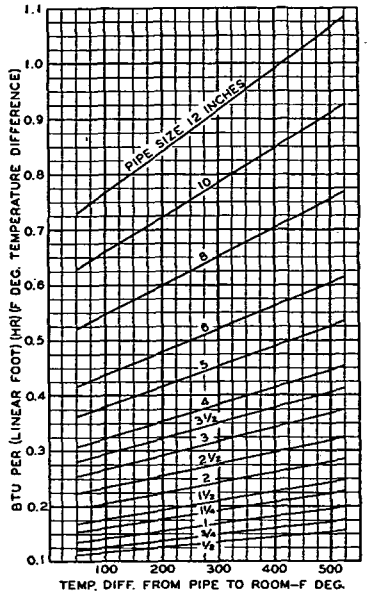


Fig. 2 .... Heat Loss through 1½ In. Thick Pipe Insulation
(Use with Table 6 for Various Insulations)

tivities of these two materials at mean temperatures of 885 and 352.5 F, interpolated from Table 1, are 0.577 and 0.466 Btu, respectively.

These values are substituted in Equation 2 and a trial calculation made.

The actual thickness of diatomaceous earth covering is 3.095 in. and that of the 85 percent magnesia is 2.125 in. For a nominal 6-in. steel pipe: $r_1 = 3.312$, $r_2 = 6.407$, and $r_3 = 8.532$. Then,

$$q_c = \frac{1200 - 135}{\frac{8.532 \log_e \frac{6.407}{3.312}}{0.676} + \frac{8.532 \log_e \frac{8.532}{6.407}}{0.466}} = \frac{1065}{8.33 + 5.24} = 78.5 \text{ Btu.}$$

The temperature drop from the outer surface of the insulation to the surrounding air for a heat loss of 78.5 Btu is found from Fig. 5 to be 49.5 deg for a 17-in. OD cylindrical surface, or 49.5 + 80 F room temperature = 129.5 F surface temperature. Since a surface temperature of 135 F was assumed, it is evident that a temperature closer to 129.5 F, or, for instance, 129 F should be used for recalculation:

$$q_c = \frac{1200 - 129}{8.33 + 5.24} = 78.9 \text{ Btu.}$$

Since the temperature drop through each material is equal to the heat flow times the actual resistance of each material, the temperature drop through the diatomaceous earth is 78.9 × 8.33 = 657 F, or the temperature between the two insulating materials is (1200 − 657) = 543 F. Since a temperature of 570 F between the two materials was assumed, it is obvious that a temperature closer to 543, or for instance 545 F may be selected. The mean temperatures of the two insulations corresponding to the new assumptions are (1200 + 545) ÷ 2 = 872.5, and (545 + 129) ÷ 2 = 337, and the interpolated conductivities corresponding to the new mean temperatures are 0.575 and 0.461 for the diatomaceous earth and 85 percent magnesia, respectively. By substituting in Equation 2

$$q_c = \frac{1200 - 129}{\frac{5.63}{0.675} + \frac{2.44}{0.461}} = \frac{1071}{8.34 + 5.30} = 78.5 \text{ Btu.}$$

Again referring to Fig. 5, it is seen that the temperature drop from the outer surface of the insulation to the surrounding air for a heat loss of 78.5 Btu = 49 deg, which corresponds to the surface temperature of 129 F last assumed. The heat loss is therefore 78.5 × 8.532 ÷ 3.312 = 202 Btu per sq ft of pipe surface. Since the surface area per linear foot of 6-in. pipe is 1.734 sq ft (Table 2), the heat loss per linear foot of pipe will be 202 × 1.734 = 351 Btu per hr.

The rate of heat loss from a surface maintained at constant temperature is greatly increased by air circulation over the surface. In the case of well-insulated surfaces, the increases in losses due to air velocity are very small as compared with increases from bare surfaces, because of the fact that air

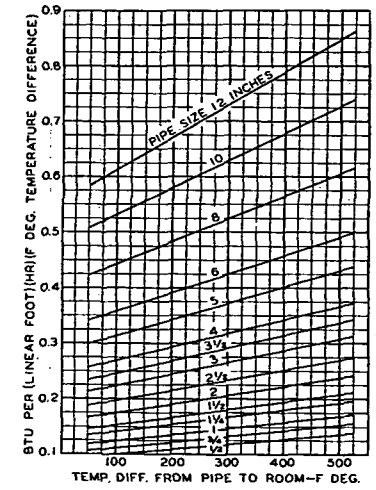


Fig. 3 .... Heat Loss through 2 In. Thick Pipe Insulation
(Use with Table 6 for Various Insulations)

flowing over the surface of the insulation can increase only the conductance of heat from surface to air, and cannot change the internal conductance of the insulation itself. The maximum increase in heat loss due to air velocity ranges from about 15 percent in the case of 1-in. thick insulation, to about 5 percent in the case of 3-in. thick insulation, provided that the insulation is thoroughly sealed so that air can flow only over the surface. If the conditions are such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given. Therefore, it is essential that insulation be applied in such a manner that air circulation within it, or between it and the pipe, is avoided.

Piping which is covered with a water-mixed product or insulation which will absorb moisture, particularly in damp locations, should be coated with a corrosion-resistant paint. This precaution may prevent early failure due to external pipe corrosion.

The frequent practice of omitting insulation on that portion of a pipe which passes through a masonry wall, or which may be in contact with other metals, should be avoided. Physical contact between the pipe surface and other structural materials of high thermal conductivity will result in heat transfer much greater than that shown in Tables 2 and 3 for transfer from bare pipe to air.

The saving due to use of insulation on piping is illustrated in *Example 4*.

Example 4: If the steam pipe given in *Examples 1* and *2* is covered with nominal 1 in. thick 85 percent magnesia, determine the resulting total annual loss through the insulation. Also compute the monetary value of the annual saving and the percentage of saving over the heat loss from the bare pipe.

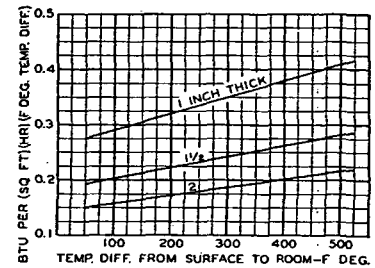


Fig. 4 .... Heat Loss through Insulation on Flat Vertical Surface
(Use with Table 6 for Various Insulations)

Solution: By referring to Fig. 1, the coefficient for 1-in. insulation on a 2-in. pipe is found to be 0.276 Btu per (hr) (linear ft of pipe) (deg temperature difference) at a temperature difference of 159.4 F. The heat insulation factor for 85 percent magnesia insulation, interpolated from Table 6, for a temperature difference of 159.4 F, is 1.085. The total hourly loss per linear foot of pipe will then be 0.276 × 159.4 × 1.085 = 47.7 Btu. The total annual loss through the insulation = 47.7 × 165 (linear ft) × (4000 hr) = 31,480 Mb. The annual bare-pipe loss as determined in the solution of *Example 1* was found to be 171,100 Mb. The saving due to insulation is then 171,100 − 31,480 = 139,620 Mb per year.

From the solution of *Example 2*, it was found that the heat *supplied to the system cost $0.804 per 1000 Mb*. Therefore, the monetary value of the saving = 0.804 (dollars) × 139.6 (thousand Mb) = $112.23 or 8.16 percent of the cost when using uninsulated pipe.

THICKNESS OF INSULATIONS TO BE APPLIED

Thermal insulation is used in a variety of thicknesses for many varied reasons. Refrigerating equipment is normally designed for minimum heat gain, and heating equipment for minimum heat loss, within economical limits. This assures minimum-sized refrigeration or heating equipment, both from the standpoint of original as well as operational cost. Present refrigeration design practice limits the heat gain to 4 to 8 Btu per (hr) (sq ft) of exposed surface. Sometimes the principal object of the insulation is to limit the heat gain or loss to a specified value to control the temperature of a manufacturing or chemical process.

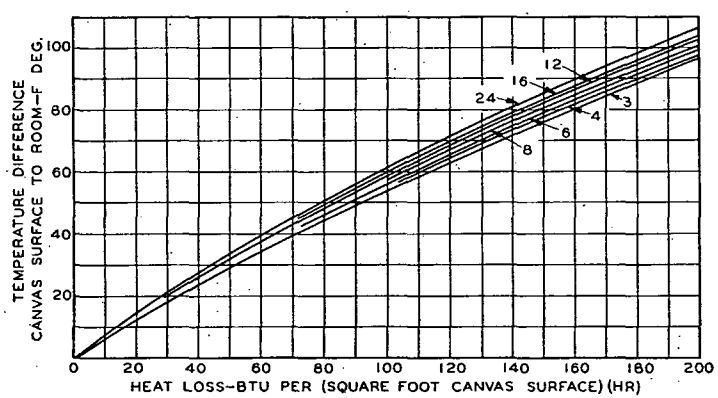
Minor refrigeration and air-conditioning equipment and cold-water lines usually utilize only sufficient insulation thickness to prevent condensation on the warm surface so as to prevent dripping. Occasionally the criterion for heated equipment and pipes is the reduction of the *outer surface* to a temperature (usually a maximum of 175 F to 200 F) low enough to prevent injury to personnel.

ECONOMICAL THICKNESS OF INDUSTRIAL INSULATION

The thicknesses of insulation which ordinarily are used for various temperature conditions are given in Table 7. Where a thorough analysis of economic thickness is desired, the thickness of insulation that will give the lowest sum of annual cost of heat loss and insulation may be found by means of Fig. 6.* As the thickness of insulation increases the annual

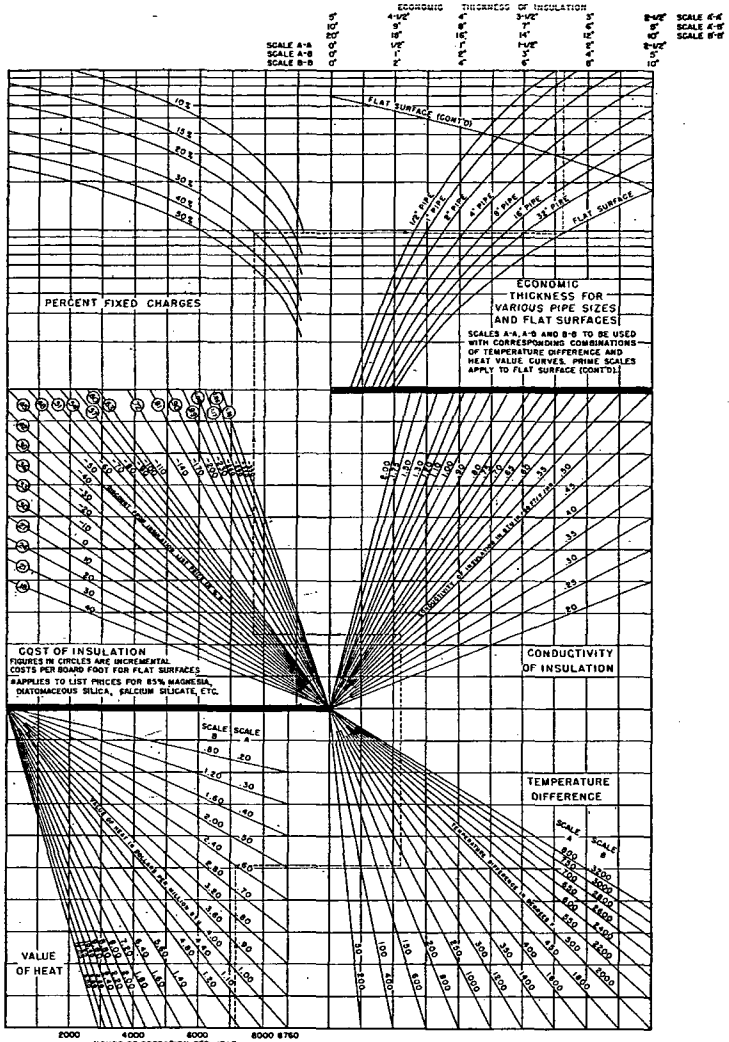
Table 6 . . . . Heat Insulation Factors for Various Insulations
(To be Applied to Figs. 1, 2, 3, and 4)

Form	Type of Pipe Insulation	Temperature Difference, Pipe to Air, F Deg							
		40	70	100	200	300	500	700	900
PIPE INSULATION	ASBESTOS								
	Molded Amosite and Binder			0.978	0.966	0.955	0.931		
	Laminated Asbestos Paper (35-40/in.)			1.152	1.131	1.110	1.067		
	Corrugated and Laminated (4 ply/in.)		1.595	1.599	1.614	1.628			
	Asbestos Paper (6 ply/in.)		1.442	1.442	1.443	1.444			
	(8 ply/in.)		1.394	1.387	1.375	1.363			
	CALCIUM SILICATE			1.041	1.018	0.996	0.950		
	CELLULAR GLASS			1.195	1.190	1.185	1.175		
	CORK (No added binder)	0.879	0.869	0.859					
	DIATOMACEOUS SILICA 22 lb						1.136	1.052	0.968
	DIATOMACEOUS SILICA 25 lb						1.296	1.206	1.115
	85% MAGNESIA	1.131	1.119	1.107	1.068	1.027	0.946		
	MINERAL WOOL (Rock, Slag, or Glass)								
	Low Temp. (Asphalt or Resin Bonded)	0.960	0.961	0.963					
	Low Temp. (Fine Fiber Res. Bond)	0.749	0.733	0.727					
	High Temp. (Metal Reinforced)			0.879	0.893	0.907	0.935		
	PLASTICS (Foamed)	0.909	0.914	0.918					
	RUBBER (Foamed)	0.765	0.762	0.759					
	WOOL FELT	0.994	0.997	1.000					
	HAIR FELT OR HAIR FELT PLUS JUTE	*0.877*	*0.884*	*0.851*					
BLOCKS AND BOARDS	ASBESTOS MOLDED AMOSITE			1.088	1.067	1.025	1.004	0.962	0.920
	CALCIUM SILICATE			0.988	0.974	0.960	0.932	0.904	0.876
	CORKBOARD (Natural Binder)	0.861	0.854	0.847					
	MINERAL WOOL (Low Temp.)	0.917	0.907	0.896					
	MINERAL WOOL (High Temp.)			0.980	0.901	0.923			
	MINERAL WOOL (Inorganic Binder)			1.003	0.990	0.978	0.953	0.928	
BLANKETS	MINERAL WOOL (Felt Flexible Type)	0.815	0.827	0.841	0.896	0.951			
	MINERAL WOOL (Felt Semi-Rigid Type)	0.830	0.840	0.851	0.895	0.938			
INSULATING CEMENT	85% MAGNESIA			1.275	1.224	1.173	1.070		
	MINERAL WOOL			1.370	1.344	1.317	1.264	1.317	



(Canvas, Roofing, and Various Colored Painted Surfaces)

Fig. 5 . . . . Heat Loss from Canvas-Covered Cylindrical Surfaces of Various Diameters



(From Reference 2 used by permission. Not to be reprinted without permission)

Fig. 6 . . . . Chart for Determining Economical Thickness of Pipe Insulation

Table 7 .... Thickness of Pipe Insulation Ordinarily Used Indoors[a]

Steam Pressure Psig or Condition	Steam Temperature Fahrenheit Degrees	Thickness of Insulation: Pipes Larger Than 4 in.	Pipes 2 in. to 4 in.	Pipes 1½ in. to 1½ in.
0 to 25	212 to 267	1 in.	1 in.	1 in.
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in.
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in.	2 in.

[a] All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in. greater than shown in this table, and covered with a waterproof jacket.

cost of heat loss decreases, but the annual cost of insulation (first cost multiplied by percent fixed charges) increases. The sum of the two costs will first decrease with increasing thickness of insulation, and then, depending on operating conditions, will increase beyond a certain thickness. Therefore, the thickness at which the sum of the costs is a minimum, is the most economical.

The method of using the chart Fig. 6 is as follows:

From the number of hours of operation per year at the bottom of the chart proceed upward to the line showing value of heat in dollars per million Btu using the appropriate scale *A* or *B*. Proceed right to temperature difference line using either scale *A* or *B*. Move upward to the line representing conductivity of the insulation being considered, then left to discount line, upward to the percent fixed charge line, right to the pipe size or flat surface line and read the economic thickness at top on the scale corresponding to the scales selected for value of heat and temperature difference. For example; with Scale *B* heat value and Scale *A* temperature difference or vice versa, use Scale *A-B* economic thickness. If the intersection with the flat surface is on the *flat surface continued* curve, the economical thickness is read on the prime scales *A'-A'*, *A'-B'*, or *B'-B'*.

Cost of insulation refers to incremental cost in place; i.e., average incremental cost per unit thickness after the first unit. The first inch of insulation will usually cost more than each additional inch. This is due to cost of finish, contractor's setup, and operating costs, which do not increase in proportion to thickness. The first inch of insulation, except in very unusual cases, provides enough saving in heat cost out of bare surface cost to more than cover these extra first costs.

Table 9 .... Data for Estimating Requirements to Prevent Freezing of Water in Pipes with Surrounding Air at −18 F

Nominal Pipe Size (Inches)	Number of Hours to Cool 42 F Water to Freezing Point: Thickness of Insulation in Inches (Conductivity, k = 0.30) 2	3	4	Water Flow Required at 42 F to Prevent Freezing, Pounds per Linear Foot of Pipe per Hour: Thickness of Insulation in Inches (Conductivity, k = 0.30) 2	3	4
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0.55	0.48
1½	1.40	1.74	2.02	0.84	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0.94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5.92	7.96	9.69	1.73	1.29	1.06
6	7.35	9.88	12.20	1.98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.43
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.45	1.93

Table 8 .... Heat Gains for Insulated Cold Pipes

Rates of heat transmission given in Btu per (hour) (Fahrenheit degree temperature difference between fluid in pipe and surrounding still air) Based on materials having conductivity, k = 0.30

Nominal Pipe Size (Inches)	Ice Water Thickness: Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Brine Thickness: Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Heavy Brine Thickness: Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface
⅛	1.5	0.110	0.502	2.0	0.098	0.446	2.8	0.087	0.394
¼	1.6	0.119	0.431	2.0	0.111	0.405	2.9	0.094	0.340
1	1.6	0.139	0.403	2.0	0.124	0.352	3.0	0.104	0.294
1¼	1.6	0.155	0.357	2.4	0.131	0.300	3.1	0.113	0.260
1½	1.5	0.174	0.351	2.5	0.134	0.270	3.2	0.118	0.238
2	1.5	0.200	0.322	2.5	0.151	0.244	3.3	0.134	0.214
2½	1.5	0.228	0.303	2.6	0.170	0.226	3.3	0.147	0.197
3	1.5	0.269	0.293	2.7	0.186	0.202	3.4	0.162	0.176
3½	1.5	0.295	0.282	2.9	0.191	0.183	3.5	0.176	0.167
4	1.7	0.294	0.248	2.9	0.209	0.176	3.7	0.182	0.154
5	1.7	0.349	0.239	3.0	0.241	0.165	3.9	0.202	0.138
6	1.7	0.404	0.233	3.0	0.259	0.150	4.0	0.228	0.130
8	1.9	0.455	0.201	3.0	0.318	0.140	4.0	0.263	0.116
10	1.9	0.559	0.198	3.0	0.383	0.135	4.0	0.309	0.110
12	1.9	0.648	0.194	3.0	0.438	0.131	4.0	0.364	0.108

To obtain discount from insulation list price, first obtain applied prices for the desired type of insulation in one-inch, two-inch, etc., thickness. Then determine the average incremental cost per inch above one-inch thickness.

Incremental cost in percent discount from list is

$$100\left(1-\frac{X}{Y}\right)$$

where

X = average incremental cost in place per inch above one inch.

Y = list price for one inch.

In figuring pipe insulation, use the list price for one-inch thickness for a pipe size consistent with the average incremental cost per inch. For example, if the average incremental cost per inch is based on applied prices for one, two, and three inches, the incremental cost of the second and third inches would be averaged and the pipe size for list price would be that nearest the average pipe size of the second and third inch.

A rapid method for determining the economical thickness of insulation by use of tables has been published.[a]

LOW TEMPERATURE PIPE INSULATION

Surfaces maintained at temperatures lower than the surrounding air are insulated to reduce the flow of heat and to prevent condensation. The insulating material should absorb a minimum amount of moisture, because the absorption of moisture substantially increases the conductivity of the material. This property is particularly important in the insulation of surfaces that are below the dew point of the surrounding air. In such cases, due to vapor-pressure difference, it is necessary to seal the surface of the insulating material against the penetration of water vapor which would condense within the material, causing a serious increase in heat flow, possible breakdown of the material, and corrosion of metal surfaces. An insulating material with a high degree of moisture absorption might pick up moisture before application and then, when the vapor seal is in place and the temperature of the insulated surface reduced, release that moisture to the cold surface. There are a number of methods of producing vapor seals, some of which have been worked out by insulation manufacturers to suit their products, and others by appliers and users. Unless time-proven methods are known, specifications of insulation manufacturers should be obtained and followed carefully.

Equipment must be carefully protected against corrosion caused by condensation of water vapor. All metallic surfaces should be coated with a vapor-impervious barrier (some types of which have an asphaltic or tar base) without any breaks or openings in order to prevent corrosion.

The thickness of insulation required to prevent condensation on the outer surface is that thickness which will raise the temperature of the outer surface of the insulation to a point slightly higher than the dew point of the surrounding vapor. The dew point for various humidities can be readily ascertained from a psychrometric chart. The external vapor barrier must be made as nearly perfect as possible in order to prevent migration of vapor into the insulation.

The approximate required thickness of insulation to prevent condensation on pipes and flat metallic surfaces may be obtained from Fig. 7 in which a surface resistance of 0.606, corresponding to a film conductance of 1.65, was used in calculating the curves. This value provides a slight factor of safety and its use is known to give satisfactory field results. In using the chart it is advisable to specify the next thicker, rather than the next thinner, commercial insulation in cases where an intermediate thickness is indicated.

Heat gains for pipes insulated with a material having an installed conductivity of 0.30 Btu per (sq ft) (hr) (F deg per in.) are given in Table 8. This table may be used for any of the commercial insulations offered for this purpose since they have conductivities very near the 0.3 value used.

INSULATION OF PIPES TO PREVENT FREEZING

If the surrounding air temperature remains sufficiently low for an ample period of time, insulation cannot prevent the freezing of still water, or of water flowing at such a velocity that the quantity of heat carried in the water is not sufficient to take care of the resulting heat losses that will cause the temperature of the water to be lowered to the freezing point. Insulation can materially prolong the time required for the water to give up its heat so that, if the velocity of the water flowing in the pipe is maintained at a sufficiently high rate, freezing may be prevented.

Table 9 may be used for making estimates of the thickness of insulation necessary to take care of still water in pipes at various water and surrounding air temperature conditions. Because damage and service interruptions may result from frozen water in pipes, it is essential that an efficient insulation be utilized. This table is based on the use of a material having a conductivity of 0.30. The initial water temperature is assumed to be 10 deg above, and the surrounding air temperature 50 deg below, the freezing point of water (temperature difference, 60 F).

The last column of Table 9 gives the minimum quantity of water at initial temperature of 42 F that should be supplied every hour for each linear foot of pipe, in order to prevent the temperature of the water from being lowered to the freezing point. The weights given in this column should be multiplied by the total length of the exposed pipe line expressed in feet. As an additional factor of safety, and in order to provide against temporary reductions in flow occasioned by reduced pressure, it is advisable to double the rates of flow listed in the table. It must be emphasized that the flow rates and periods of time designated apply only for the conditions stated. To estimate for other service conditions, the following method of procedure may be used.

If water enters the pipe at any temperature other than 42 F, the time required to cool it to the freezing point will be equal to $(t - 32)/10$ times that given in the table, or the rate of flow of water may be changed to $10/(t - 32)$ times the indicated flow rate in the last columns of Table 9. However, if the water enters the pipe at 34 F, it will be cooled to 32 F in one-fifth of the time given in the table. It will then be necessary to increase the rate of flow so that five times the specified quantity of water will have to be supplied in order to prevent freezing.

If the minimum air temperature is −38 F (temperature difference 80 F) instead of −18 F, the time required to cool the water to the freezing point will be 60/80 of the time given in the table, or the necessary quantity of water to be supplied will be 80/60 of that given.

In making calculations to arrive at the values given in Table 9, the loss of heat stored in the insulation, the effect of a varying temperature difference due to the cooling of pipe and water, and the resistance of the outer surface of the insulation to the transfer of heat to the air, have all been ne-

glected. When these factors enter into the computations it is necessary to enlarge the factor of safety. Also as stated, the time shown in the table is that required to lower the water to the freezing point. A longer period would be required to freeze the water, but the danger point is reached when freezing starts. The flow of water will stop and the entire line will be in danger as soon as the water freezes across the section of the pipe at any point.

If it is necessary to calculate the time required for the water to cool to 32 F, the following equation may be used

$$H = \frac{C_p(t_1 - 32)}{q_s\left(\frac{r}{6} \times r_s\right)} \qquad (3)$$

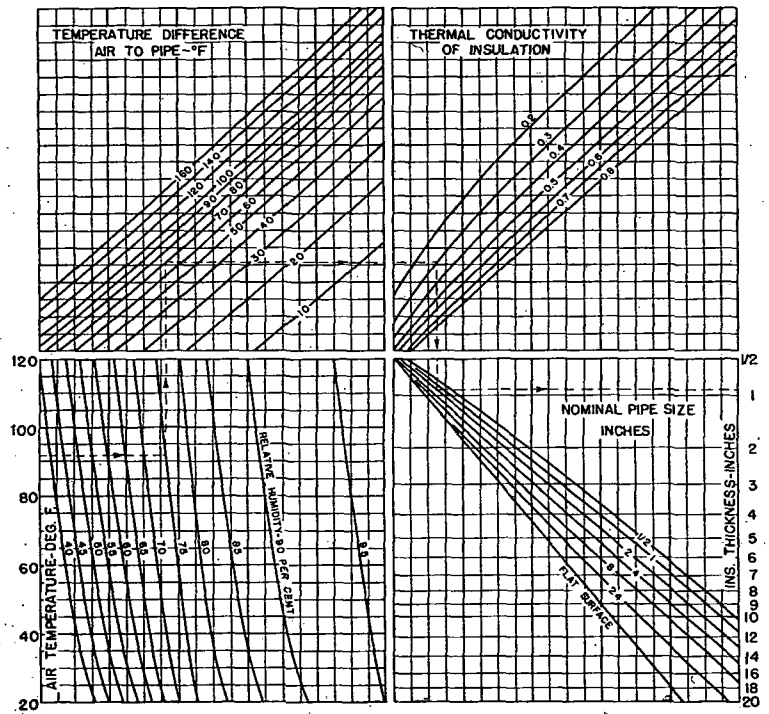
C_p = Capacity of pipe, lb water/lin. ft = $\frac{\pi r_s^2}{144} \times 62.4$

$$q_s = \frac{k\left[\left(\frac{t_1 + 32}{2}\right) - t_2\right]}{r_s \log_e \frac{r_2}{r_1}} \qquad (4)$$

where

H = hours for water to cool to 32 F.
r_s = inside radius of pipe, inches.
t_1 = initial water temperature.
C_p may be obtained from manufacturers' catalogs and handbooks.
See Equation 1 for definitions of other symbols.

When water must remain stationary longer than the times designated in Table 9, the only safe way to insure against freezing is to install a steam or hot water pipe parallel to the cold water pipe or to place an electric resistance heater along



* Solve problems as indicated by dotted line, entering chart at lower left-hand scale.

Fig. 7 .... Thickness of Pipe Insulation to Prevent Condensation on Outer Surface*

the side of the exposed water line. The heating system and the water line are then insulated so that the heat losses from the heating system are not excessive, and the heating effect is concentrated against the water pipe where it is needed. For this form of protection 2 in. of an efficient insulation may be applied.

APPLICATION OF INDUSTRIAL INSULATION

Small pipes are normally insulated with half-sectional insulation split horizontally and furnished with factory applied jackets forming hinge and lap. Large piping is insulated with segmental blocks, wired or banded in place, or with blanket insulations, particularly where removal for frequent servicing of the pipe is necessary. Indoors, sectional insulation is applied with the canvas lap pasted. Outdoors, it is necessary to provide a weatherproof finish. Fittings and bends are insulated with portions of standard preformed insulation, blanket insulations, or insulating cements. Fitting insulation should be carefully specified to be compatible with the pipe insulation. Insulation on lines carrying cold water, brine, or other cold fluids must be protected to prevent the infiltration of water vapor into the insulation.

Flat, curved, and irregular surfaces such as boilers, breechings, tanks, and vessels, are normally insulated with blocks or lagging, or with blanket forms of insulation. The insulation is secured in a variety of ways depending upon the form of insulation and contour of the surface to be insulated. One specification includes wiring the blocks in place, stretching a hexagonal mesh netting tightly over the insulation, and fastening securely with wires. Over the netting a coat of mineral wool or asbestos cement is applied with a second coat of asbestos cement trowelled to a smooth even finish. For outdoor installations weather protection is then applied. For insulation over surfaces at temperatures below ambient, water-vapor protection must be applied for both indoor or outdoor applications.

The application of insulation to air-conditioning ducts is accomplished in a variety of ways depending largely on ambient conditions and temperature differentials between the air in the room and conditioned air. Boards or blankets are normally used in a single layer 1 in. thick. Boards are fastened to the duct surfaces with spot or solid application of adhesive, and further reinforced with wires, bands, spindle anchors, or sheet-metal screws. Although, where the service is not too severe, it is permissible merely to point the joints with a mastic compound, in most cases the insulations should be protected against water-vapor permeation. Where blankets are used the insulation can be wrapped around the duct, and held in place with tape over all joints or with spirally wrapped twine. Again in most cases the insulation should be provided with an outer vapor barrier protection.

UNDERGROUND PIPE INSULATION

Underground steam distribution pipes may be installed in tunnels or in protective structures of various types, sizes, and shapes. Conduits for piping are described in Chapter 27.

Pipes in tunnels are covered with sectional insulation to provide maximum thermal efficiency, and are also finished with good mechanical protection in the form of metal or waterproofing membrane outer jackets. In some instances, where actual submersion of hot lines may occur, it has been found good practice to fasten the covering securely with corrosion-resistant wire, and then sew on a wire-inserted asbestos fabric jacket with wire. This jacket is porous. The principle of withstanding submersion is that water may enter as water, then actually boil at the pipe surfaces and escape as steam without rupturing the insulation or jacket. Conduit systems are in more general use than tunnels. Pipes carried in conduits may be insulated with sectional insulation; however, the more usual practice is to fill the entire section of the conduit around the pipes with high quality, loose insulating material. The insulation must be kept dry at all times, and for this purpose effective waterproofing membranes enclose the insulation. A drainage system is also provided to divert water which may tend to enter the conduit.

Table 10 .... Thickness of Loose Insulation for Use as Fill in Underground Conduit Systems

Steam Pressure Psig or Condition	Steam Temperature Fahrenheit Degrees	Minimum Thickness of Insulation in Inches					Minimum Distance Between Steam and Return
		Steam Lines			Return Lines		
		Pipes Less than 4 in.	Pipes 4 in. to 10 in.	Pipes Larger than 12 in.	Pipes Less than 4 in.	Pipes 4 in. and Larger	
Hot water, or 0 to 25	212 to 267	1½	2	2½	1½	1½	1
25 to 125	267 to 352	2	2½	3	1½	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¾	1½	1½

The economical thickness of insulation for underground work is difficult to determine accurately due to the many variables which have to be considered. As a result of theories previously developed, together with other experimental data which have been presented, the usual endeavor is to secure not less than 90 percent efficiency for underground piping. A comparison of heat loss under actual conditions vs. calculated values has been published by the National District Heating Association.[5] Table 10 can be used as a guide in arriving at the minimum thickness of loose insulation fills to use for laying out conduit systems. Other factors such as the number of pipes and their combination of sizes, as well as the standard conduit sizes, are primary controlling factors in the amount and thickness of insulation for use.

When sectional insulation is applied to pipes in tunnels or conduits, usual practice is to apply the most efficient materials ½ in. less in thickness than that determined by the use of Fig. 6. The data in Fig. 6 are based on conditions of insulation exposed to the air, whereas normal ground temperature is substituted for air temperature in determining the temperature difference for use with the chart when applying it for underground pipe system estimates.

REFERENCES

[1] R. H. Heilman: Heat loss from copper piping (*Heating, Piping and Air Conditioning*, September 1933, p. 458).
[2] *Technical Data Sheet* (Johns-Manville Corp., August 1955).
[3] U. W. Smith: Rapid method of determining the economical thickness of pipe insulation (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, October 1947, p. 118).
[4] J. R. Allen: Theory of heat losses from pipes buried in the ground (ASHVE Transactions, Vol. 26, 1920, p. 335).
[5] *District Heating Handbook* (National District Heating Association, 1951, 3rd ed., pp. 193-194).

BIBLIOGRAPHY

F. C. Houghten and Carl Gutberlet: Heat emission from iron and copper pipe (ASHVE TRANSACTIONS, Vol. 39, 1933, p. 97).

Heat Transmission from Surfaces (Philip Carey Mfg. Co., Bulletin 102-A).

R. H. Heilman: Surface heat transmission (Sec. 1, *Mechanical Engineering*, May 1929, p. 355).

S. Crocker: *Piping Handbook* (McGraw-Hill Book Co., New York, 1945, 4th ed.).

P. Swain: Insulation handbook (*Power*, Vol. 94, Nos. 2, 3, 5, 7, and 9, 1950).

U. W. Smith: How to insulate equipment (*Plant Engineering*, Vol. 1, December 1947, p. 32).

T. S. Nickerson and G. M. Dusinberre: Heat transfer through thick insulation on cylindrical enclosures (*ASME Transactions*, Vol. 70, 1948, p. 903).

H. J. Stoever: *Applied Heat Transmission* (McGraw-Hill Book Co., New York, 1941).

CHAPTER 33

FUELS AND COMBUSTION

Solid Fuels: Analysis, Classification of Coals, Classification of Cokes, Combustion of Solid Fuels, Firing Methods for Solid Fuels, Secondary Air, Draft Requirements and Regulations, Furnace Volume; Fuel Oils: Classification, Air Required; Fuel Gases: Classification, Heat Value, Combustion; General Combustion Principles; Air Required; Efficiency from Flue Gas Analysis; Heat Balance; Condensation and Corrosion; Soot; Air Supply to Furnace Rooms

FUELS may be classified according to their physical state as solid, liquid, or gaseous. The principal fuels used for domestic heating are coal, oil, and gas. However, coke, wood, kerosine, sawdust, briquettes, and other substances are used for heating in special applications or in localities where an adequate supply is available. Experiments are in progress in the use of a colloidal suspension of coal particles in fuel oil, but this fuel has not attained widespread usage as yet. The choice of fuel is usually based on dependability, cleanliness, availability, economy, operating requirements, and control.

SOLID FUELS

Analysis of Fuels

Coal has a complex composition that makes classification into clear-cut types difficult. Chemically it consists of carbon, hydrogen, oxygen, nitrogen, sulfur, and a mineral residue called ash. A chemical analysis provides some indication of the quality of a coal, but does not define its burning characteristics sufficiently. The coal user is interested principally in the available heat per pound of coal, the handling and storing properties, the amount of ash and dust produced, and the burning characteristics. A description of the qualities of coals and their characteristics can best be obtained from publications of the U. S. Bureau of Mines. A treatment applicable to heating boilers is given in a U. S. Bureau of Mines bulletin.[1]

There are two types of coal analyses, namely, the *proximate analysis* and the *ultimate analysis*. In the *proximate analysis* the proportions of moisture, volatile matter, fixed carbon, sulfur, and ash are determined. This analysis is quite easily made and is satisfactory for indicating most of the characteristics which are of interest to the user. For the proximate analysis the moisture is determined by observing the loss of weight of a sample of coal when dried at about 220 F. To determine the volatile matter, the dried sample is heated to about 1750 F in a closed crucible, and the loss of weight is noted. The remaining sample is then burned in an open crucible, and the accompanying loss of weight represents the fixed carbon. The unburned residue is ash. Although determined separately, the sulfur content is frequently reported with the proximate analysis because the usefulness of a coal for certain purposes depends on its sulfur content.

In the *ultimate analysis*, which is difficult to make, the percentages of carbon, hydrogen, oxygen, nitrogen, sulfur, and ash in the coal sample are determined. It is used for detailed studies of fuels, and in computing a heat balance when required in testing of heating devices. Typical ultimate analyses of the various kinds of coal are shown in Table 1.[a]

Other important qualities of coals are the screen sizes, ash fusion temperature, friability, caking tendency, and the qualities of the volatile matter. In considering these factors the following points are of interest. The volatile products given off by coals when they are heated differ materially in the ratios by weight of the gases to the oils and tars. No heavy oils or tars are given off by anthracite, and very small quantities are given off by semi-anthracite. As the volatile matter in the coal increases to as much as 40 percent of ash and moisture-free coal, increasing amounts of oils and tars are released. For coals of higher volatile content, the relative quantity of oils and tars decreases and is therefore low in the sub-bituminous coals and in lignite. The percentage of ash and its fusion temperature do not indicate the composition or distribution of the constituents.

Classification of Coals

A classification of coals is given in Table 2, and a brief description of the kinds of fuel is given in the following paragraphs, but it should be recognized that there are no distinct lines of demarcation between the kinds, and that they graduate into each other.

Anthracite is a clean, dense, hard coal which creates little dust in handling. It is comparatively hard to ignite, but it burns freely when well started. It is non-caking, it burns uniformly and smokelessly with a short flame, and it requires no attention to the fuel bed between firings.[a] Standard anthracite sizing specifications are shown in Table 3.

Semi-anthracite has a higher volatile content than anthracite. It is not so hard, and ignites somewhat more easily. Otherwise their properties are similar.

Semi-bituminous coal is soft and friable, and fines and dust are created by handling it. It ignites somewhat slowly and burns with a medium length of flame. Its caking properties increase as the volatile matter increases, but the coke formed is relatively weak. Having only half the volatile matter content of the bituminous coals, it can be burned with less production of smoke, and is sometimes called a *smokeless coal*.

Bituminous coal is a term covering a large range of coals and includes many types having distinctly different composition, properties, and burning characteristics. The coals range from the high-grade bituminous coals of the East to the poorer coals of the West. Their caking properties range from coals which melt completely, to those from which the volatiles and tars are distilled without change of form, so that they are classed as non-caking or free-burning. Most bituminous coals are strong and non-friable enough to permit the screened sizes being delivered free from fines. In general, they ignite easily and burn freely. The length of flame varies with different coals, but it is long. Much smoke and

Table 1 .... Typical Ultimate Analyses for Coals

Rank	Btu per Lb		Constituents, Percent						
	Moist, Mineral-matter-free[a]	Moist, as Received	Oxygen	Hydrogen	Carbon	Nitrogen	Sulfur	Ash	$O_2 + H_2 + C$
Anthracite	14,600	12,910	5.0	2.9	80.0	0.9	0.7	10.5	87.9
Semi-Anthracite	15,200	13,770	5.0	3.9	80.4	1.1	1.1	8.5	89.3
Low-Volatile Bituminous	15,350	14,340	5.0	4.7	81.7	1.4	1.2	6.0	91.4
Medium-Volatile Bituminous	15,200	13,840	5.0	5.0	79.0	1.4	1.5	8.1	89.0
High-Volatile Bituminous *A*	14,500	13,090	9.2	5.3	73.2	1.5	2.0	8.8	87.7
High-Volatile Bituminous *B*	13,500	12,130	13.8	5.5	68.0	1.4	2.1	9.2	87.3
High-Volatile Bituminous *C*	12,000	10,750	21.0	5.8	60.6	1.1	2.1	9.4	87.4
Sub Bituminous *B*	10,250	9,150	29.5	6.2	52.5	1.0	1.0	9.8	88.2
Sub Bituminous *C*	9,000	8,940	35.8	6.5	46.7	0.8	0.6	9.6	89.0
Lignite	7,500	6,900	44.0	6.9	40.1	0.7	1.0	7.3	91.0

[a] (Btu as received) × 100 ÷ (100 − 1.1 Ash)

Table 2 .... Classification of Coals by Rank[a]

Legend: F.C. = Fixed Carbon. V.M. = Volatile Matter. Btu = British thermal units.

Class	Group	Limits of Fixed Carbon or Btu Mineral-Matter-Free Basis	Requisite Physical Properties
I. Anthracite	1. Meta-anthracite	Dry F.C., 98 percent or more (Dry V.M., 2 percent or less)	Non-agglomerating
	2. Anthracite	Dry F.C., 92 percent or more and less than 98 percent (Dry V.M., 8 percent or less and more than 2 percent)	
	3. Semi-anthracite	Dry F.C., 86 percent or more and less than 92 percent (Dry V.M., 14 percent or less and more than 8 percent)	
II. Bituminous[d]	1. Low volatile bituminous coal	Dry F.C., 78 percent or more and less than 86 percent (Dry V.M., 22 percent or less and more than 14 percent)	Either agglomerating[b] or non-weathering[f]
	2. Medium volatile bituminous coal	Dry F.C., 69 percent or more and less than 78 percent (Dry V.M., 31 percent or less and more than 22 percent)	
	3. High volatile *A* bituminous coal	Dry F.C., less than 69 percent (Dry V.M., more than 31 percent); and moist[c] Btu, 14,000[e] or more	
	4. High volatile *B* bituminous coal	Moist[c] Btu, 13,000 or more and less than 14,000[e]	
	5. High volatile *C* bituminous coal	Moist Btu, 11,000 or more and less than 13,000[e]	
III. Sub-bituminous	1. Sub-bituminous *A* coal	Moist Btu, 11,000 or more and less than 13,000[e]	Both weathering and non-agglomerating[b]
	2. Sub-bituminous *B* coal	Moist Btu, 9500 or more and less than 11,000[e]	
	3. Sub-bituminous *C* coal	Moist Btu, 8300 or more and less than 9500[e]	
IV. Lignitic	1. Lignite	Moist Btu *less than* 8300	Consolidated
	2. Brown coal	Moist Btu less than 8300	Unconsolidated

Adapted from *ASTM Standards*, 1937, *Supplement*, p. 145, American Society for Testing Materials.

[a] This classification does not include a few coals which have unusual physical and chemical properties and which come within the limits of fixed carbon or Btu of the high-volatile bituminous and sub-bituminous ranks. All of these coals either contain less than 48 percent dry, mineral-matter-free fixed carbon, or have more than 15,500 moist, mineral-matter-free Btu.

[b] If agglomerating, classify in low-volatile group of the bituminous class.

[c] Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal.

[d] It is recognized that there may be non-caking varieties in each group of the bituminous class.

[e] Coals having 69 percent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu.

[f] There are three varieties of coal in the high-volatile *C* bituminous coal group, namely, Variety 1, agglomerating and non-weathering; Variety 2, agglomerating and weathering; Variety 3, non-agglomerating and non-weathering.

Table 3 .... Standard Anthracite Specifications[a]

Test Mesh Round

Size of Coal	Through In.	Over In.	Oversize Max %	Undersize %		Maximum Impurities %		
				Max	Min	Slate[b]	Bone[b] or Ash[c]	
Broken	4⅜	3¼ to 3	—	15	7½	1½	2	11
Egg	3¼ to 3	2 1/16	5	15	7½	1½	2	11
Stove	2 1/16	1⅜	7½	15	7½	2	3	11
Nut	1⅜	13/16	7½	15	7½	3	4	11
Pea	13/16	9/16	10	15	7½	4	5	12
Buckwheat	9/16	5/16	10	15	7½	....	....	13
Rice	5/16	3/16	10	17	7½	....	....	13
Barley	3/16	3/32	10	20	10	....	....	15
No. 4	3/32	3/64	20	30	10	....	....	15
No. 5	3/64	....	30	No limit		....	....	16

[a] Approved and adopted, effective July 28, 1947, by the Anthracite Committee (*Manual of Statistical Information*, Anthracite Institute.)

[b] When slate content in the sizes from Broken to Nut inclusive is less than above standards, bone content may be increased by one and one-half times the decrease in the slate content under the allowable limits, but slate content specified above shall not be exceeded in any event.

[c] Ash determinations are on a dry basis and include bone and ash.

A tolerance of 1 percent is allowed on the maximum percentage of undersize and the maximum percentage of ash content. The maximum percentage of undersize is applicable only to anthracite as it is produced at the preparation plant.

Slate is defined as any material which has less than 40 percent of fixed carbon.

Bone is defined as any material which has 40 percent or more, but less than 75 percent of fixed carbon.

soot are possible, if improperly fired, especially at low rates of burning.

Sub-bituminous coals occur in the western states; they are high in moisture when mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly, and have a medium length flame; are non-caking and free-burning; the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value; it is clean to handle. It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and it also is more liable to spontaneous ignition. Freshly mined lignite, because of its high moisture, ignites slowly. It is non-caking. The char left after the moisture and volatile matter are driven off burns very easily, like charcoal. The lumps tend to break up in the fuel bed, and pieces of char falling into the ashpit continue to burn. Very little smoke or soot is formed.

Classification of Cokes

Coke is produced by the distillation of the volatile matter from coal. The type of coke depends on the coal or mixture of coals used, on the temperatures and time of distillation and, to some extent, on the type of retort or oven. *Petroleum coke* is produced as a residue from the destructive distillation of oil.

COMBUSTION OF SOLID FUELS

Firing Methods for Anthracite

An anthracite fire should never be poked or disturbed, as this serves to bring ash to the surface of the fuel bed, where it may melt into clinker.

Egg size is suitable for large fire-pots (grates 24 in. and over) if the fuel can be fired at least 16 in. deep. For best results this coal should be fired deeply.

Stove is the proper size of anthracite for many boilers and furnaces. It burns well on grates at least 16 in. in diameter, on which it is fired about 12 in. deep. The fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for fire-pots up to 20 in. in diameter, and is usually fired to a depth of from 10 to 15 in.

Pea size coal is often an economical fuel to burn. When fired carefully, pea coal can be burned on standard grates. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire-door. A satisfactory method of firing pea coal consists of drawing the red coals toward the front end, and piling fresh fuel toward the back of the firebox.

Pea size coal requires a strong draft, and therefore the best results generally will be obtained by keeping the choke damper open, and regulating solely by means of the cold-air check and the air-inlet damper.

Buckwheat size coal, for best results, requires more attention than pea size coal, and in addition the smaller size of the fuel makes it more difficult to burn on ordinary grates. Greater care must be taken in shaking the grates than with the pea coal to prevent fuel from falling through the grate. In house heating furnaces, the coal should be fired lightly and more frequently than pea coal. When banking a buckwheat coal fire, it is advisable after coaling to expose a small spot of hot fire by putting a straight poker down through the bed of fresh coal. This will serve to ignite the gas that will be distilled from the fresh coal and prevent delayed ignition which, in some cases, depending upon the thickness of the bed of fresh coal, is severe enough to blow open the doors and dampers of the furnace. Where frequent attention can be given and care is exercised in manipulation of the grates, this fuel can be burned satisfactorily without the aid of any special equipment, except small mesh grates.

In general, it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and, consequently, to keep the system warm all the time, rather than to allow the system to cool off at times and then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep any clinker formed in an easily broken up condition so that it readily can be shaken through the grate. Forced draft and small mesh grates or, for greater convenience, domestic stokers are frequently used.

Buckwheat anthracite No. 2, or rice size, is used principally in stokers of the domestic, commercial and industrial type. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

Firing Methods for Bituminous Coal

A commonly recommended procedure for firing domestic heating units, called the *side-bank method*, requires the movement of live coals to one side or the back of the grate, and placing the fresh fuel charge on the opposite side. The results are a more uniform release of volatile gases, and the subjection of these gases to the high temperature of the red coals. If the fresh charge is covered with a layer of fine coal, still better results may be obtained because of slower release of volatile matter.

Bituminous coal should never be fired over the entire

fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

The importance of firing bituminous coal in small quantities at short intervals is discussed in a U. S. Bureau of Mines technical paper.[4] Better combustion is obtained by this method in that the fuel supply is maintained more nearly proportional to the air supply.

If the coal is of the caking kind, the fresh charge will fuse into one solid mass that can be broken up with the stoking bar and leveled from 20 min to an hour after firing, depending on the temperature of the firebox. Care should be exercised when stoking not to bring the bar up to the surface of the fuel, as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible, and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

It is acknowledged that it may be difficult to apply the outlined methods to domestic heating boilers of small size, especially when frequent attendance is impracticable. The adherence to these methods insofar as practicable, however, will result in better combustion.

Preventing Smoke

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Special care must be taken in hand-firing bituminous coals.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned, and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the solution of the problem presents many difficulties, and may require a considerable investment in special apparatus.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal has been removed, will not, as a general rule, produce as much dust and cinders as will result from the burning of non-coking coals and slack coals when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of floor space, because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of dust and cinder catcher must be installed.

Firing Methods for Semi-Bituminous Coal

The Pocahontas Operators' Association recommends the central cone method of firing, in which the coal is heaped on the center of the bed to form a cone, the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, while the fines remain in the center and are coked. The poking should be limited to breaking down the coke without stirring. Grates should be rocked gently. It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides usually admits the required air.

Firing Methods for Coke

Coke ignites less readily than bituminous coal and more readily than anthracite, and burns rapidly with little draft. In order to control the air admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly. A coke fire responds rapidly to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to prevent the fire from burning too rapidly. In order to obtain the same interval of attention as with other fuels, a deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather, and should be shaken only until the first red particles drop from the grates in cold weather. The best size of coke for general use, for small fire-pots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a 1½ in. screen. For large fire-pots where the fuel can be fired over 20 in. deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a coke of uniform size is always more satisfactory. Large sizes of coke should either be mixed with fine sizes or broken up before using.

SECONDARY AIR

When bituminous coal is hand-fired in a furnace, the volatile matter in the fuel is driven off by heat leaving coke on the grate. The product of combustion of the coke is CO_2 and under certain conditions some CO may arise from the bed. The combustion of the volatile matter and the CO may amount to the liberation of from 40 to 60 percent of the heat in the fuel in the combustion space over the fuel bed.

The air that passes through the fuel bed is called *primary air*, and the air that is admitted over the fuel bed in order to burn the volatile matter and CO is called *secondary air*.

This process of combustion is illustrated in Fig. 1.[4] The free oxygen of the air passes through the grate and the ash above it, and burns the carbon in the lower 3 or 4 in. of the fuel bed forming carbon dioxide. This layer noted as the oxidizing zone, is indicated by the symbols CO_2 and O_2. Some of the carbon dioxide of the oxidizing zone is reduced to carbon monoxide in the upper layer of the fuel bed, noted as the reducing zone and indicated by the symbols CO_2 and CO. The gases leaving the fuel bed are mainly carbon monoxide, carbon dioxide, nitrogen, and a small amount of free oxygen. Free oxygen is also admitted through the firing

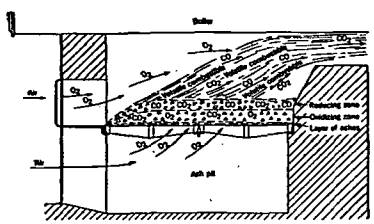


Fig. 1 .... Combustion of Fuel in a Hand-Fired Furnace

door in an attempt to burn carbon monoxide, as well as the volatile combustible distilled from the freshly fired fuel.

Size of the fuel is a very important factor in fixing the quantity of secondary air required for non-caking coals. With caking coals it is not so important, because small pieces fuse together and form large lumps. Fortunately, a smaller size fuel gives more resistance to air flow through the fuel bed, and thus automatically causes a larger draft above the fuel bed. More secondary air is drawn through the same slot openings, but, nevertheless, the smallest size of fuel will require the largest secondary air openings. For certain sizes of fuel, no secondary air openings are required, and for large sizes, too much excess air may pass through the fuel bed.

In general, the efficiency of domestic hand-fired furnaces and boilers burning either anthracite or bituminous coal, can be increased for an hour or two after firing, if some secondary air is admitted through slots of the fire door. However, unless the slots are closed when secondary air is no longer beneficial, the decrease in efficiency during the remainder of the firing cycle, because of excess air, may more than offset the gain resulting from the secondary air at the beginning of the firing period. Unless the secondary air can be readjusted between firings, it is probable that a greater average efficiency will be obtained for domestic hand-fired devices by leaving the secondary air slots closed at all times. There is usually an appreciable amount of air leakage around the firing door and secondary air slots of domestic furnaces and boilers.

Draft Requirements for Coal and Coke Firing

The draft required to effect a given rate of burning the fuel is dependent on the following factors: (1) kind and size of fuel, (2) grate area, (3) thickness of fuel bed, (4) type and amount of ash and clinker accumulation, (5) amount of excess air present in the gases, (6) resistance offered by the boiler passes to the flow of the gases, and (7) accumulation of soot in the passes.

Insufficient draft will necessitate additional manipulation of the fuel bed, and more frequent cleanings to keep its resistance down. Insufficient draft also restricts the control that can be accomplished by adjustment of the dampers. For draft requirements see Chapter 36.

Draft Regulation for Coal and Coke Firing

Since heating load demands vary in most installations, it is necessary to vary the rate of fuel burning. The maintenance of the proper air supply for the various rates of burning is accomplished by regulation of the drafts. Methods of draft regulation used for solid fuel are shown in Fig. 2.

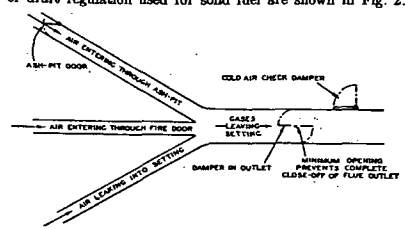


Fig. 2 .... Methods of Draft Regulation in a Hand-Fired Furnace

The air enters through the ashpit draft door, firing door, and by leakage through the setting, whereas the gases leave only through the outlet. By throttling flue gases with the damper in the outlet all the air entering by each of the three intakes is reduced in the same proportion, thus maintaining about the same percent of excess air. If inlet air is controlled by the ashpit draft door, the air admitted through the ashpit is reduced, while it is increased through the other two intake openings, resulting in an increase of excess air. A considerable increase in the efficiency of hand-fired furnaces and boilers can be realized by regulating the air supply by means of the damper in the outlet instead of the ashpit damper. Use of the ashpit damper is required, of course, for low rates of combustion. The cold air check damper is to be used only when chimney draft is excessive. It is normally closed unless the rate of combustion cannot be controlled by closing of the outlet and ashpit dampers.

Furnace Volume for Coal and Coke

The furnace volume is influenced by the kind of coal used. Bituminous coals, due to their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals, provision should be made for mixing the combustible gases thoroughly, so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly. Anthracite requires comparatively little combustion space.

CLASSIFICATION OF FUEL OILS

Fuel oils are mixtures of hydrocarbons derived from crude petroleum by refining processes. Crude oil is distilled in towers at atmospheric pressure to produce gasoline, naphtha, kerosine and gas oil, and leave straight-run residues. The straight-run residues are distilled under vacuum to produce heavy distillates. The vacuum-flashed residue remaining can then be thermally cracked to produce more gasoline, cracked naphtha, cracked distillates, and cracked residue. The exact processes used depend on the proportions of the various end products desired, and to some extent upon the composition and characteristics of the crude oil.

Fuel oils may be designated as straight-run, thermally-cracked, catalytically-cracked, or blended, depending on the refining process used to produce them. Straight-run fuels are those produced by distillation under atmospheric pressure without decomposition of the hydrocarbons by cracking. Thermally-cracked fuels are those produced by a cracking process involving elevated temperatures (850–1100 F) and pressures (300 psig) to decompose the heavier hydrocarbons. Catalytically-cracked fuels are those produced with the aid of an alumina-silica catalyst in the cracking process at lower pressures than those used for thermal cracking. Blended fuel oils are mixtures of any of the above three types.

Grade Classification of Fuel Oils

Fuel oils are most commonly classified by dividing them into grades in accordance with the *Commercial Standard* (CS12-48) entitled *Fuel Oil* published by the *U. S. Department of Commerce*. These specifications, given in Table 4, conform to *ASTM Materials Specifications for Fuel Oils* D 396 – 48. Table 4 also shows specifications for PS 300 and PS 400 fuel oils which are numbers used in the Pacific

Table 4 .... Detailed Requirements for Fuel Oils[a]

Grade of Fuel Oil[b]	Flash Point F	Pour Point F	Water and Sediment % by Vol.	Carbon Residue on 10% Residuum %	Ash % by Wt.	Distillation Temperatures F			Saybolt Viscosity Seconds				Kinematic Viscosity Centistokes			
						10% Point	90% Point	End Point	Universal at 100 F		Furol at 122 F		at 100 F		at 122 F	
	Min.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
1. A distillate oil intended for vaporising pot-type burners and other burners requiring this grade[d] API gravity 35 (min.)	100 or Legal	0	Trace	0.15		420		625					2.2	1.4		
2. A distillate oil for general purpose domestic heating for use in burners not requiring No. 1 API gravity 26 (min.)	100 or Legal	20[e]	0.10	0.35		[c]	675		40				(4.3)			
4. An oil for burner installations not equipped with preheating facilities	130 or Legal	20	0.50		0.10				125	45			(26.4)	5.8		
5. A residual type oil for burner installations equipped with preheating facilities	130 or Legal		1.00		0.10					150	40			32.1	(81)	
6. An oil for use in burners equipped with preheaters permitting a high viscosity fuel	150 or Legal		2.00[f]								300	45			(638)	(92)
PS 300	150 or Legal		1.0								40	25				
PS 400	150		2.0[f]									60				

[a] Recognizing the necessity for low sulfur fuel oils used in connection with heat-treatment, non-ferrous metal, glass and ceramic furnaces and other special uses, a sulfur requirement may be specified as 0.5% for No. 1, 1.0% for No. 2, and no limit for Nos. 4, 5 and 6.

Other sulfur limits may be specified only by mutual agreement between the buyer and seller.

[b] It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c] Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0 F under any condition.

[d] No. 1 oil shall pass test for corrosion made in accordance with paragraph 15 *ASTM Specifications for Fuel Oils*, D 396 - 48.

[e] The 10 percent point may be specified at 440 F maximum for use in other than atomizing burners.

[f] The amount of water by distillation, plus the sediment by extraction, shall not exceed 2.00 percent. The amount of sediment by extraction, shall not exceed 0.50 percent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 percent.

PS 300 and PS 400 are Pacific Specification numbers. Balance of table from *Commercial Standard* CS 12-48 and *ASTM Materials Specification for Fuel Oils* D 396 - 48.

PS 300 is a residual oil for use without preheating in furnaces and burners requiring a low viscosity fuel, and commonly described as light fuel oil, domestic fuel oil, or low viscosity fuel oil.

PS 400 is a residual oil for use in furnaces and burners equipped with preheaters permitting a high viscosity fuel, and commonly described as industrial fuel oil, heavy fuel oil, or high viscosity fuel oil.

coast area. These oils correspond approximately to ASTM designations No. 5 and No. 6 respectively.

Oils are classified by their viscosities. Other characteristics of fuel oils which determine their grade classification in the *Commercial Standard*, and their suitability for given uses are the flash point, pour point, water and sediment content, carbon residue, ash, and distillation characteristics.

The flash point of an oil is important with regard to safety in handling and in storage. The distillation characteristics determine whether or not the oil can be completely evaporated in some types of burners, and whether cracking will be likely to occur prior to combustion. A low pour point and low water content are necessary for handling and outdoor storage in cold climates. Sediment should be low to prevent clogging of strainers. Carbon residue is related to the rate of accumulation of unburned material in some types of burners. A low viscosity allows the fuel oil to flow through supply lines readily and to be broken up into small droplets in atomizing type burners. The sulfur content is of importance because undesirable compounds of sulfur may be formed in certain industrial processes. Some of these characteristics of a fuel oil are required to lie within certain limits for each of the grades of fuel oil listed in *Commercial Standard* CS12-48. Some fuel oils do not fall into any of the grade classifications of the *Commercial Standard* because failure to comply with all of the requirements of one grade does not automatically place the fuel oil in the next lower grade, unless it meets all of the requirements of the lower grade.

Table 5 .... Approximate Gravity and Calorific Value of Standard Grades of Fuel Oil

Commercial Standard No.	Approximate Gravity API	Weight lb per gallon	Calorific Value Btu Per Gallon
1	38–45	6.951–6.675	137,000–132,900
2	30–40	7.296–6.870	141,800–135,800
4	12–32	8.212–7.206	153,300–140,600
5	8–20	8.448–7.778	155,900–148,100
6	6–18	8.571–7.882	157,300–149,400

By ultimate analysis the No. 1 and No. 2 fuel oils contain 84 to 86 percent carbon, up to 1.0 percent sulfur, and the remainder predominantly hydrogen. The heavier grades of fuel oil, Nos. 4, 5 and 6, may contain as much as 88 percent carbon, as low as 11 per cent hydrogen, and considerably more sulfur than is permissible in the domestic grades.

Due to variation in the constituents of different fuel oils and the different refining methods used, the API gravities and calorific values of the different grades of fuel oil cover a range in each grade with some overlapping between domestic grades and between commercial and industrial grades. API *gravity*, generally used when referring to specific gravity of oil, is obtained by the following formula in which the term *specific gravity at 60/60 F* indicates the ratio of the weight of a given volume of oil at 60 F to the weight of the same volume of water at 60 F:

$$\text{Degrees API} = \frac{141.5}{\text{specific gravity at } 60/60 \text{ F}} - 131.5$$

The API gravity of water at 60 F is 10.0. The relation between the API gravity of fuel oils and their calorific value is shown in Table 5. Grades No. 1 and No. 2, which are distillates, are used predominantly in domestic heating equipment whereas grades 5 and 6 are used in commerical and industrial burners. Grade 6 requires preheating to increase its fluidity and to permit atomization, whereas grade 5 may be used in some burners without preheating. Grade 4 fuel oil does not require preheating and can even be burned satisfactorily in a limited number of domestic burners.

Sources and Properties of Residual Fuel Oils

Fuel oils Nos. 5 and 6 always contain residues remaining from the refining of crude oils. No. 4 oil will sometimes consist entirely of heavy distillates but generally contains residual stocks.

The residues may be classified, in accordance with their previous processing, as straight-run, vacuum-flashed, or cracked residues. Straight-run residues are the materials in crude oil that cannot be distilled at atmospheric pressure without appreciable decomposition and hence are removed as bottom fractions from crude topping units. These materials, often referred to as *topped* or *reduced* crude, are sometimes blended into fuel oil, but more frequently are processed, either by vacuum distillation or by thermal cracking, to produce additional distillates. Residues from vacuum distillation processes are widely used as residual fuel blending stocks. Vacuum-flashed residues from some crudes must be subjected to a mild thermal cracking, known as viscosity-breaking, before they can profitably be blended into fuel oils. Residues produced by thermal cracking of straight-run or vacuum-flashed residues or by thermal reforming of heavy naphthas, also are used in residual fuel oils.

In processing reduced crudes and other stocks that yield residual fuel-blending stocks, the refiner evaluates the severity of treatment against, among other factors, the effect on certain properties of the residues. Perhaps the most significant of these properties is viscosity. Maintenance of fuel viscosity within a certain range by preheating of the oil is necessary to obtain satisfactory combustion efficiency with most burners for heavy oil. In most instances there is a maximum temperature to which the oil should be preheated. Consequently a maximum viscosity is usually placed on the finished product.

The second property of oil that is markedly sensitive to refining procedures is specific gravity. Residues may have gravities on either side of API 10, i.e., they may be either lighter or heavier than water. Since residual fuels in many instances are stored and used in the presence of water the densities of the fuel oils must differ sufficiently from water to assure their separation into distinct layers.

The limits imposed by viscosity and specific gravity specifications are met in part by refining practices and in part by blending residues with distillates. These distillates may be straight-run gas oils and distillates produced by thermal or catalytic cracking. Blending of residues and distillates generally entails down-grading of the latter. Consequently efforts are made to reduce to a minimum the quantities of *cutter stock* (the refinery term for distillates used for this purpose) necessary to meet manufacturing specifications. Heavy catalytically- or thermally-cracked distillates are generally used as cutter stock, not only because they are of less value for further processing than straight-run gas oils, but also because they have greater solvency for asphaltic materials in cracked residues.

This discussion of residual fuel oils has been concerned principally with No. 6 fuel oil, the heaviest grade of residual fuel. Two other grades of residual fuels, Nos. 4 and 5, are produced and marketed. These fuels have lower viscosities and higher gravities than No. 6 fuel oil. Generally these specifications are met by increasing the ratio of cutter stock to residue, but occasionally these lighter fuels are made by blending No. 6 and No. 2 fuel oils.

Sometimes No. 4 oil will consist entirely of heavy distillates but generally this grade of fuel contains residual stocks. Due to their higher distillate content both No. 4 and No. 5 grades bring higher prices than No. 6 fuel oil, and frequently they are priced on a different basis, i.e., in cents-per-gallon rather than dollars-per-barrel.

Oil Burning Indexes

A number of indexes have been used, or proposed, as an indication of the burning qualities of distillate fuel oils based on one or more physical measurements made on the oil. These may be summarized as follows:

A. Indexes based on a single physical test: (1) API gravity,[a] (2) Aniline point,[a] (3) Institute of Petroleum smoke test, (4) Carbon-Hydrogen ratio based on flue gas analysis or ultimate analysis, and (5) Percent aromatics determined by sulfuric acid absorption tests.[a]

B. Indexes based on two or more physical tests: (1) Diesel index[a] based on API gravity and aniline point, (2) Institute of Petroleum cetane number[a] based on the API gravity and 50 percent distillation point, (3) Universal Oil Products characterization factor[a] based on specific gravity and average

boiling point, (4) Burning index[20] based on API gravity and 50 percent distillation point, and (5) Estimated Carbon-Hydrogen ratio[21] based on API gravity aniline point, and boiling point.

Various investigators have shown correlation between one or more of these indexes and the performance of fuel oils in oil burners. Experiments conducted with the Oil Heat Institute Reference Test Unit[12] indicated good correlation between the smoking tendency of fuel oils and API gravity, burning index, Diesel index, and hydrogen-carbon ratio for a limited number of oils in laboratory apparatus simulating a pressure-atomizing burner. These results are shown graphically in Fig. 3. Smoking tendency is given here in terms of smoke-spot

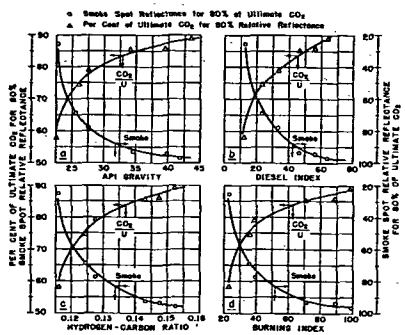


Fig. 3 .... Correlation of Burning Qualities of Fuel Oils with Four Combustion Indexes

reflectance, the light reflectance of a smoke-soiled filter paper. A high reflectance, relative to a clean filter paper, indicates low smoking tendency. CO_2/U is the observed CO_2 divided by the ultimate or maximum theoretical CO_2 expressed as a percentage. Reid and Hersberger[20] have related burning qualities and burning index for various oils in a wall-flame burner. Cauley and Delgass[21] cite test results on combustion indexes obtained with vaporizing burners. The present experimental data are probably too meager as yet to correlate adequately any one of these indexes with burning qualities of oil fuel for all types of burners. Few attempts have been made to suggest limits for any of these fuel oil indexes for particular applications, even though correlations between them and burning qualities have been observed.

Experiments have shown that the rate of thermal decomposition or cracking of hydrocarbons becomes appreciable at a temperature of approximately 680 F at atmospheric pressure. Thus distillate fuel oils, whose end point approximates this temperature, cannot be completely evaporated in vaporizing-type oil burners at atmospheric pressure without leaving a residue. A complete distillation curve cannot usually be determined for fuel oils containing fractions that evaporate above 680 F.

Since No. 1 grade fuel oil in *Commercial Standard* CS12-48 has a maximum end point of 625 F, although an end point this high is rare, it can in most cases be completely evaporated in atmospheric vaporizing burners without cracking, although occasionally an oil is found that undergoes cracking at temperatures below 625 F. By the same criterion, No. 2 grade fuel oil in the *Commercial Standard*, which can have a maximum distillation temperature of 675 F at the 90 percent point, would generally be cracked in a vaporizing burner. Vaporizing-type burners, therefore, can use only No. 1 fuel oil with assurance that thermal decomposition will not occur during combustion. On the other hand either No. 1 or No. 2 fuel oils may be employed in high- or low-pressure atomizing burners when the temperatures developed in the combustion chamber are high enough to assure complete combustion, even if the fuel oil is thermally decomposed.

In vaporizing burners there is complete evaporation of fuel before it is exposed to intense heat. Thorough mixing of the air and gasified fuel promote complete combustion without smoke and with a minimum of excess air. In pressure-type burners good atomization of the fuel, good mixing of the atomized oil with the air, air turbulence, and high combustion chamber temperatures (preferably red hot) promote smokeless combustion with a minimum of excess air.

Natural draft burners depend on the motivating force of a chimney to induce enough air into the burner for complete combustion. Forced draft burners are supplied with combustion air by means of a blower or fan; the chimney merely conducts the flue gases outdoors and prevents leakage of flue gases into the building. More details on the operation of the different kinds of oil burners and on chimneys and draft will be found in Chapters 34 and 36 respectively.

FUEL GASES

Fuel gases employed for various heating and air conditioning processes throughout the United States fall into three broad classifications: natural, manufactured, and liquefied petroleum. Natural gas is a mixture of several combustible gases and, usually, a small percentage of inert gases obtained from geologic formations. Manufactured gas is made by the distillation or cracking of oil or coal, by the steam-carbon reaction, or by combinations of these processes. Liquefied-petroleum gases (propane and butane) are higher hydrocarbon gases normally obtained as a by-product of oil refineries or by stripping natural gas. These two compounds are generally gaseous under usual atmospheric conditions although they can be liquefied by the application of moderate pressures at normal temperatures.

In American gas practice the heating value of a gas and also appliance efficiencies are based on the gross heating value. This value is the number of Btu liberated by complete combustion, at constant pressure, of one cubic foot of gas saturated with water vapor and measured at 60 F and 30 in. of mercury, with air at the same temperature and pressure. Products of combustion are cooled to the initial temperature of the gas and air and the water formed by combustion of free and combined hydrogen is condensed to the liquid state.

Classification of Gases

Representative properties of gaseous fuels commonly employed for domestic heating processes are shown in Table 6.[12, 14, 15, 16]

Natural gas contains from 55 to 98 percent methane with various percentages of higher hydrocarbons, chiefly ethane. In addition to these components, small quantities of non-combustible gases such as carbon dioxide, nitrogen, and helium are sometimes present. Percentages of the different components vary with the area from which natural gas is withdrawn. They may even vary slightly from any given well

Table 6 .... Typical Gas Analyses

Constituents of Gas—Percent by Volume

No. Type	Source	CO_2	O_2	N_2	CO	H_2	CH_4	C_2H_6	C_3H_8	C_4H_{10}		Specific Gravity[a]	Btu per Cu Ft[a] Gross	Net
1 Natural gas	Typical	..	..	0.8	..	..	83.4	15.8	..	..	..	0.61	1129	1021
2 Propane	Commercial (natural gas)	..	..	..	..	..	..	2.2	97.3	0.5	..	1.55	2558	2358
3 Propane	Commercial (refinery gas)	..	..	..	..	..	..	2.0	72.9	0.8	C_4H_8 24.3	1.77	2504	2316
4 Butane	Commercial (natural gas)	..	..	..	..	..	..	..	6.0	94.0[c]	..	2.04	3210	2961
5 Butane	Commercial (refinery gas)	..	..	..	..	..	..	..	5.0	66.7[b]	C_4H_8 28.3	2.00	3184	2935
6 Butane-air	.....	..	..	..	..	..	..	..	..	17.2	Air 82.8	1.16	550	516
										(Illuminants)				
7 Reformed natural	Straight shot generator	2.1	0.3	1.5	13.6	48.3	33.0	..	..	1.2	..	0.41	559	497
8 Refinery oil	Vapor phase crack	0.2	0.2	0.5	1.2	13.1	23.3	21.9	..	39.6	..	0.89	1475	1351
9 Oil gas	Portland, Ore.	1.2	0.5	2.4	7.7	54.2	30.1	..	..	3.9	..	0.37	570	510
10 Coal gas	Contin, Vert.	3.0	0.2	4.4	10.9	54.5	24.2	..	..	2.8	..	0.42	532	477
11 Coke oven gas	By-product	2.0	0.03	4.8	5.5	51.9	32.3	..	..	3.2	..	0.40	569	509
12 Producer gas	Bituminous	4.5	0.6	50.9	27.0	14.0	3.0	..	..	..	..	0.86	163	153
13 Blast furnace gas	.....	11.5	..	60.0	27.5	1.0	..	..	..	..	..	1.02	92	92
14 Blue gas	Bituminous	5.5	0.9	27.6	28.2	32.5	4.6	..	..	0.7	..	0.70	260	239
15 Carburetted water gas	Heavy oil with blow run	6.0	0.9	12.4	26.8	32.2	13.5	..	..	8.2	..	0.66	530	451
16 Carburetted water gas	High Btu	0.7	0.3	5.8	11.7	28.0	36.1	..	..	17.4	..	0.63	840	770
17 Mixed Gas	Natural, Blue, Producer, Reformed Natural	3.9	0.1	15.5	10.3	17.8	46.9	5.4	0.1	..	..	0.63	650	586
18 Mixed Gas	Coal, Natural	1.1	0.4	6.6	4.0	35.8	45.7	4.6	..	Ill. 1.8	..	0.46	700	627
19 Mixed Gas	Natural, Coke Oven	1.0	0.2	5.5	2.0	20.9	66.6	2.6	..	Ill. 1.2	..	0.52	802	721
20 Sewage Gas	Decatur, Ill.	22.0	..	6.0	..	2.0	68.0	..	..	..	..	0.79	690	621

This table abstracted from *Gaseous Fuels*, American Gas Association, and from A.G.A. Laboratories Data.

[a] At 60 F and 30 in. mercury, absolute pressure. [b] Divided 16.5% isobutane, 50.1% n-butane. [c] Divided 33.3% isobutane, 70.7% n-butane.

during its lifetime but these variations are inconsequential insofar as the utilization of the gas is concerned. Heating values of natural gases vary from 900 to 1400 Btu per cu ft but the usual range for use is from 1000 to 1050 Btu (gross) per cu ft. A typical analysis is given in Table 6.

Manufactured gases commonly produced are listed in Table 6. Gross, or higher, calorific values of typical send-out gases made from these manufactured gases generally range from 500 to 600 Btu per cu ft. Due largely to the greatly increased demand for city gases the tendency during recent years has been to increase rather than to decrease heat content of manufactured gases, thus making it possible to serve more customers through the existing distribution system.

Mixed gases are a result of increased distribution of natural gas, through transcontinental transmission lines, into areas having existing manufactured gas facilities. In such instances some gas companies supply a 600 to 800 Btu mixture (see Table 6). In some territories these mixtures are distributed as an intermediate step in changing over from manufactured gas to natural gas. Although the burden of adjusting installed heating and air-conditioning equipment and supplying new orifices and burner equipment is generally assumed by the gas companies when the gas is changed, it is advisable to consult the local gas company to insure that equipment is provided with proper orifices and burners when installed.

Most states enforce legislation through their public service commissions to require delivery of a gas of specified average or minimum heating value within their respective limits. Any given heating value within reason fortunately may be maintained and yet permit considerable latitude in the composition of the gas distributed. Hence the constituents of city gases are not necessarily the same in different districts nor even at successive stations in the same district. In every community, however, the objective is to maintain variations in composition and gas pressure within limits that will provide satisfactory operation and performance of all common types of gas burning equipment.

Liquefied petroleum gases, such as propane and butane or mixtures thereof have calorific values ranging from 2500 to 3200 Btu per cu ft. These fuels are often supplied as liquids under pressure in tanks or bottles. In such cases the liquid evaporates when pressure is relieved, the heat necessary for vaporization being obtained from the surrounding air or ground. As butane boils at 32 F some provision is necessary for maintaining the gas above this temperature or for lowering the partial pressure by dilution if the gas is utilized in colder climates. Propane, with a boiling point of −40 F, may be served in localities where temperatures substantially below freezing are encountered. When employed for heating purposes, these gases are usually stored in high pressure tanks and delivered by tank truck in much the same manner as fuel oil. Both gases, mixed with air, or in undiluted form, are also extensively employed by gas companies to augment their base load supplies during peak load periods. In some smaller communities, where gas manufacturing plants are not economically feasible and natural gas is unavailable, liquefied petroleum gases or liquefied petroleum gas-air mixtures are supplied through mains in much the same manner as manufactured or natural gas.

COMBUSTION OF GASEOUS FUELS

Gas burners employed in domestic heating appliances are generally of the non-luminous flame or Bunsen type. Part of the air required for combustion is inspirated as *primary air* into the burner mixing tube where it mixes with gas, and then takes part in combustion at the burner ports. As the amount of primary air is seldom sufficient to support complete combustion, additional air is supplied to the burner flames around the periphery of its ports. This *secondary air* is induced into the appliance and around the base of each separate burner flame by force of the issuing mixture of gas and primary air and by draft inspiration inherent in the heat of the flames. If a Bunsen type burner is properly adjusted, its flames will generally have a clear, blue appearance. Yellow flames are indicative of insufficient primary air supply, and possibly of incomplete combustion. An appreciable updraft is seldom, if ever, present even in flue-connected gas heating appliances, because most appliances of this kind are equipped with a draft hood which reduces the chimney draft at the appliance. It is important to note that gas furnaces and boilers, as well as most other classes of heating equipment, are designed to create their own draft.

The air-to-gas ratio in a Bunsen burner head has a decided effect on the rate of flame propagation. The gas-air mixture must flow from the burner ports faster than the flame burns, otherwise flashback will occur. The flashback condition normally results either from an excess amount of primary air or insufficient gas or both. Conversely, the velocity of the issuing mixture must not be so high that the flame will be blown from the ports, a condition known as *lifting*. Fortunately, contemporary types of such burners have a rather wide range of flexibility in capacity and adjustment. In addition to this characteristic, gas supply is normally so uniform that if a gas heating appliance is properly adjusted when it is installed, its burners, with occasional cleaning, should provide trouble-free service for years. When problems incident to changeover of the gas supply are involved, they are generally assumed by the local utility providing the supply of gas. In many instances when the gas supply is changed, it is necessary to change the burners to those suitable to burn the new kind of gas. It should be recognized that a change in fuel gas will change the operating CO_2 value. For example, an appliance operating on carburetted water gas at 20 percent excess air will have 14.2 percent CO_2 in the flue gases. If a change is made to coke oven gas at the same gas input rate, with the excess air maintained at 20 percent, the operating CO_2 would drop to 9.2 percent.

Luminous flame burners are occasionally used in central heating gas appliances. With these devices all air required for complete combustion is supplied to the flames as secondary air. Two fundamental advantages of this type of burner are that the possibilities of flashback are eliminated, and that a much higher gas velocity is needed to blow the flames off the ports. On the other hand, if there is any appreciable amount of flame impingement on any portion of the heating surface, or if secondary air is not effectively supplied to the flames, soot may be formed and also combustion may be incomplete.

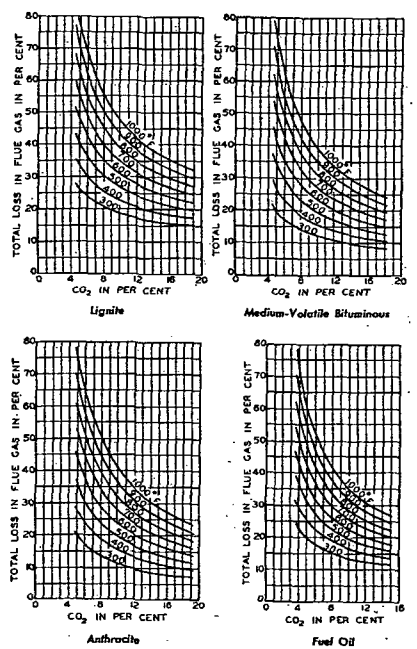
In some types of gas burners radiant baffles are used to convert part of the energy formed during the process of combustion to radiant heat. These baffles may also serve to direct the flow of products of combustion along the heat-absorbing surface.

Gas-designed furnaces and boilers approved by the American Gas Association are certified for operation at the rating shown on the nameplate. Considerations relating to safety, performance, and service life, require that such appliances be adjusted at inputs that do not exceed the nameplate input rating. These appliances normally draw in from 20 to 50 percent excess air, depending on the type and general design. As has been indicated, some excess air is necessary to insure complete combustion at all times and also to provide a reasonable degree of flexibility in performance.

Various types of appliances used for gas space heating purposes are described in Chapter 34.

FUNDAMENTAL PRINCIPLES OF COMBUSTION

Regardless of the type of fuel under consideration, its combustion results in the production of gaseous products. Many kinds of solid fuels contain minerals which cannot be burned and are therefore left as a residue commonly called ash. Moreover, unless sufficiently high temperatures are employed and an ample supply of oxygen properly distributed is present, the combustible constituents of solid, liquid, and even gaseous fuels cannot be completely burned. Incomplete or partial combustion of all fuels produces toxic gases, such as carbon monoxide, with smaller quantities of aldehydes,



Flue Gas Temperature Shown. Loss is Based on 65 F Room Temperature

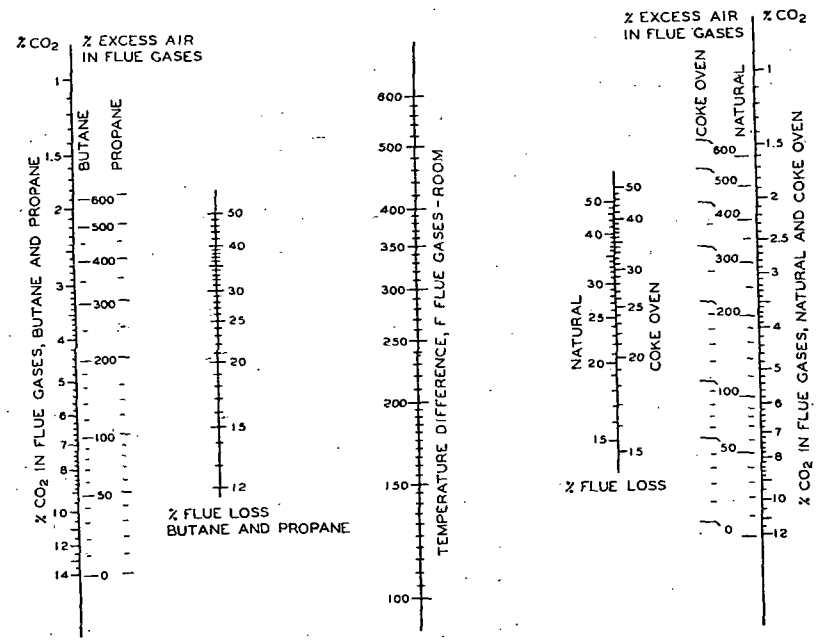
Fig. 4 .... Flue-Gas Losses with Various Fuels[3]

ketones, and other hydroxylated hydrocarbon compounds. This fact indicates that, while combustion processes involving common types of fuel may be regulated by experienced operators to produce the most efficient results, normal combustion processes can be so unbalanced as to create hazards unless both design and operation are planned with a knowledge of the fundamental principles of combustion.

Combustion may be defined as the chemical combination of a substance with oxygen resulting in the evolution of heat, and usually some light. The rate of combustion depends upon the rate of reaction of the substance with oxygen, upon the rate at which oxygen is supplied, and upon the temperature obtained due to surrounding conditions. This is combustion in its simplest form. All solid, as well as liquid and even gaseous fuels generally contain several combustible elements in combination with others which, depending on their nature, affect oxygen requirements and thus govern the combustion process. For a continuous reaction, as in heating processes, it is necessary to establish an effective balance between rates of removing heat and of supplying fuel and air or oxygen to keep the reaction going. In establishing such a balance, consideration must be given to the removal or venting of products of combustion, so that the entire process is one of flow wherein draft conditions in the combustion space are important.

Complete combustion is obtained when all combustible elements in a fuel are oxidized by all of the oxygen with which they will combine. All oxygen or air supplied is generally not utilized, and this excess portion is commonly referred to as *excess oxygen* or *excess air*. Excess air is usually expressed as a percentage of the air required for perfect combustion.

Perfect combustion results when the exact amount of oxygen required for complete combustion of all elements of a fuel is supplied and utilized. The percentage of carbon dioxide contained in the products of combustion from such a reaction is obviously the maximum attainable and is referred to as the *ultimate CO_2 or maximum theoretical percentage of carbon dioxide*. This condition of perfect combustion, without having excess air or oxygen left from the reaction, is seldom, if ever, realized in practice. Most types of heating equipment must be sufficiently flexible in performance to provide complete combustion with not only variations in the quality of a fuel but also changes in the rate at which it is supplied. This situation makes it advisable, from a practical standpoint, to insure *complete com-*



Adapted from American Gas Association Laboratories Flue Loss Charts.

Fig. 5 .... Alignment Chart for Calculation of Flue Losses for Butane, Propane, Coke Oven, and Natural Gases

Table 7 .... General Data of Combustible Elements and Compounds

Substance	Molecular Symbol	Chemical Reaction of Combustion	Ignition Temperature[a] F Deg	Calorific Value: Btu per Lb Gross	Calorific Value: Btu per Lb Net	Calorific Value: Btu per Cu Ft[b] Gross	Theoretical Oxygen and Air Requirements: Lb per Lb O_2	Lb per Lb Air	Cu Ft per Cu Ft O_2	Cu Ft per Cu Ft Air
Carbon (to *CO*)	—	$2C + O_2 = 2CO$	—	3950	—	—	1.33	5.76	—	—
Carbon (to CO_2)	—	$2C + 2O_2 = 2CO_2$	—	14093	—	—	2.66	11.53	—	—
Sulfur (to SO_2)	—	$S + O_2 = SO_2$	—	3983	—	—	1.00	4.29	—	—
Sulfur (to SO_3)	—	$2S + 3O_2 = 2SO_3$	—	5940	—	—	1.50	6.43	—	—
Carbon Monoxide	*CO*	$2CO + O_2 = 2CO_2$	1166–1319	4347	—	321.8	0.57	2.47	0.5	2.39
Methane	CH_4	$CH_4 + 2O_2 = CO_2 + 2H_2O$	1260–1380	23879	21520	1013.2	3.99	17.27	2.0	9.53
Acetylene	C_2H_2	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$	763–824	21500	20776	1499	3.07	13.30	2.5	11.91
Ethylene	C_2H_4	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	986–1123	21644	20295	1613.8	3.42	14.81	3.0	14.29
Ethane	C_2H_6	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	990–1120	22320	20432	1792	3.73	16.12	3.5	16.68
Hydrogen	H_2	$2H_2 + O_2 = 2H_2O$	1063–1166	60958[a]	51571[a]	325	7.94	34.34	0.5	2.39
Hydrogen Sulfide	H_2S	$2H_2S + 3O_2 = 2H_2O + 2SO_2$	599–608	7100	6545	647	1.41	6.10	1.5	7.15
Propane	C_3H_8	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	950–1080	21661	19944	2590	3.63	15.70	5.0	23.82
n-Butane	C_4H_{10}	$2C_4H_{10} + 13O_2 = 8CO_2 + 10H_2O$	890–1020	21308	19680	3370	3.58	15.49	6.5	30.97
Commercial Propane	—	—	920–1020	21560	19865	2520	3.60	15.58	4.9	23.4
Commercial Butane	—	—	900–1000	21180	19591	3260	3.54	15.3	6.3	30.0

[a] Values in table taken chiefly from page 51 of *Fuel Flue Gases* published by American Gas Association.
[b] Gas measured at 60 F and 30 in. Hg. [c] Value from National Bureau of Standards.

bustion but not *perfect combustion* in the sense expressed above. To attempt to do so would undoubtedly result eventually in unsatisfactory performance especially from a safety standpoint. Consequently, common types of heating equipment are usually designed, installed, and adjusted to operate with some excess air. The exact percentage of such air depends on the type of fuel being utilized, as well as anticipated variations in its quantity and quality. Despite these practical limitations, however, it should not be inferred that common types of fuels cannot be utilized economically. Reference to flue loss charts such as Figs. 4 and 5 for gas burning equipment and to the air requirements discussion in Chapter 34 shows that reasonable quantities of excess air can be used without appreciable reductions in operating efficiencies.

Oxygen combines with the combustible elements and compounds of any fuel in accordance with fixed laws. The reactions and resultant products of perfect combustion of common fuel constituents are shown in Table 7. All of the oxygen required for combustion is normally obtained from the surrounding air, which is a mechanical mixture of nitrogen and oxygen with small amounts of carbon dioxide, water vapor and inert gases. For practical combustion calculations, air is considered to consist of 20.9 percent oxygen and 79.1 percent nitrogen by volume, or 23.15 percent oxygen and 76.85 percent nitrogen by weight. The nitrogen, being inert, passes through the reaction without change. Table 7 gives the air quantities corresponding to the oxygen required for perfect combustion.

Air supplied to the combustion reaction is in most instances introduced in two ways. *Primary air* is introduced through or with the fuel, and *secondary air* is supplied to the flames issuing from the fuel.

Incomplete combustion is obtained when any of the combustible elements are not completely oxidized in the combustion reaction. This condition not only represents inefficient use of the fuel but also presents a hazard because carbon monoxide is usually one of the products of incomplete combustion. For example, a hydrocarbon may not oxidize completely to carbon dioxide and water, as indicated in Table 7, but may also form alcohols, ketones, aldehydes, or carbon monoxide depending on where and how the reaction is interrupted. Too low a temperature (such as may be caused by flame impingement on a cold surface), a poor oxygen supply to the flames (due to insufficient or poorly located air supply, or smothering by products of combustion not properly vented), or insufficient mixing of the air and fuel, are the primary causes of incomplete combustion.

Heat of Combustion

As previously stated, the process of combustion results in the evolution of heat. The heat generated by the complete combustion of a unit of fuel is constant for a given combination of combustible elements and compounds, and is known as the *heat of combustion*, *calorific value*, or *heating value* of the fuel. The heat of combustion of the several substances found in the more common fuels is given in Table 7.

The calorific value of a fuel may be determined either by direct measurement of the heat evolved during combustion in a calorimeter, or it may be computed from the ultimate analysis and the heat of combustion of the several chemical elements in the fuel. When the heating value of a fuel is determined in a calorimeter, the water vapor is condensed and the latent heat of vaporization is included in the heating value of the fuel. The heating value so determined is termed the *gross* or *higher* heating value, and this is what is ordinarily meant when the heating value of a fuel is specified. In burning the fuel, however, since the products of combustion are not cooled to the dew-point and the higher heating value cannot be utilized.

When combustion is complete, the carbon in the fuel unites with oxygen to form carbon dioxide, CO_2, the hydrogen unites with oxygen to form water vapor, H_2O, and the nitrogen, being inert, passes through the reaction without change. When combustion is incomplete, some of the carbon may unite with oxygen to form carbon monoxide, *CO*, and some of the hydrogen and hydrocarbon gases may not be

burned at all. When carbon monoxide or other combustible gases are present in the flue gases, there is a loss of heat produced per unit of fuel consumed, and consequently, a lower combustion efficiency is obtained. Incomplete combustion may result from any or all of the following three conditions: (1) inadequate air supply, (2) insufficient mixing of air and gases, and (3) a temperature too low to produce ignition or maintain combustion.

AIR REQUIRED FOR PERFECT COMBUSTION

Air requirements for combustion of solid and liquid fuels are ordinarily expressed in pounds. On the other hand similar requirements for gaseous fuels are usually stated on a cubic foot basis. For solid and gaseous fuels this method of treatment corresponds to the standards of measurement commonly employed by these two industries.

The weight of air required for perfect combustion per pound of solid or liquid fuel may be calculated after substitution of the proper percentages by weight of the various elements obtained from an ultimate analysis of the fuel in Equation 2. For gaseous fuels see Equation 3 for volume of air required.

Solid or Liquid Fuels:

Pounds air required per pound fuel

$$= \frac{34.56}{100}\left[\frac{C}{3} + \left(H - \frac{O}{8}\right) + \frac{S}{8}\right] \quad (2)$$

For Gaseous Fuels:

Cubic feet air required per cubic foot gas

$$= 2.39\,(CO + H_2) + 9.53\,CH_4 + 16.68\,C_2H_6 + 23.82\,C_3H_8 + 30.97\,C_4H_{10} + 11.91\,C_2H_2 + 14.29\,C_2H_4 + 7.15\,H_2S - 4.78\,O_2 + 30.47 \text{ Illuminants} \quad (3)$$

Gaseous fuels may contain a wide variety of components classified as *illuminants*, which are not separated by the usual methods of gas analysis. The principal illuminants in addition to ethylene (C_2H_4) and acetylene (C_2H_2) which are included in Equation 3, with the air required per cubic foot of gas are: propylene, 21.44; butylene, 28.58; pentene, 35.73; benzene, 35.73; toluene, 42.88; and xylene, 50.02. Since toluene and xylene are normally scrubbed from the gas before distribution, they may be disregarded in computing air required for a fuel gas. An approximate value of 30.47 (as shown in Equation 3) may, therefore, be employed. If ethylene and acetylene are included as *illuminants*, it is suggested that the value 19.65 be used.

If it is desired to make the gas calculations on a weight basis the equation is expressed as follows:

Pounds air required per pound fuel

$$= 2.47\,CO + 34.34\,H_2 + 17.27\,CH_4 + 16.12\,C_2H_6 + 15.70\,C_3H_8 + 15.49\,C_4H_{10} + 13.30\,C_2H_2 + 14.81\,C_2H_4 + 6.10\,H_2S - 4.32\,O_2 \quad (4)$$

Where approximate results only are desired, values appearing in Table 8 may be substituted for Equations 2, 3, and 4; or the air required for perfect combustion may be estimated by assuming that 0.9 cu ft air is required per 100 Btu of fuel.

Where extreme precision is involved it is suggested that the reader refer to scientific literature published on the subject of combustion and related topics by the various industries concerned. If approximate values for theoretical air requirements suffice, or if complete information on the fuel is not available, the following values should also be found helpful:

1. *Solid Fuels* (Pounds air per pound fuel): Anthracite, 9.6; Semi-Bituminous, 11.2; Bituminous 10.3; Lignite 6.2; and Coke 11.2.

2. *Fuel Oil* (Pounds air per gallon): Commercial Standard No. 1, 102.6; No. 2, 105.5; No. 5, 112; No. 6, 114.2.

3. *Gaseous Fuels* (Cubic feet of air per cubic foot): Natural, 10.0; Mixed Natural and Manufactured, 8.0; Manufactured, 4.7, Propane, 23.8, Butane, 31.0.

COMBUSTION EFFICIENCY FROM THE FLUE-GAS ANALYSIS

Excess Air

A commonly employed index of efficiency of combustion is the relation existing between the amount of air theoretically required for perfect combustion and the amount of air actually supplied. Since the difference between air supplied for combustion and theoretical air required is characterized as excess air, its percentage may be calculated by use of the following equation,

Percent excess air

$$= \left(\frac{\text{Air supplied} - \text{Theoretical air}}{\text{Theoretical air}}\right) \times 100 \quad (5)$$

The amount of dry air supplied per pound of fuel burned may be obtained from Equation 6 which has reasonable precision for most solid and liquid fuels. Values for CO_2, *CO* and N_2 are percentages by volume from the flue-gas analysis, and *C* is the weight of carbon burned per pound of fuel, corrected for carbon in the ash.

Pounds dry air supplied per pound of fuel

$$= \frac{3.04N_2}{(CO_2 + CO)} \times C \quad (6)$$

Because excess air calculations are almost invariably made from Orsat analysis results, and theoretical air requirements

Table 8 .... Approximate Air Requirements for Theoretically Perfect Combustion of Fuels[a]

Type of Fuel	Air Required for Perfect Combustion: Lb per Lb Fuel	Air Required for Perfect Combustion: Cu Ft per Unit[b] Fuel	Approximate Precision, %	Exceptions
Solid	Btu per lb × 0.00073	Btu per lb × 0.0097	3	Fuels containing more than 30% water
Liquid	Btu per lb × 0.00071	Btu per lb × 0.0094	3	Results low for gasoline and kerosine
Gas	Btu per lb × 0.00067	Btu per cu ft × 0.0089	5	Gases of 300 Btu per cu ft or less

[a] Values in table taken from page 378 of *Gaseous Fuels*, 1948, published by *American Gas Association*.
[b] Units for solid and liquid fuels in pounds, for gas in cubic feet.

are not always known, another convenient method of expressing the relation of Equation 5 is as follows:

$$\text{Percent excess air} = \frac{100(O_2 - CO/2)}{N_2 \times 0.264 - (O_2 - CO/2)} \quad (7)$$

As measurement standards for gaseous fuels are almost universally expressed in cubic feet, Equation 8 may be employed for computing excess air on a percentage basis for gases.[7]

$$\text{Percent excess air} = \frac{(U - CO_2)}{CO_2} \times 100\,\frac{P}{A} \quad (8)$$

where

U = ultimate carbon dioxide, percent of flue gases resulting from perfect combustion.
CO_2 = carbon dioxide content of flue gases, percent.
P = dry products from perfect combustion, cubic feet per cubic foot of gas burned.
A = air theoretically required for complete combustion, cubic feet per cubic foot of gas burned.

As the ratio of P/A is approximately 0.9 for most city gases, a value of 90 may be substituted for 100(P/A) in Equation 8 for rough calculation.

Carbon-hydrogen ratios of different fuels vary considerably, hence the maximum or ultimate CO_2 attainable also varies. Where they are unknown, theoretical maximum CO_2 values may be calculated from a flue-gas analysis by use of Equation 9.

Maximum theoretical % CO_2

$$= \frac{\%\ CO_2 \text{ in flue gas sample} \times 100}{100 - \left(\dfrac{O_2 \text{ in same sample}}{0.21}\right)} \quad (9)$$

Approximate maximum CO_2 values for perfect combustion of several common types of fuel are shown in Table 9 together with values of CO_2 that will be attained with different amounts of excess air. Desirable values to be attained in practice depend upon the fuel, the method of firing, and other considerations. In general, fuels burned in suspension, such as gas, oil, and pulverized coal, can be burned with a lower amount of excess air than fuels burned on grates.

Table 9 . . . . Approximate Maximum Theoretical CO_2 Values, and CO_2 Values for Various Fuels with Different Percentages of Excess Air

Type of Fuel	Maximum Theoretical or Ultimate Percent CO_2	Percent CO_2 at Given Excess Air Values		
		20%	40%	60%
Coke	21.0	17.5	15.0	13.0
Anthracite	20.2	16.8	14.4	12.6
Bituminous Coal	18.2	15.1	12.9	11.3
No. 1 and 2 Fuel Oil	15.0	12.3	10.5	9.1
No. 6 Fuel Oil	16.5	13.6	11.6	10.1
Natural Gas	12.1	9.9	8.4	7.3
Carburetted Water Gas	17.2	14.2	12.1	10.6
Coke Oven Gas	11.2	9.2	7.8	6.8
Mixed Gas (Natural and Carburetted Water Gas)	15.3	12.5	10.5	9.1
Propane Gas (Commercial)	13.9	11.4	9.6	8.4
Butane Gas (Commercial)	14.1	11.6	9.8	8.5

To produce heat efficiently by burning any common fuel, a number of basic requirements must be met: (1) adequate heat absorbing surface of proper shape and construction is necessary in the appliance, (2) heat transfer surfaces must be clean, (3) a minimum amount of excess air must be present, (4) air employed for combustion and combustible gases must be properly mixed, and (5) flue-gas losses must be reduced to a safe minimum.

If insufficient heating surface is employed, or if heat transfer surfaces are covered with soot, ash, or scale, flue losses will generally be excessive due to the large amount of sensible heat escaping to the chimney. Too much excess air dilutes flue gases excessively and increases sensible flue gas loss. On the other hand, a deficiency of excess air will in all probability cause incomplete combustion, and consequent escape of some unburned combustible gases from the appliance. Highest combustion efficiency is obtained when sufficient excess air is supplied and properly mixed with the combustible gases. Even with appliances connected to an effective flue every reasonable precaution should be taken to insure as complete combustion as possible at all times. This is of vital importance in unvented equipment such as a gas space heater, for example. It is considered good practice to supply from 20 to 50 percent excess air, the exact amount depending on the kind of fuel burned and especially on the type of equipment in which it is utilized.

Flue Gas Analysis

As radiation, convection, and conduction losses from common types of central heating appliances are largely employed in heating occupied spaces, flue-gas losses logically become the item of principal concern. Another reason for their importance is the fact that they are usually larger than all other heat losses combined. These important considerations and others indicate not only the desirability but absolute necessity of a reasonably accurate method for determining flue losses even if only approximate operating efficiencies are to be computed.

Customary procedure in arriving at flue-gas losses is to make an analysis of a representative sample of flue gases and to measure their temperature. This information will enable the observer to compute the amount of flue gases produced, the excess air, the actual quantity of air supplied for combustion, and the flue losses. While the analysis of flue gases has been well described in several governmental and other scientific publications, the subject is of such importance that it warrants brief repetition here. Carbon dioxide and oxygen content are of principal interest in determining flue losses. Either or both of these values may be employed in such calculations. While the former constituent is probably most commonly employed, special considerations may make the latter of greater interest. Fortunately, both can be determined readily by use of an Orsat gas analysis apparatus, a device of reasonably simple construction and design. For field testing and burner adjustment, simple portable devices are available for determining carbon dioxide only. See Chapter 44 for details regarding the operation of the Orsat apparatus.

The weight of dry flue gas per pound of fuel burned is

widely used in combustion loss calculations. For solid fuels this item may be determined by application of Equation 10.

Pounds dry flue gas per pound fuel

$$= \frac{11CO_2 + 8O_2 + 7(CO + N_2)}{3(CO_2 + CO)} \times C \quad (10)$$

Values for CO_2, O_2, CO, and N_2 are percentages by volume from the flue gas analysis, and C is the weight of carbon burned per pound of fuel, corrected for carbon in the ash.

Total dry gas volumes of flue gases resulting from the combustion of one cubic foot of gaseous fuels for various percentages of CO_2 may be determined by application of Equation 11:

Cubic feet dry flue gases per cubic foot fuel gas

$$= \frac{\text{Cubic feet } CO_2 \text{ produced per cubic foot of gas burned} \times 100}{\text{Percent } CO_2 \text{ by analysis}} \quad (11)$$

After obtaining the quantity of flue gases from Equations 10 or 11, the excess air quantity may be determined by subtracting the quantity of dry flue gases which would result from perfect combustion. Computations of flue losses are described in the next section on Heat Balance.

Application of the preceding equations and tables are illustrated by *Examples 1* and *2*.

Example 1: The analysis of the flue gases resulting from the burning of a natural gas is 10.0 percent CO_2, 3.1 percent O_2, and 86.9 percent N_2 by volume. The analysis of the fuel is 90 percent CH_4, 5 percent N_2, and 5 percent C_2H_6 by volume. Find U the maximum theoretical percent CO_2 and the percent excess air.

Solution: From Equation 9:

$$U = \frac{(10.0)(100)}{100 - \left(\dfrac{3.1}{0.21}\right)} = 11.8\%\ CO_2$$

From Equation 8,

$$\text{Percent Excess Air} = \frac{(11.8 - 10.0) \times 90}{10} = 16.2$$

Example 2: For the analyses in *Example* 1 find, per cubic foot of fuel gas, the cubic feet of dry air required for combustion, the cubic feet of each constituent in the flue gases, and the total volume of dry and wet flue gases.

Solution: From Equation 3 (or Table 7) the volume of dry air required for combustion is: $(9.53)\ (CH_4) + (16.68)(C_2H_6) = 9.53 \times 0.90 + 16.68 \times 0.05 = 9.41$ cu ft/cu ft gas.

From Table 7, the constituents per cubic foot of gas are:

Nitrogen, N_2:

From methane = $(0.9\ CH_4)(9.53 - 2.0) = 6.78$
From ethane = $(0.05\ C_2H_6)(16.68 - 3.5) = 0.66$
Nitrogen in fuel = 0.05
Nitrogen in excess air = $0.791 \times 0.162 \times 9.41 = 1.20$
Total Nitrogen 8.69 cu ft

Oxygen, O_2:
Oxygen in excess air = $0.209 \times 0.162 \times 9.41 = 0.32$ cu ft

Carbon dioxide, CO_2:
From methane = $(0.9CH_4)(1.0) = 0.90$
From ethane = $(0.05C_2H_6)(4.0/2.0) = 0.10$
Total Carbon Dioxide 1.00 cu ft

Water vapor, H_2O (does not appear in Orsat analysis):
$(0.9\ CH_4)(2.0) = 1.8$
$(0.05\ C_2H_6)\ (6.0/2.0) = 0.15$
Total water vapor = 1.95 cu ft

Total volume of dry gas per cubic foot of gas:

$$8.69 + 0.32 + 1.00 = 10.01 \text{ cu ft}$$

Total volume of wet gases per cubic foot of gas (neglecting water vapor in combustion air):

$$10.01 + 1.95 = 11.96 \text{ cu ft}$$

The cubic feet of dry flue gas per cubic foot of fuel gas may also be computed from Equation 11 as follows:

$$\frac{(1.00)(100)}{10.0} = 10.0 \text{ cu ft}$$

HEAT BALANCE

The usual practice in analyzing the performance of heating appliances is to make an accounting, insofar as possible, of the disposition of all heat units available in the quantity of fuel burned. This accounting is called a *heat balance*. Various components of this balance are generally expressed in terms of Btu per pound of fuel burned, or as a percentage of its calorific value. Components of special interest are listed as items 1 to 7 inclusive.

1. Useful heat transferred to heating medium, usually computed by determining the rate of flow of the heating fluid through the heating device, and the change in enthalpy of the fluid (heat added) between the inlet and outlet.

2. Heat loss in the dry chimney gases.

$$h_1 = w_g c_p\,(t_g - t_a) \quad (12)$$

3. Heat loss in water vapor formed by the combustion of hydrogen.

$$h_2 = \frac{9H_2}{100}\,(1089 - t_a + 0.455t_g) \quad (13)$$

4. Heat loss in water vapor in the air supplied for combustion.

$$h_3 = 0.455\ M\ w_a\,(t_g - t_a) \quad (14)$$

5. Heat loss from incomplete combustion.

$$h_4 = 10143C\left(\frac{CO}{CO_2 + CO}\right) \quad (15)$$

6. Heat loss from unburned carbon in the ash or refuse.

$$h_5 = 14600^a\left(\frac{C_a}{100} - C\right) \quad (16)$$

7. Radiation and all other unaccounted for losses.

Radiation and convection losses from a heating appliance are not usually determined by direct measurement. For this reason they, together with any other losses not measured, are determined by subtracting the total of items 1 to 6 from the heat of combustion of the fuel. If the heating appliance is located within the heated space, however, radiation and convection losses may be considered as useful heat rather than lost heat. They may, therefore, be omitted from calculations of heat losses, or added to item 1. If there is CO in the flue gases, small amounts of unburned hydrogen and hydrocarbons will probably also be present. The small losses

due to incomplete combustion of these latter gases would also be included in item 7.

Symbols used in Equations 12 to 16 inclusive are:

h_1 = heat loss in the dry chimney gases, Btu per pound of fuel.
h_2 = heat loss in water vapor from combustion of hydrogen, Btu per pound of fuel.
h_3 = heat loss in water vapor in combustion air, Btu per pound of fuel.
h_4 = heat loss from incomplete combustion of carbon, Btu per pound of fuel.
h_5 = heat loss from unburned carbon in the ash, Btu per pound of fuel.
w_g = weight of dry flue gas per pound of fuel (from Equation 10), pounds.
c_p = mean specific heat of flue gases at constant pressure (c_p ranges from 0.242 to 0.254 for flue gas temperatures from 300 F to 1000 F),[3] Btu per pound.
t_g = temperature of flue gases at exit of heating device, Fahrenheit.
t_a = temperature of combustion air, Fahrenheit.
H_2 = percentage of hydrogen in fuel by weight from ultimate analysis of fuel burned.
1091.8 = enthalpy of saturated water vapor at a temperature of 70 F, Btu per pound.
M = humidity ratio of combustion air, pounds of water vapor per pound of dry air.
w_a = weight of combustion air per pound of fuel used, pounds, from Equations 2, 4, 5, 6, 7 and 8.
CO, CO_2 = percentages of CO, CO_2 in flue gases by volume.
C = weight of carbon burned per pound of fuel corrected for carbon in ash, pounds.

$$C = \frac{WC_a - W_aC_a}{100W} \qquad (17)$$

where

C_a = percentage of carbon in the fuel by weight from the ultimate analysis.
W_a = weight of ash and refuse, pounds.
C_a = percent of combustible in ash by weight (combustible in ash is usually considered to be carbon).
W = weight of fuel used, pounds.

Flue-gas losses for solid and liquid fuels, listed as items 2, 3 and 4 of the heat balance, may be determined with sufficient precision for most purposes from curves shown in Fig. 4,[3] if CO_2 content and temperature of flue gases are known. Values of the losses plotted for fuel oil were computed from the ultimate analysis of a typical fuel oil used in domestic burners, while those presented for the several ranks of coal were computed from the typical ultimate analyses shown in Table 1. The curves for medium-volatile bituminous coal may be used for high-volatile bituminous coal with negligible error.

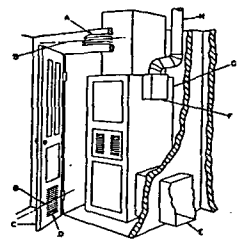
Utilization of gaseous fuels, for numerous reasons, is generally a more simple process than is the case with either solid or liquid fuels. Accordingly, the determination of a practical heat balance is also a more simple procedure in that items 5 and 6 do not generally apply to gas installations. A series of typical alignment charts has been combined in Fig. 5 for use in determining flue losses of items 2, 3 and 4 from common types of gas burning appliances. To determine flue losses place a straight edge extending from the corrected temperature reading to the percent CO_2 recorded. Percent flue loss is indicated where the straight edge intersects the flue loss column. The operating efficiency of a gas appliance can then be computed with sufficient precision by application of Equation 18.

Table 10 .... Average Flue Gas Dew Point for Various Fuels[4]

Type of Fuel	Average Dew Point Temperature, F
Anthracite	68
Semi-Bituminous Coal	84
Bituminous Coal	93
Oil	111
Natural Gas	127
Manufactured Gas	137
Propane Gas (2500 Btu/cu ft)	119
Butane Gas (3200 Btu/cu ft)	124
Butane-Air Gas Mixture (535 Btu/cu ft)	121

Percent Combustion Efficiency =

$$\frac{\left(\begin{matrix}\text{Gross Btu of fuel}\\ \text{gas per cubic foot}\end{matrix}\right) - \left(\begin{matrix}\text{total flue losses per}\\ \text{cubic foot fuel gas}\end{matrix}\right)}{\text{Gross Btu of fuel gas per cubic foot}} \times 100 \qquad (18)$$



A. Ventilating air outlet register for appliance room, 1 sq in. free area for each 1000 Btu per hr appliance input located above relief opening of draft hood. Register must not be blocked by drapes or other furnishings.

B. Both registers must either face same large ventilated interior space or extend to such space by means of ducts. Vertical distance C/L to C/L of registers should be not less than 3½ ft.

C. Suggest room access door be not less than 6 ft high by a width sufficient to provide for installation or removal of appliance. At least 3 ft horizontal clearance should be provided in front of appliance when closet door is open, or 18 in. when door is closed.

D. Combustion and ventilation air inlet register for appliance room, 1 sq in. free area for each 1000 Btu per hr appliance input, located at or below combustion air inlet to appliance. Register must not be blocked by drapes or other furnishings.

E. Air circulated by appliance must be handled by ducts which are sealed to appliance casing and are entirely separate from means provided for supplying combustion and ventilation air.

F. Spacing between draft hood and wall at least 6 in. (unless approved for closer spacing). If flue products may be directed toward wall, 12 in. spacing recommended.

G. No part of appliance casing closer than 6 in. to wall (unless approved for closer spacing).

H. Flue should terminate above peak of roof and above nearby walls to assure satisfactory flue performance.

* A value of 14600 applies in calculating ash pit loss; in calculating heat of formation of carbon compounds use 14093 Btu per lb.

Fig. 6 .... Illustration Showing Air Openings Necessary to Supply Air for Combustion When Appliance is Installed in Confined Space

Table 11 .... Permanent Openings for Air Supply to Spaces Containing Fuel-Burning Equipment

	Fuel-Burning Equipment	Free Area of Ventilating Opening*	
		Air Inlet	Air Outlet
Gross output capacity exceeding 250,000 Btu per hr	Equipment in a large enclosure having adequate air infiltration	None required	None required
	Equipment in a large enclosure having unusually tight construction	Total free area communicating with outdoor air not less than ½ sq in. per (1000 Btu) (hr) furnace input	Total free area communicating with outdoor air not less than ½ sq in. per (1000 Btu) (hr) furnace input
	Equipment in a small enclosure in a building having adequate air infiltration	Opening in party wall or door, (above draft hood) with free area of not less than 1 sq in. per (1000 Btu) (hr) furnace input	Opening in a party wall or door, (below combustion air inlet to appliance) with free area of not less than 1 sq in. per (1000 Btu) (hr) furnace input
	Equipment in a small enclosure in a building of unusually tight construction	Grilled opening connecting directly with outdoors, through duct if necessary; free area not less than ½ sq in. per (1000 Btu) (hr) furnace input	Grilled opening connecting directly with outdoor air, through duct if necessary. Area and location same as above.
Gross output capacity not exceeding 250,000 Btu per hr	Equipment in an enclosure having adequate air infiltration	Total free area equivalent to not less than the combined area of the flue outlets of all fuel-burning equipment in the space	None required
	Equipment in an enclosure having unusually tight construction	Total free area equivalent to not less than twice the combined area of the flue outlets, of all fuel-burning equipment in the space	None required

* Operation of exhaust fans, kitchen ventilation systems, fire places, etc. may require special consideration. For residential closet installations see Fig. 6.

Reference to Table 9 will show that ultimate CO_2 percentage values of fuel gases vary. While personal errors involved in CO_2, temperature, and chart determinations, would doubtless more than offset any inaccuracies due to universal use of the alignment charts shown, precise laboratory work may require a more exact method. For more complete information the reader is referred to *combustion*, 3rd Edition, and *Gaseous Fuels*, (published by American Gas Association) and particularly to tables covering various properties of different commercial gases included in these publications.

CONDENSATION AND CORROSION

Sulfur dioxide or sulfur trioxide, formed by the combustion of sulfur in fuels, are the principal corroding substances in flue gases. They become active whenever sufficient moisture is present for the formation of sulfurous or sulfuric acid,[18] and they lower the dew points of flue gases appreciably. Therefore, unless heating equipment is designed for operation at flue-gas temperatures below the dew point, which is seldom the case, it is always advisable to maintain temperatures above this value in all parts of the appliance. Excessive spot temperatures in the combustion chamber or elsewhere, on the other hand, are also destructive in that they may result in rapid oxidation of ordinary heating surfaces. *American Standard Requirements* for gas furnaces, floor furnaces, and recessed heaters, for example, specify that minimum spot heating surface temperatures during normal operation must neither fall below 178 F (50 F above average dew point) nor exceed 830 F to 1230 F on any portion of the heating surface, depending on the type and thickness of the metal. In any event it is usually desirable to maintain flue-gas temperatures within the limits indicated not only throughout the appliance, but in its connecting vent, flue, or chimney as well. Otherwise, excessive condensation and corrosion problems, with resultant customer dissatisfaction, will in all probability be the result. Average dew-point temperatures of flue gases resulting from the combustion of various fuels, when burned with the amount of excess air normally supplied to insure complete combustion, are shown in Table 10.

SOOT

The deposit of soot on the flue surfaces of a boiler or heater acts as an insulating layer over the surface, and reduces the heat transmission to the water or air. The *Bureau of Mines Report of Investigations* No. 3272[19] shows that the loss of seasonal efficiency is not so great as has been believed, and usually is not over 6 percent because the greater part of the heat is transmitted through the combustion chamber surfaces. The *Bureau of Standards Report* BMS 54[20] points out that, although the decrease in efficiency of an oil fired boiler, due to soot deposits, is relatively small, the attendant increase in stack temperature may be considerable.

The soot accumulation clogs the flues, reduces the draft, and may prevent proper combustion. Soot can probably be most effectively removed by a jet of compressed air, by means of a brush, or a vacuum cleaner. However, it has been found that copper chloride, lead chloride, tin chloride, zinc chloride, common salt and some other salts are partially effective in removing soot from furnaces and boilers when properly used.[21] A discussion of instruments and methods of evaluating smoke will be found in Chapter 44.

AIR SUPPLY TO FUEL-BURNING EQUIPMENT

All rooms or spaces containing boilers, furnaces, water heaters, or any other fuel-burning equipment must be provided with a constant supply of combustion air at adequate static pressure to insure proper combustion in the fuel burners. Additional air is required to replace air entering chimneys through draft hoods and barometric draft dampers, and also to provide ventilation in closely confined boiler and furnace rooms.

Experience has shown that the ventilating air openings

required in heater rooms depend not only on the size of the heating plant but also on the type of fuel being burned, the relative size of the equipment room, the tightness of building construction, and the operation of exhaust fans or other devices which affect the static pressure in a building.

Because of the number of variables involved in establishing air requirements, and the variations in building construction and arrangement, there is no universally accepted rule for providing ventilating openings. However, the importance of providing proper ventilation should not be underestimated, and a failure to do so may result in erratic or even dangerous operating conditions for equipment.

Rules for providing air supply openings are found in technical standards,[22, 23] in state and municipal building codes and in service and installation bulletins published by various manufacturers of fuel-burning appliances. Table 11 is a compilation of reasonably consistent data taken from a number of these sources, and should be used only as an aid to independent judgment for any particular installation.

Ventilating air requirements are particularly critical for residential type appliances in closely confined rooms. Fig. 6 illustrates minimum requirements for this type of installation.

REFERENCES

[1] *Five Hundred Tests of Various Coals in Householding Boilers* (*U. S. Bureau of Mines Bulletin* No. 276).

[2] A. P. Kratz, S. Konzo, and D. W. Thompson: *Combustion Efficiencies as Related to Performance of Domestic Heating Plants* (University of Illinois, *Engineering Experiment Station Circular* No. 44).

[3] *Quality of Anthracite as Prepared at Breakers* (*U. S. Bureau of Mines Report of Investigation*, 1935 R. I. 3283).

[4] *Hand Firing Soft Coal Under Power Plant Boilers* (*U. S. Bureau of Mines Technical Paper* No. 80).

[5] *ASTM Test Designation* D287-39.

[6] *ASTM Test Designation* D611-43T.

[7] W. A. Sullivan and E. B. Glendenning: Tomorrow's fuel oil (*Fueloil and Oil Heat*, Vol. 4, May 1945, p. 36).

[8] The correlation of cetane number with other physical properties of diesel fuels (*Journal of the Institute of Petroleum*, Vol. 30, 1944, p. 193).

[9] K. M. Watson, E. F. Nelson, and G. B. Murphy: Characterization of petroleum fractions (*Industrial and Engineering Chemistry*, Vol. 27, December 1935, p. 1464).

[10] J. C. Reid and A. B. Hersberger: Burning index for distillates (*Fueloil and Oil Heat*, Vol. 5, January 1947, p. 90).

[11] S. P. Cauley and E. B. Delgass: Carbon hydrogen ratio of catalytically cracked distillate fuel oils (*Oil and Gas Journal*, Vol. 45, July 27, 1946, p. 166).

[12] D. W. Locklin and G. V. Parmalee: Rating of fuel oils by a test unit (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 139).

[13] *Gaseous Fuels* (American Gas Association, 1948, p. 32).

[14] V. J. Altieri: *Gas Analysis and Testing of Gaseous Materials* (American Gas Association, 1945, 1st ed.).

[15] *Tentative Methods of Test for Specific Gravity of Gaseous Fuels* (American Society for Testing Materials, *ASTM Designation*: 1070-49).

[16] *Standard Method of Test for Calorific Value of Gaseous Fuels by the Water-flow Calorimeter* (American Society for Testing Materials, *ASTM Designation*: 900-48).

[17] *Domestic Gas Range Research* (*American Gas Association Laboratories Bulletin* 7, p. 64).

[18] W. R. Morgan: *Condensation of Moisture in Flues* (University of Illinois, *Engineering Experiment Station Circular* No. 22).

[19] *Effect of Soot on Heat Transmission in Boilers* (*U. S. Bureau of Mines Report of Investigation* No. 3272).

[20] *Effect of Soot on the Rating of an Oil-fired Heating Boiler* (*National Bureau of Standards Report* BMS 54).

[21] P. Nicholls and C. W. Staples: *Removal of Soot from Furnaces and Flues by the Use of Salts and Compounds* (*U. S. Bureau of Mines Bulletin* No. 360).

[22] *American Standard for Installation of Gas Piping and Gas Appliances in Buildings* (American Gas Association, 1954, ASA Z 21.30).

[23] *NBFU Standard for the Installation of Oil Burning Equipments* (National Board of Fire Underwriters, *NBFU Pamphlet* No. 31).

BIBLIOGRAPHY

Haslam and Russell: *Fuels and Their Combustion* (McGraw-Hill Book Co., New York, 1926).

A. D. Pratt: *Principles of Combustion in the Steam Boiler Furnace* (Babcock and Wilcox Co.).

H. J. Rose and F. P. Lasseter: Smoke-producing tendencies in coals of various ranks (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 329).

C. A. Barnes: *Fundamentals of Combustion in Small Stokers* (*Bituminous Coal Research, Inc. Technical Report* No. IV).

A. P. Kratz, J. R. Fellows, and J. C. Miles: *Hand-firing of Bituminous Coal in the Home* (*Illinois Engineering Experiment Station Circular* No. 46).

G. H. Cady: *Classification and Selection of Illinois Coals* (*Illinois State Geological Survey Bulletin* No. 62).

Bituminous Coal Research, Inc.:

J. M. Pilcher and R. A. Sherman: *The Treatment of Coal with Oil and Other Petroleum Products* (Technical Report VI).

R. A. Sherman and G. W. Land: *Dustless Treatment of Coals with Materials Other Than Oil* (Information Bulletin No. 4).

Questions and Answers on the Use of Fuel Oil for Dustless Treatment (Information Bulletin No. 6).

R. B. Engdahl: *Application of Overfire Air Jets* (Technical Report VII).

Bureau of Mines Publications:

Henry Kreisinger and F. K. Ovits: *Sampling and Analyzing Flue Gases* (Bulletin No. 97).

P. Nicholls and B. A. Landry: *Coke as a Domestic Heating Fuel* (Report of Investigations, R. I. 2980).

John Blizard, James Neil, and F. C. Houghten: *Value of Coke, Anthracite, and Bituminous Coal for Generating Steam in a Low-pressure Cast-iron Boiler* (Technical Paper No. 303).

P. Nicholls: *Effect of Preheat, and Distribution of Ash in Fuel Beds* (Bulletin 378).

J. F. Barkley: *Handbook, Questions and Answers for the Home Fireman* (Revised).

Anthracite Industries Laboratories Publications:

Comparison of Sizes, Egg, Stove and Chestnut Anthracite (Report 2015).

Domestic Survey (Report 2018).

Utilization of Anthracite for Domestic Heating (Report 2062).

The Crater Method of Firing (Report 2204).

Anthracite Industries Manual (Report 2403).

Oil and Gas Publications:

J. A. Moyer: *Oil Fuels and Burners* (McGraw-Hill Book Co., New York, 1937).

Combustion ("Industrial Gas Series," American Gas Association).

Comfort Heating (American Gas Association).

Gaseous Fuels (American Gas Association, 1954).

F. H. Faust and G. T. Kaufman: *Handbook of Oil Burning* (Oil-Heat Institute of America, 1951).

E. R. Weaver: Formulas and graphs for representing the interchangeability of fuel gases (*National Bureau of Standards Journal of Research*, 1951, Research Paper R. P. 2193).

Lewis and von Elbe: *Combustion Flames and Explosions of Gases* (Academic Press, New York, 1951).

CHAPTER 34

AUTOMATIC FUEL-BURNING EQUIPMENT

Classification of Stokers, Combustion Process and Adjustments, Furnace Design, Rating; Classification of Oil Burners, Combustion Process, Combustion Chamber Design; Commercial and Industrial Oil Burners, Types of Burners; Storing and Handling Heavy Fuel Oil; Oil Storage Tanks and Piping; Classification of Gas-Fired Heating Equipment, Combustion Process, Ratings; Sizing of Gas Piping, Commercial and Industrial Gas Equipment; Fuel-Burning Rates; Controls for Automatic Equipment

AUTOMATIC mechanical equipment for the combustion of solid, liquid, and gaseous fuels is considered in this chapter. The type of fuel to be burned affects the design of the heat exchanger (furnace, boiler, etc.); therefore, one should refer to Chapter 35, Heating Boilers, Furnaces, Space Heaters when considering fuels and fuel-burning equipment. Some heat exchangers are designed specifically for one type of fuel. As a result, maximum efficiency and best performance require the simultaneous consideration of fuel and fuel-burning equipment during the initial stages of equipment selection.

MECHANICAL STOKERS

A mechanical stoker is a device that feeds a solid fuel into a combustion chamber, provides a supply of air for burning the fuel under automatic control and, in some cases, incorporates a means of removing the ash and refuse of combustion automatically. Coal can be burned more efficiently by a mechanical stoker than by hand firing because the stoker provides a uniform rate of fuel feed, better distribution in the fuel bed, and positive control of the air supplied for combustion.

In many cities, measures for control of excess smoke require installation of stokers even in residences, unless fuel used contains less volatile matter than bituminous coal.

Classification of Stokers According to Capacity

Stokers may be classified roughly according to their coal burning capacities, as follows:

Class 1	10 to 100 lb per hr
Class 2	100 to 300 lb per hr
Class 3	300 to 1200 lb per hr
Class 4	over 1200 lb per hr

Class 1 Stokers

These stokers are used primarily for home heating and are designed for quiet, automatic operation. Simple, trouble-free construction and attractive appearance are desirable characteristics of these small units.

A common stoker in this class (Fig. 1) consists essentially of a coal hopper, a screw for conveying the coal from the hopper to the retort, a fan that supplies the air for combustion, a transmission for driving the coal feed worm, and an electric motor for supplying power for coal feed and air supply.

Air for combustion is admitted to the fuel through tuyeres at the top of the retort which may be either round or rectangular. Stokers in this class are made for burning anthracite, bituminous, semi-bituminous, and lignite coals, and coke. The U. S. Department of Commerce has issued commercial standards for household anthracite stokers.[1]

Units are available in either the hopper type, as shown in Fig. 1, or in the bin-feed type as shown in Fig. 2. Some stokers, particularly those designed for use with anthracite, automatically remove ash from the ashpit and deposit it in an ash receptacle. Most of the bituminous models, however, require hand removal of the ash from the fuel bed after it is fused into a clinker.

Stokers in this class feed coal to the furnace intermittently in accordance with temperature or pressure demands. A special control is used to insure sufficient stoker operation to maintain a fire during periods when no heat is required.

Class 2 Stokers

Stokers of this class are usually of the underfeed motor-driven screw type, Fig. 3, and are used extensively for heating apartments, schools, hotels, and industrial plants. They are also built in plunger-feed type, Fig. 4, with electric motor or steam or hydraulic cylinder coal feed drive. The tuyeres and retort design vary according to the fuel and load conditions. They are available for burning all types of anthracite, bituminous and lignite coals.

Stationary type grates or dead plates surround the retort of bituminous models to collect the ash and clinkers which are removed periodically by hand. Anthracite stokers in this class may be equipped with moving grates that discharge the ash into a pit, below the grates, having capacity for several weeks of ash accumulation.

Class 3 and 4 Stokers

These stokers are: underfeed side cleaning, underfeed rear cleaning, overfeed flat grate, or overfeed inclined grate type.

Underfeed side cleaning stokers are made in sizes up to about 500 boiler horsepower (or approximately 17 million Btuh). They are not so varied in design as those in the smaller classes, although the principle of operation is similar. A stoker of this type is illustrated in Fig. 4.

The rear cleaning underfeed stoker is usually of the mul-

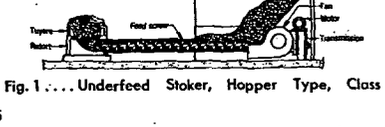


Fig. 1 . . . . Underfeed Stoker, Hopper Type, Class 1

tiple retort design. It is used in some of the largest industrial plants and central power stations. Zoned air control has been applied to these stokers, both longitudinally and transversely of the grate surface.

The overfeed flat grate stoker is represented by the various chain- or traveling-grate stokers.

Another distinct type of overfeed flat-grate stoker is the spreader type in which coal is distributed either by rotating paddles (Fig. 5) or by air over the entire grate surface (Fig. 6). This type of stoker is adapted to a wide range of fuels and has a wide application on small sized fuels, and on fuels such as lignites, high-ash coals, and coke breeze.

The overfeed inclined-grate stoker operates on the same general combustion principle as the flat-grate stoker, the main difference being that rocking grates, set on an incline, are provided in the former to advance the fuel during combustion.

Combustion Process

In anthracite stokers of the Class 1 underfeed type, burning takes place entirely within the stoker retort. The refuse of combustion spills over the edge of the retort into an ashpit or receptacle from which it may be removed either manually or automatically. Anthracite for stoker firing is usually pea, buckwheat, rice, or barley size (see Chapter 33 for a description of size).

Because the majority of the small bituminous coal stokers operate on the underfeed principle, a general description of their operation is given. When the coal is fed into the retort, it moves upward toward the zone of combustion and is heated by conduction and radiation from the burning fuel in the combustion zone. As the temperature of the coal rises, it gives off moisture and occluded gases, which are largely noncombustibles. When the temperature increases to around 700 or 800 F the coal particles become plastic, the degree of plasticity varying with the type of coal.

A rapid evolution of the combustible volatile matter occurs during and directly after the plastic stage. The distillation of volatile matter continues above the plastic zone where the coal is coked. The strength and porosity of the coke formed will vary according to the size and characteristics of the coal. While some of the ash fuses into particles on the surface of the coke as it is released, most of it remains on the hearth or grates and, as this ash layer becomes thicker with time, that portion exposed to the higher temperatures surrounding the retort fuses into a clinker. The temperature in the fuel bed, the chemical composition and homogeneity of the ash, and the time of heating govern the degree of fusion.

Most bituminous coal stokers of Classes 1, 2, and 3 require manual removal of the ash in clinker form.

In the underfeed side-cleaning stokers the fuel is introduced at the front of the furnace to one or more retorts, and is advanced away from the retort as combustion progresses,

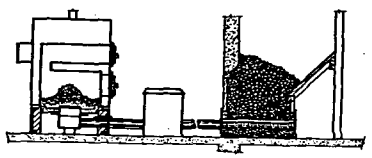


Fig. 2 .... Underfeed Stoker, Bin Feed Type, Class 1

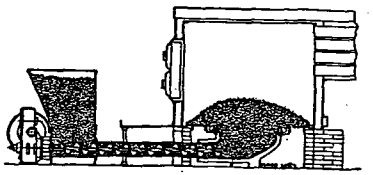


Fig. 3 .... Underfeed Screw Stoker, Hopper Type, Class 2 or 3

while finally the ash is disposed of at the sides. This type of stoker is suitable for all bituminous coals, while in the smaller sizes it is suitable for small sizes of anthracite. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process the volatile gases are released, are mixed with air, and pass through the fire where they are burned. The ash may be continuously or periodically discharged at the sides.

The underfeed rear-cleaning stoker accomplishes combustion in much the same manner as the side-cleaning type, but consists of several retorts placed side by side and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side-cleaning underfeed type.

Overfeed flat-grate stokers receive fuel at the front of the grate in a layer of uniform thickness and move it horizontally to the rear of the furnace. Air is supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ashpit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze, and also for bituminous coals, the characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker requires an arch over the front of the fuel bed to maintain ignition of the incoming fuel, and frequently a rear combustion arch.

Overfeed inclined-grate stokers are provided with rocking grates and ash plates on which ash is accumulated and dumped periodically. This type of stoker is suitable for all types of coking fuels, but preferably for those of low volatile content. Its grate action keeps the fuel bed broken up, thereby allowing free passage of air. Because of its agitating effect on the fuel, it is not desirable for badly clinkering coals. It usually should be provided with a front arch to ignite the volatile gases.

Combustion Adjustments

The coal feeding rate and air supply to the stoker should be regulated so as to maintain a balance between the load

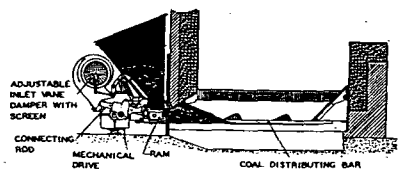


Fig. 4 .... Underfeed Side Cleaning Stoker

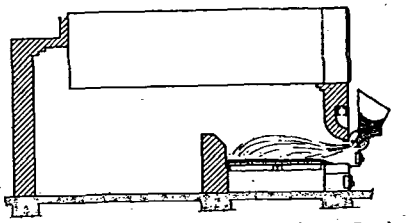


Fig. 5 .... Overfed Spreader Stoker (Rotor Type)

demand and the heat liberated by the fuel. Under such conditions, no manual attention to the fuel bed should be required, other than the removal of clinker in stokers which operate on this principle of ash removal.

As in all combustion processes, the maintenance of the correct proportions of air and fuel is essential. It is desirable to supply the minimum amount of air required to properly burn the fuel at the rate of feed.

While there may be only slight variations in the rate at which the coal is being fed, due to variations in the size or density of the coal, there may be wide variations in the rate of air flow as the result of changes in fuel-bed resistance. These changes in resistance may be caused by changes in the porosity of the fuel bed due to variations in size or friability of the coal, ash and clinker accumulation, and variations in depth of the fuel bed. Because of this variable fuel bed resistance, many bituminous stokers, even in the smaller domestic sizes, incorporate air controls which automatically compensate for these changes in resistance and maintain a constant air-to-fuel ratio. The efficiency of combustion may be determined by analyzing the flue gases, as explained in Chapters 33 and 44.

It is desirable on most stoker installations to provide automatic draft regulation in order to reduce air infiltration and provide better control during the banking, or off, periods of the stoker. It is imperative that adequate combustion air be supplied and, therefore, it is usually necessary to provide a direct outdoor opening for this purpose. (See Chapter 33.)

Combustion Chamber Design

The Stoker Manufacturers Association has published standard recommendations on setting heights for stokers having capacities up to 1200 lb of coal per hour.[a]

The empirical formulas for determining these setting heights are:

For burning rates up to 100 lb coal per hour

$$H = 0.1125\ B + 15.75 \qquad (1)$$

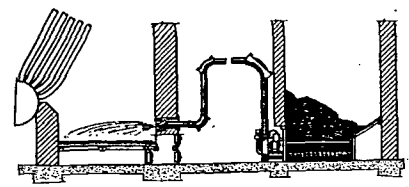


Fig. 6 .... Overfeed Spreader Stoker (Pneumatic Type)

For burning rates from 100 to 1200 lb coal per hour

$$H = 0.03\ B + 24 \qquad (2)$$

where

H = minimum setting height, inches, measured from dead plates to crown sheet for steel boilers. For cast-iron boilers height may be ⅓ H.

B = burning rate coal per hour, pounds.

Standards for minimum firebox dimensions and base heights have been formulated by the Stoker Manufacturers Association as shown in Fig. 7.[a]

In considering these recommendations, it should be understood that they show the average recommended minimum. There are many factors affecting the proper application of stokers to various types of boilers and furnaces, and, in certain instances, setting height or firebox dimensions shown in the standards may be modified without impairing performance. Such modification will depend upon the experience of the installer or designer with a particular stoker, the type of fuel used, and the construction of the boiler or furnace.

Installation of stokers (particularly smaller sizes) on the side of the boiler or furnace may facilitate clinker removal.

Rating and Sizing Stokers

The Stoker Manufacturers Association[a, 9] has suggested a method for rating solid fuel stokers as follows:

$$\frac{\text{Load (Btu per hour)}}{\text{Heating value of coal (Btu per pound)} \times \text{overall efficiency of stoker and boiler or furnace}} = \text{Stoker burning rate required (pounds of coal per hour)}$$

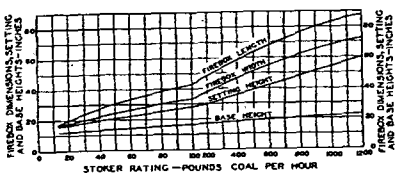
In determining the total load placed on a stoker-fired boiler by a steam or hot water heating system, a piping and pickup factor of 1.33 is commonly used in sizing the stoker, but this factor should be increased at times due to unusual conditions.

Controls

The control of coal-fired equipment is discussed in the section Controls for Automatic Fuel-Burning Equipment.

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for the oxidation of hydrocarbon liquids (fuel oil) and air under controlled con-



[a] For reference in selecting or designing boilers and furnaces for stoker firing. Dimensions shown are for net inside clearance at grate level using coal with heating value of not less than 13,000 Btu per pound. Under certain conditions smaller fireboxes will permit satisfactory performance but these dimensions are preferred normal minimums.

Fig. 7 .... Suggested Minimum Firebox Dimensions and Base Heights[a]

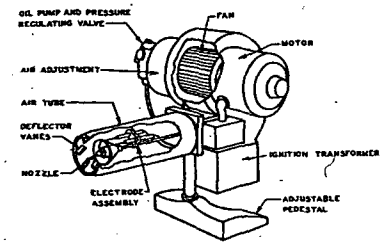


Fig. 8 .... High-Pressure Atomizing Oil Burner

conditions. Two methods, atomization and vaporization, are employed for the preparation of the fuel oil for the combustion process. Air for combustion is supplied by a blower or by draft produced by the chimney. Ignition is accomplished by an electric spark, electric resistance wire, gas pilot flame, or oil pilot flame. Burners of different types operate with luminous or non-luminous flame. Operation may be intermittent or continuous with high-low flame.

Oil burners must be used with a sealed heat exchanger (warm air furnaces, boilers, water heaters, space heaters, etc.), and must have a proper venting system.

While most oil burners operate from automatic temperature-*censing controls*, some of the simpler types are operated manually.

Classification of Burners

Domestic oil burners may be classified by type of design or operation into the following groups: pressure-atomizing or gun, rotary, and vaporizing or pot. These are further classified as mechanical draft, and natural draft.

Pressure Atomizing (Gun-Type) Burner

Gun-type burners are usually designed to burn No. 1 or 2 grade fuel oil. They may be divided into two classes: high-pressure and low-pressure atomization.

The *high-pressure atomizing* type, illustrated in Fig. 8, is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and arranged so that a

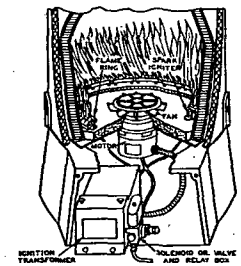


Fig. 9 .... Wall Flame Vertical Rotary Burner

spray of atomized oil is introduced at about 100 psi, and mixed in the combustion chamber with the air stream emerging from the air tube.

The oil rate can be changed by replacing the nozzle with one of different rating. *Oil nozzles* are precision devices for metering the oil and discharging the oil in a fine spray or mist. Small oil-rate changes can be made by adjustment of pump pressure. Good atomization is affected at pressures between 75 and 140 psig. Since the objective is to mix intimately the oil spray particles and air, a variety of baffles or mixing devices are employed in the air tube. The air velocity and direction is important in this mixing.

The oil pump can perform both as a suction pump (oil tank below burner), and a pressure pump for atomization.

A blower is used to supply the air for combustion. A damper or other adjustable means at the burner and a draft regulator maintain a proper air supply. Ignition is established by a high-voltage electric spark that may be operative

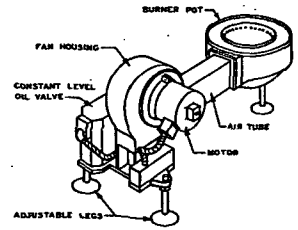


Fig. 10 .... Vaporizing Forced-Draft Pot-Type Burner

continuously while the burner is running, or just at the beginning of the running period. The burner operates on the intermittent *on-off* principle, and with a luminous flame.

The combustion process is completed in a chamber constructed of refractory material, or stainless steel, this being a part of the installation.

The *low-pressure atomizing* burner differs from the high-pressure type mainly by having means for supplying a mixture of oil and primary air to the burner nozzle. The air pressure before mixing and the pressure of the oil-air mixture at the nozzle vary with different makes of burners, but are in a low range of 1 to 15 psig for the air and 2 to 7 psig for the mixture, respectively. The various parts of the burner, except for oil and air mixing parts, are the same as shown in Fig. 8.

Rotary Burner

Most of the smaller rotary burners are of the vertical wall flame type (Fig. 9), and are used with No. 1 or 2 grade fuel oil. Oil is conveyed by gravity to the burner and is centrifugally thrown and mixed with air to a periphery ring of stainless steel or refractory material. A refractory hearth is used to prevent excessive heat loss below the burner and to prevent air leakage. Oil rate and air quantity can be adjusted to suit requirements.

The vertical rotary burners are further characterized by their installation within the ashpit of the boiler or furnace. Various types of ignition are utilized, gas and electric, either

spark or hot wire. The air for combustion is supplied partially by natural draft, and partially by fan effect of the central spinner element.

Vaporizing Burner

In the vaporizing burner, fuel oil is ignited (manually or electrically) and vaporized in a vessel or pot which is open at the top. Heat for vaporization is supplied by the combustion process. Openings in the side walls of the burner admit primary air which forms a rich mixture of air and oil vapors in the burner. Adjacent to the outlet opening, sufficient additional or secondary air is admitted to complete combustion. The openings for admitting air are arranged to obtain gradual and intimate mixing of air and oil vapor for combustion.

Fuel is fed by gravity from a constant level control valve, and the flow is either *on* (at rated capacity) or *off* (at pilot flow), according to the demand of the thermostat. However, the high fire can be reduced and the pilot fire can be increased to give almost any desired control characteristic within the range of the burner. The majority of vaporizing

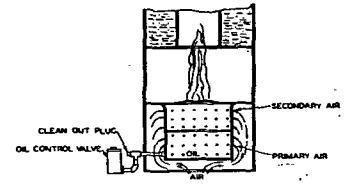


Fig. 11 .... Vaporizing Gravity Pot-Type Burner

burners are manufactured in sizes up to one gallon per hour input. Most vaporizing burners are limited to use with No. 1 fuel oil having a maximum end point of 625 F and a minimum API gravity of 35 deg.

A barometric draft regulator is required to maintain the recommended draft. A draft of not more than 0.06 in. of water column is recommended for most natural draft burners. When burners are equipped with mechanical forced draft, a slightly lower chimney draft can be used. A burner of this type is illustrated in Fig. 10. A gravity type is shown in Fig. 11.

Oil-Fired Boiler and Furnace Units

A number of types of specially designed oil-fired boiler-burner and furnace-burner units are available. Various locations of burners will be noted in such units; some having the combustion chamber and burner at the top, some at the bottom, and some at the center of the appliance. Typical combination units are shown in Figs. 12 and 13. The coordinated design of boiler (or furnace) and burner elements insures the optimum in operating characteristics, and the maintenance of balanced performance. This type of equipment usually has more heating surface, and better flue proportions and gas travel than conventional boilers or furnaces. Some of the better conversion installations, however, may equal the unit type in performance.

Operating Requirements for Oil Burners

The U. S. Department of Commerce, in conjunction with the oil burner and heating appliance industries, has established commercial standards for conversion burners and burner-appliance units which cover installation, construction and performance tests.*

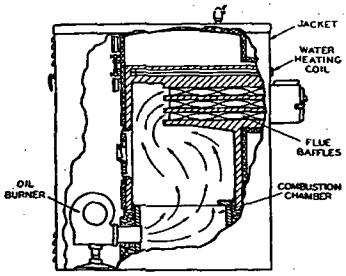


Fig. 12 .... Typical Boiler-Burner Unit

Combustion Process

Efficient combustion must produce a clean flame and use a relatively small excess of air, i.e., between 25 and 50 percent. This can be done only by vaporizing the oil quickly and completely, and mixing it vigorously with air in a combustion chamber hot enough to support the combustion. A vaporizing burner prepares the oil for combustion by transforming the liquid fuel to the gaseous state by the application of heat.

In an atomizing burner, the oil is mechanically separated into very fine particles so that the surface exposure of the liquid to the radiant heat of the combustion chamber is vastly increased and vaporization thereby promoted. The result is the ability to burn more and heavier oil within a given combustion space. Because the air enters the combustion chamber with the liquid fuel particles, mixing, vaporization, and burning occur all at once in the same space. This produces a luminous flame. A deficient amount of air is indicated by a dull red or dark orange flame with smoky tips.

An excessive supply of air may produce a brilliant white flame or a short ragged flame with incandescent sparks flashing through the combustion space. While extreme cases may be detected, it is not possible to distinguish, by eye, the effect of the finer adjustment which competent installation requires.

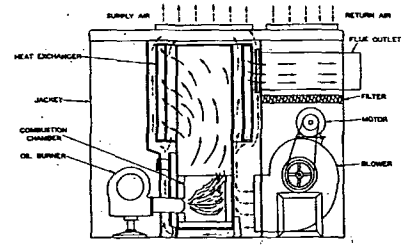


Fig. 13 .... Typical Oil-Fired Furnace Unit

Combustion Adjustments

The present-day oil burner with mechanical oil and air supply, properly installed and equipped with an automatic draft regulator, is capable of maintaining efficient combustion for a considerable period following the initial adjustments of oil and air. Eventually, certain changes will occur, and may be such that the amount of excess air will decrease below allowable limits. A decrease in air supply while the oil delivery remains constant, or an increase in oil delivery while the air supply remains constant, will make the mixture of oil and air too rich for clean combustion. The more efficient the adjustment, the more critical it will be. The oil and air supply rates must remain constant.

The following factors may influence the oil delivery rate: (1) changes in oil viscosity due to temperature change or variations in grade of oil delivered; (2) erosion of atomizing nozzle; (3) fluctuations in bypass relief pressures; and (4) possible variations in methods of atomization. Any change due to partial stoppage of oil delivery will increase the proportion of excess air. This will result in less heat, reduced economy, and possibly a complete interruption of service.

The following factors may influence the air supply: (1) changes in combustion draft due to a variety of causes (i.e., changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other purposes, possible stoppage of the chimney, and changes in draft resistance of boiler due to partial stoppage of the flues); and (2) changes in air inlet adjustments at the fan.

There should be no air leakage into the boiler or furnace setting. The overfire draft should be reduced as low as is consistent with the proper disposal of the products of combustion. When using mechanical draft burners with average conditions, the combustion chamber draft should not be allowed to exceed 0.02–0.05 in. water. An automatic draft regulator is necessary in order to maintain constant draft conditions which, in turn, improve efficiency of operation. The draft regulator should be adjusted in accordance with the manufacturer's instructions.

Even though a fan is generally used to supply the air for combustion, in most oil burners, the importance of a proper chimney should not be overlooked. The chimney should have sufficient height and size to insure a uniform draft within the limits given, if maximum efficiency throughout the heating season is to be maintained.

Measurement of the Efficiency of Combustion

Since efficient combustion is based upon a clean flame and definite proportions of oil and air employed, it is possible to determine the results by analyzing the combustion gases. It is customary to analyze only for carbon dioxide (CO_2) and to obtain the temperature of the stack gas. A showing of 10 to 12 percent indicates the best adjustment, if the flame is clean. Most of the good installations operate with 8 to 10 percent CO_2. Taking into account the potential hazard of low excess air (high CO_2), a setting to give 10 percent CO_2 constitutes a reasonable standard for most oil burners. *Commercial Standard* CS-75 requires that oil burners labeled as complying with the standard shall obtain smoke-free combustion at 10 percent CO_2. In all cases smokeless combustion is a requirement for oil burners.

Combustion Chamber Design

With burners requiring a refractory combustion chamber, the size and shape should be in accordance with the manufacturer's instructions. It is important that the chamber be as nearly airtight as is possible, except when the particular burner requires a secondary supply of air for combustion.

The atomizing burner is dependent upon the surrounding heated combustion chamber surfaces to vaporize the oil and support combustion. Unsatisfactory combustion may be due to inadequate atomization and mixing. A combustion chamber can only compensate for these things to a limited extent. If liquid fuel continually reaches some part of the firebrick surface, a carbon deposit will result. The combustion chamber should enclose a space having a shape similar to the flame, but large enough to avoid flame contact. The nearest approach in practice is to have the bottom of the combustion chamber flat, but far enough below the nozzle to avoid flame contact, the sides tapering from the air tube at the same angle as the nozzle spray, and the back wall rounded. A plan view of the combustion chamber resembles in shape the outline of the flame. This insures quick vaporization, rapid combustion and better mixing by eliminating dead spaces in the combustion chamber. An overhanging arch at the back of the fire pot is sometimes used to increase the flame travel and give more time for mixing and burning, and sometimes to prevent the gases from going too directly into the boiler flues. When good atomization and vigorous mixing are achieved by the burner, combustion chamber design becomes a less critical matter. Where secondary air is used, combustion chamber design is quite important. When installing some of the vertical rotary burners, the manufacturer's instructions must be followed carefully when installing the hearth upon which class successful performance depends in this type of burner.

Boiler Settings

Since the volume of space available for combustion is a determining factor in oil consumption, it is general practice to remove grates and extend the combustion chamber downward to include or even exceed the ashpit volume; in new installations the boiler may be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horsepower, and in this volume from 1.5 to 2.5 lb of oil per hour can properly be burned. This corresponds to an average liberation of about 38,000 Btuh per cu ft. At times much higher fuel rates may be satisfactory. For best results, care should be taken to keep the gas velocity below 40 fps. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important, and frequently it is advisable to consult an authority on this subject. It is essential in combustion chamber design to prevent flame impingement upon either metallic or firebrick surfaces. Manufacturers of oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

Controls

The control of oil-fired equipment is discussed in the section Controls for Automatic Fuel-Burning Equipment.

COMMERCIAL AND INDUSTRIAL OIL BURNERS

Oil-burning equipment for commercial and industrial applications is usually designed for burning the lower cost heavy fuel oils such as U. S. Commercial Standard Grades Nos. 4, 5, and 6. The viscosity of these oils is much greater than that of the lighter domestic grades and, therefore, the equipment required for satisfactory storage, pumping, and combustion differs greatly from that used in the typical domestic oil-burning system.

Both the initial cost and the operating cost of an oil-burning system may be affected materially by the characteristics of the industrial fuel oil to be used. These characteristics should, therefore, be given careful consideration when preparing the specifications of the fuel oil for which the system is to be designed.

Classification of Burners

Oil burners are usually classified according to the method used for atomizing the oil as (1) horizontal rotary-cup atomizing, (2) mechanical pressure-atomizing, (3) steam-atomizing, and (4) air-atomizing.

Horizontal Rotary-Cup Burner

In this type of burner, oil is delivered into a horizontal cup that is rotated at high speed. As the thin film of oil is spun from the rim of the cup, it enters a cone of high velocity primary air where very effective atomization is obtained. In most applications secondary air for combustion is supplied by natural draft through checkered openings in the floor of the combustion chamber. In some applications, however, the secondary air is admitted through openings surrounding the burner nozzle and may be supplied by forced draft.

The rotary-cup burner has been the most popular type in the capacity range from approximately 25 to 500 boiler horsepower. Its principal advantages are that it is a self-contained integral unit; it is readily adaptable to manual, semi-automatic, or fully-automatic control; and the firing rate may be easily modulated through a wide range while maintaining high combustion efficiencies. These burners are frequently used for firing packaged steam generator units.

Mechanical Pressure-Atomizing Burner

In this burner oil is atomized by passing it through a specially designed nozzle under pressures of approximately 100 to 250 psig. The air for combustion may be supplied by either natural or forced draft, and is usually admitted through an air register surrounding the burner nozzle or through checkered openings in the floor of the combustion chamber. This burner is popular for marine service and for large capacity multiple installations. It is not readily adapted to fully automatic operation.

Steam-Atomizing Burner

High-pressure steam is used in this burner for atomizing the oil. Oil is delivered to the nozzle at pressures ranging from approximately 5 to 50 psig, and steam at approximately 50 to 150 psig is admitted to the nozzle by various methods to assist in the atomization. Combustion air is usually supplied by natural draft through checkered openings in the floor of the combustion chamber or through an air register surrounding the nozzle. This burner is relatively low in first cost but is not adapted to fully automatic operation or wide variations in firing rate.

Air-Atomizing Burner

The air-atomizing burner is similar to the steam-atomizing type, except that air under pressure instead of steam is used for atomization. The performance characteristics of the two burners are also similar but higher temperature oil is usually required when using air for atomization.

STORING AND HANDLING HEAVY FUEL OILS

Preheating of Heavy Fuel Oils

Unlike the lighter grades of fuel oils that can be pumped, atomized, and burned efficiently without prior preheating, the heavier grades of residual fuel oils require conditioning by the application of heat before use. Preheating is employed to reduce the viscosity of the oil so that it can be pumped and atomized properly by the burner. The function of the burner is to atomize the oil and mix it thoroughly with the proper amount of air for prompt ignition and efficient combustion. The optimum viscosity will vary from 100 to 450 seconds Saybolt Universal viscosimeter, with the type of burner and is reached at different temperatures for different grades of oil. The range of atomizing viscosity is about as follows for normal grades of oil:

Type of Burner	Range of Atomizing Viscosities
Mechanical-Atomizing	100–300 SSU
Steam-Atomizing	200–250 SSU
Rotary-Cup-Atomizing	250–450 SSU

Table 1 lists temperatures needed to bring various oil viscosities to the proper atomizing levels. These can be used with some variation to suit individual operations.

The correct degree of preheat must be used for each grade of oil. Underheating or overheating affects the terminal vis-

Table 1 .... Preheating Temperatures to Obtain Suitable Viscosities for Atomizing

Viscosity of Oil[a]		Oil Temperature (Fahrenheit) Required				
		Desired SSU Viscosity				
In SSF at 122 F	In SSU at 100 F	100	150	200	300	450
..	100	100	80	70	61	50
..	150	120	100	90	77	66
..	200	135	110	100	87	75
21	300	150	128	115	100	88
28	400	160	138	128	108	96
30	500	170	145	133	115	102
39	750	185	160	147	126	114
50	1000	190	168	155	133	120
60	1200	200	175	160	138	125
75	1500	205	180	168	145	132
100	2000	215	190	175	150	137
115	2500	220	195	182	156	143
135	3000	225	200	187	160	146
145	3500	229	205	190	162	150
160	4000	235	208	194	165	152
170	4500	237	212	197	167	155
190	5000	240	215	200	168	157
200	5500	245	218	204	172	160
220	6000	248	220	208	175	162
260	7000	250	225	210	177	165
285	8000	254	227	214	180	167
320	9000	258	230	216	183	169
342	10,000	260	235	218	186	171

[a] SSF = seconds Saybolt Furol viscosimeter. SSU = seconds Saybolt Universal viscosimeter.

From *Fuel Oil Manual*, by Paul F. Schmidt (The Industrial Press) 1951, p. 105. Columns for 300 and 450 viscosity from Davis Engineering Corp.

cosity and actually interferes with the atomizing process. Underheated oil is too viscous for proper atomization, resulting in poor combustion, the generation of smoke and soot, and an excessively high oil consumption. Overheated oil, on the other hand, is too fluid for correct atomization and is subject to the same type of poor operating results as the underheated oil.

No. 4 fuel oils, with a maximum viscosity of 125 seconds Saybolt Universal at 100 F, as a rule do not require any preheating before atomization. They are sufficiently fluid at normal ambient temperatures for good burner operation.

No. 5 fuel oils are of two types: a light grade having a viscosity ranging between 150 and 200 seconds Saybolt Universal, that does not require preheating and hence is known as a *cold* No. 5; and a heavier grade having a viscosity ranging between 200 and 800 seconds Saybolt Universal, that requires preheating and hence is termed a *hot* No. 5 oil.

No. 6 fuel oils (also known as Bunker C) always require preheating, as their viscosities range from 900 to 10,000 seconds Saybolt Universal (50 seconds to 300 seconds Saybolt Furol).

A number of factors affect the selection of a fuel oil preheating method. The three basic factors are:

1. Type of installation (commercial heating, industrial process, power generating, etc.).
2. Type of operation (automatic, semi-automatic, manual, intermittent, continuous).
3. Type of fuel (No. 5, No. 6, Bunker C).

The pour point of an oil is another characteristic that could influence the preheating requirements, especially in areas subject to severe winter temperatures. An oil with a sufficiently high pour point may require preheating and continuous circulation at temperatures above its pour point at all stages between storage tank and burner to insure its easy pumpability. Oil specifications should be checked for this characteristic and limitations imposed, if feasible. Otherwise the oil piping system must be designed with this high pour specification in mind.

Four mediums for preheating fuel oil are in common use: steam, hot water, gas, and electricity.

With steam as the heating medium the heater may be of shell-and-tube (either straight or U-tube) type or the heaters may be of a bayonet type inserted in the oil tank. In a bayonet-type heater an open-end tube is enclosed in a larger tube which has a closed outer end. Steam enters through the inner tube while condensate is drained from the outer tube. A heater of bell-shaped type may also be connected into the oil suction line at the tank for the purpose of raising the temperature of the oil drawn from the tank.

The steam preheating method has several limitations. Mainly, it is dependent upon continuity of burner operation to keep steam available to prevent viscosities beyond pumpability. Therefore, its practicability and economy are limited to installations having continuous operation. Another limitation is that the relatively high temperature of steam may cause changes in the chemical structure of lighter grades of residual fuel oils. To prevent excessively high viscosities in the fuel oil piping, constant oil circulation is maintained or the oil piping is enclosed with steam lines in the same thermal insulation. The economy of the steam preheating method is also dependent on the availability of steam at the pressure and quantity required, and on the distance from boiler to storage tank.

The hot water preheating method utilizes equipment similar to that used for steam. Because of the possibility that oil may penetrate into the boiler through faulty heat exchanges, some devices have been developed to prevent this condition. Double-transfer heaters and fuel-oil detectors are those most frequently used. Similarly, as with steam, continuity of burner operation, or cycling burner operation, is needed to maintain the temperature of the heating medium. Since oil heating is limited by the water temperature in the boiler less transmission and transfer losses, such losses must be taken into account, particularly on installations having a considerable length of oil piping.

The indirect gas-fired preheating method is a package arrangement that uses gas as the primary energy source. It generates its own supply of low-pressure steam in a closed vapor-condensate cycle. The oil is pumped through the steam oil heater as required to compensate for radiation losses in the oil-circuit lines, whenever pumping action stops. During the layover period, the main boilers and oil burners are entirely inactive. If separate oil transfer pumps are not available, a small circulating oil pump is added to the package to provide circulation in the oil lines during the gas-firing periods.

The electrical preheating method involves use of immersion heaters plus one of two special arrangements for heating the fuel oil piping. One special arrangement is a heating cable (soil cable or plastic covered wire) wrapped around the supply line or all oil piping together with a vertical-element immersion heater in the suction pipe in the oil tank. The closeness and evenness of the spiral winding of the cable determine the heat input. The on-off periods are thermostatically controlled. For protection of the heating cable, the temperature of the oil piping has to be limited and usually must not exceed 170 F. Where pipe and oil temperatures permit wrapping of all oil piping, constant oil circulation is not required. The efficiency of this preheating method is lower than for element-immersion or electric-impedance heating because of transfer losses between the heating element and the pipe.

The other special arrangement for electrical preheating is a packaged system of electric-impedance heating for all oil piping together with immersion element heating sufficient to heat the oil used. Low-voltage high-amperage alternating current of the 60-cycle frequency is passed through the oil piping itself, generating an even amount of heat in the pipe walls. The majority heating effects are due to hysteresis and eddy currents. When removing oil from unheated storage hot excess oil is returned to blend with cold oil from storage in a special tank unit. Since all oil piping is kept at design temperature under all conditions of flow or standby, constant oil circulation or cycle precirculation is not needed. The thermostatic control of pipe and oil temperature results in fully automatic operation.

Oil burners designed for automatic operation on No. 5 or No. 6 oil are generally equipped with thermostatically controlled electric immersion heaters. The function of a burner-immersion heater is to heat the oil at the rate being burned from the transportation temperature (usually at 120 F level for No. 6 oil) to the temperature required for efficient combustion. This temperature ranges from 130 F to 220 F, depending on the grade of fuel oil and design of burner. In an emergency, the burner-immersion heater may be used alone to heat a modulated, very low quantity of oil for combustion. However, in a normal operation, it functions in conjunction with the primary preheating system.

Maintaining Piping and Tank Temperature

In using No. 5 and No. 6 oils it is necessary to make special arrangements to keep the oil fluid in the piping at all times and to keep at least a portion of the oil in the storage tank fluid as well.

The *piping* may be heated by one or more of the following methods:

1. Wrapping the supply line with electrically heated tape.
2. Passing a current through the supply line itself.
3. Enclosing the supply line with a steam or hot water line in the same envelope.
4. Insulating the supply line so as to decrease the thermal losses.
5. Setting up a dual system in which the primary preheaters have a capacity greater than the burner requirement and returning the excess heated oil continually to the fuel oil tank.

The oil *in the tank* or a portion of it may be heated by one of the following methods:

1. By steam coils in the bottom of the tank.
2. By hot water coils in the bottom of the tank.
3. By a suction-bell, which is provided with either steam or hot water coils, in the bottom of the tank.
4. By providing a hot-well that extends from the top to the bottom of the tank, and provides enough heated oil to supply the burner under a full load.
5. By submerging an electric preheater in the wall or bottom of the tank.

Pipe Sizes for Heavy Oil

Table 2 gives recommended good practice pipe sizes for handling No. 5 and No. 6 oils.

Oil Storage Tanks and Piping

Storage tanks, also the piping and pumping facilities for delivering the oil from the tank to the burner, are very important considerations in the design of an industrial oil-burning system. The construction and location of the tank and oil piping are usually subject to local regulations which should always be consulted, and to the standards of the National Board of Fire Underwriters, such as *NFBU Pamphlet* No. 31.

The size of the storage tank is usually based on the rate at which oil is to be consumed, the space available, method of oil delivery (truck or tank car) and the ease with which deliveries can be made under all weather conditions. A minimum capacity of at least the maximum weekly usage is desirable. If deliveries are to be made by tank car, the capacity should not be less than 15,000 gal.

The 275-gal oil storage tank is commonly used for domestic installations. In most localities it may be installed in basements without being enclosed. Where a greater storage capacity is necessary, local regulations usually permit inside installation of two tanks of 275 gal each in accordance with *NBFU Standards Pamphlet* No. 31.

Controls

The control of oil-fired equipment is discussed in the section Automatic Control of Fuel-Burning Equipment.

GAS-FIRED HEATING EQUIPMENT

A gas burner is defined by the American Gas Association as "a device for the final conveyance of the gas, or a mixture of gas and air, to the combustion zone." Burners used for domestic heating are of the atmospheric injection, luminous flame, or power burner types.

Because of the ease with which gas fuel may be controlled, automatic gas-fired heating equipment has become very widely used, and is available in a number of types of domestic gas heating appliances and systems. These may be classified in types designed for central heating plants and those for unit application. Gas-designed units and conversion burners are available for the several kinds of central systems in which gravity and forced warm air furnaces or steam and hot water boilers are used, and for other applications where warm air floor furnaces and room heaters are installed in the space being heated.

Central Heating Systems

Boilers and furnaces specially designed for gas-firing incorporate design features for obtaining maximum efficiency and performance. Small flue passes to secure good heat transfer, the use of materials resistant to the corrosive effects of products of combustion, and draft hoods are notable features. Control equipment includes gas-pressure regulators, automatic pilots, and limit controls designed to protect the appliance and to insure safety of operation. A boiler designed for gas-burning is illustrated in Fig. 14.

Gas-designed boilers for hot water or steam heating are available in cast-iron, steel, and non-ferrous metals. The burners are located beneath the sections, and the flue gases pass upwards between the sections to the flue collector. Some boilers are designed to provide domestic hot water through the use of tankless or instantaneous heaters.

Warm air furnaces are of two types, gravity and forced

Table 2 .... Recommended Minimum Suction and Return Pipe Sizes for Fuel Oil Lines[a]
(For Nos. 5 and 6 Fuel Oil)[b]

Equivalent Length of Suction Line Feet[c]	Pump Capacity, Gallons per Hour													
	50		100		150		200		300		600		900	
	Grade of Oil (Number)													
	5	6	5	6	5	6	5	6	5	6	5	6	5	6
	Diameter of Pipe, Inches													
10–50	1¼	1¼	1¼	2	2	2	2	2	2	2	2½	3	3	4
60–100	1¼	1¼	2	2	2	2	2	2½	2½	2½	3	3	3	4
110–150	1¼	2	2	2	2	2½	2½	2½	2½	3	3	4	4	4

[a] Pipe sizes shown are based on vertical suction lift not to exceed 10 ft from bottom of storage tank to oil pump.
[b] A suction line length in excess of 100 linear feet is not recommended for systems using No. 5 or No. 6 oil. Where the length is greater a separate pump should be installed near the storage tank.
[c] Equivalent length of pipe includes all horizontal and vertical measured piping plus proper allowance for valves and fittings as follows:
Multiply pipe diameter in inches by factor below to obtain equivalent pipe length in feet.
Elbow = Factor 3; Tee = Factor 5; Gate Valve = Factor 1½; Globe Valve = Factor 9.

air. The former depends upon a thermal head to provide circulation of air over the heat exchanger and thence directly to the space to be heated, usually through ducts. Most gravity furnaces can be equipped with humidifiers. Forced warm air furnaces utilize a motor-driven blower to circulate air over the heat exchanger and through the ducts. A common type is quite similar to Fig. 13 except that gas burners, pilots, controls, and gas pressure regulator are substituted for the oil burner, and a draft hood is attached to the outlet of the furnace. Due to the pressure developed by the blower, ductwork becomes smaller and the location of the furnace less critical than in gravity systems. Designs of forced air furnaces vary from the conventional types that deliver heated air to overhead ductwork to the downflow types

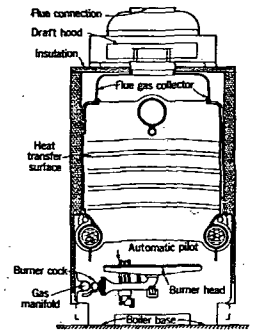


Fig. 14 .... Gas-Fired Boiler

wherein the heated air is delivered at the bottom of the furnace, this latter type being designed for perimeter or radiant-panel heating. Several types of low height horizontal furnaces are available for installation in attics or under floors of unexcavated houses. These furnaces are designed for compactness. The A.G.A. listed output at the boiler nozzle or at the bonnet for forced air furnaces is 80 percent of the approved input to the burners. The corresponding output for gravity furnaces is 75 percent.

Conversion Burners

Conversion burners are complete burner and control units designed for installation in existing boilers and furnaces. A typical gas conversion burner is shown in Fig. 15. Atmospheric conversion burners may have drilled-port, slotted-port, or single-port burner heads. Single-port types have become increasingly popular in the last few years due to their simplicity of construction and ease of installation. In rectangular or round, drilled and slotted-port burners, baffles of clay or metal are usually used to direct the products of combustion toward the side walls of the combustion chamber. In single-port burners the flame usually impinges on a horizontal cast-iron, stainless steel, or ceramic flame spreader which directs the products of combustion toward the side walls. For certain applications, particularly wet-base boilers, a horizontal single-port burner is utilized. This design produces a horizontal flame which strikes a curved baffle that spreads the flame in a fan shape.

Several power burners in domestic sizes are available. These are of gun-burner design and feature small mechanical blowers for supplying all the air required for combustion. These are particularly desirable for handling restricted flue boilers or furnaces where an atmospheric burner cannot obtain the desired output.

Conversion burners for domestic application are available in sizes ranging from 50,000 to 400,000 Btuh capacity. It should be noted that the American Standard Listing Requirements are limited at present to conversion burners having input capacities up to 400,000 Btuh. Burners of larger capacity, for use in large steel boilers, are usually engineered by a contractor specializing in this type of work, or by the local utility company. They are available in a large number of sizes and types. In some cases the burner may be an assembly of multiple burner heads filling the entire firebox. For conversion burner installation in boilers requiring more than 400,000 Btuh input, reference should be made to the current *American Standard Requirements for Installation of Gas Equipment in Large Boilers*, ASA Z21.33.

Domestic sizes of conversion burners should conform to current *American Standard Listing Requirements for Domestic Gas Conversion Burners*, ASA Z21.17. As the successful and safe performance of a gas conversion burner depends on numerous factors other than those incorporated in such equipment, installations of this kind must be made in strict accordance with current *American Standard Requirements for Installation of Domestic Gas Conversion Burners*, ASA Z21.8. Also, draft hoods conforming to current *American Standard Listing Requirements for Draft Hoods*, ASA Z21.12, should be installed (in place of the dampers used with a solid fuel) on all boilers and furnaces converted to burn gas. The A.G.A. estimates the output of conversion burners at 72 percent efficiency. This value is based on average installations and average heating equipment performance.

Floor Furnace

The floor furnace is used to supply central heating in mild climates, or for auxiliary heating or heating of single rooms in colder climates. It is used for heating first floors, or where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where

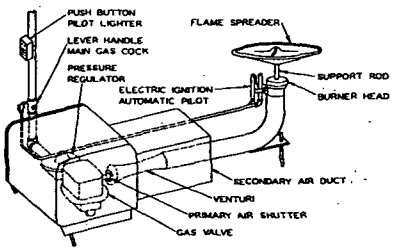


Fig. 15 .... Typical Single-Port Gas Conversion Burner

all rooms are on the ground floor, thus giving the heating system flexibility.

In floor-furnace applications, the heating element, gas piping, and flue vent piping are suspended below the floor, and the only part in the room being heated is the circulating air register which may be of the single type installed in the floor, or of the dual type installed in a partition, and heating two rooms. The A.G.A. output rating of floor furnaces is based on an efficiency of 70 percent for the gravity-type and 75 percent for the fan-type unit.

Another type of heating appliance is the vented recessed heater which is either a gravity or fan-type appliance designed for installation in or permanent attachment to a wall, floor, ceiling, or partition. Such heaters may have short stub ducts to conduct heated air to adjacent rooms. They are installed primarily in new homes, usually being plastered into the wall as a permanent part of the building. The A.G.A. heat output rating is required to be 75 percent of the approved heat input to the burners for fan-type units, and 70 percent for gravity-type units.

Space Heaters

Space heaters are defined as heating units that take the air for combustion from the space being heated. They may be broadly classified as room heaters and units heaters.

Room heaters are used for heating single rooms or connecting rooms with good circulation between them and, except for wall-type heaters, they are semi-portable. Unvented-type room heaters should not be used unless space is properly ventilated to prevent excess moisture and to provide sufficient combustion air to prevent formation of carbon monoxide. All types of room heaters are capable of automatic control, although they are generally controlled manually. When equipped for automatic control, they must have an automatic pilot as part of the control equipment. Room heaters may be classified as follows:

Circulators, vented and unvented, that heat the room mainly by convection, although some have radiants over the burner, with windows in the front to allow some heat by radiation.

Radiant heaters, usually unvented, although some vented types are available, have a refractory directly above the burners which is heated to incandescence, and gives off an appreciable part of its heat in the form of radiant energy.

Wall heaters, vented or unvented, are usually a type of radiant heater constructed with sufficient insulation (either solid or circulating air) to prevent overheating of the casing, and are built into a wall with the front flush with the wall surface. Their heat input rating is limited to a maximum of 25,000 Btuh.

Although an output rating is not required by *A.G.A.*, the efficiency of a vented room heater must be not less than 70 percent when rated above 20,000 Btuh input, and 65 percent when rated at less than 20,000 Btuh input.

Unit heaters are used extensively for heating large spaces such as stores, garages, and factories. These heaters consist of a burner, heat exchanger, fan for distributing the air, draft hood, automatic pilot, and controls for burners and fan. They are usually mounted in an elevated position from which the heated air is directed downward by louvers. Some unit heaters are suspended from the ceiling, and others are free-standing floor units of the heat tower type. Unit heaters are classified for use with, or without, ducts depending on specific A.G.A. approval. When connected to ducts they must have sufficient blower capacity to deliver an adequate air quantity against the duct resistance. Heat output is required to be at least 75 percent of heat input to the burners.

Duct furnaces are usually of the unit heater type without the fan, and are used for heating air in existing duct systems where blowers are provided for moving the air through the system. Duct furnaces are tested for operation at much higher static pressures than are obtained in unit heaters.

Combustion Process and Adjustments

Most domestic gas burners are of the atmospheric injection (Bunsen) type in which primary air is introduced and mixed with the gas in the throat of the mixing tube. For normal operation of most atmospheric type burners, 40 to 60 percent of the theoretical air as primary air will give best operation. Slotted-port and ribbon burners may require from 50 to 80 percent primary air for proper operation. The amount of excess air required in practice depends upon several factors, notably: uniformity of air distribution and mixing, direction of gas travel from burner, and the height and temperature of combustion chamber. With power type burners utilizing motor driven blowers to provide both primary and secondary air, the excess air can be closely controlled while securing proper combustion.

Secondary air is drawn into gas appliances by natural draft. As with other fuels, excess secondary air constitutes a loss, and should be reduced to a proper minimum, which usually cannot be less than 25 to 35 percent, if the appliance is to meet ASA approval requirements. Yellow flame burners depend upon secondary air, alone, for combustion.

The flame produced by atmospheric injection burners is non-luminous. Air shutter adjustments for manufactured gas should be made by closing the air shutter until yellow flame tips appear, and then by opening the air shutter to a final position at which the yellow tips just disappear. This type of flame obtains ready ignition from port to port, and also favors quiet flame extinction. When burning natural gas, the air adjustment is generally made to secure as blue a flame as is obtainable without lifting of flames from burner ports. Manufacturers' instructions for burner adjustment should be followed.

Gas-designed equipment does not usually incorporate any means for varying the secondary air supply (and hence the CO_2). The amount of effective opening and baffling is determined by A.G.A. tests and cannot be varied. Gas conversion burners, however, do incorporate means for controlling secondary air to permit adjustment over a wide range of inputs according to the type of installation encountered. The manufacturer's specific instructions contained in the installation manual required to be supplied with each burner should be followed. It is possible through the use of suitable indicators to determine whether carbon monoxide is present in flue gases. For safe operation carbon monoxide should not exceed 0.04 percent (air-free basis) when operating at the maximum gas input to the burners.

During recent years, the use of fuel gas (natural, manufactured, and liquefied petroleum) has become increasingly popular for heating tourist cabins and motels. Due largely to the intermittent use of such equipment and the fact that so many of the users are entirely unfamiliar with the fundamentals governing its operation, many special and unusual problems arise. The importance of safeguarding the public interest has prompted some state and city officials to adopt and enforce legislation governing not only the manner in which these appliances are installed but the extent to which they must be equipped with reliable safety devices.

for example, connection to an effective flue is mandatory. Furthermore, all such heating equipment must incorporate an automatic ignition pilot that will shut off gas supply to the main burner or burners in the event of pilot outage. If liquefied petroleum gas is used, the automatic pilot must cut off all gas supply to the appliance, including the pilot burner, in case the pilot flame becomes extinguished.

Rating and Sizing Gas Appliances

Input rating for a gas appliance is established by demonstrating that the appliance can meet the Approval Requirements of the ASA. The tests are conducted at the A.G.A. Testing Laboratories. Output rating is determined from the approved input and a minimum required efficiency stated in the Approval Requirements, and is the heat available at the outlet.

Although gas-burning equipment usually is completely automatic, maintaining the temperature of rooms at a predetermined figure, there are some manually controlled installations. In order to overcome effectively the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 percent greater than the equivalent standard radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up loads and consequently, it is possible to use a lower selection factor. For a gas-fired boiler or furnace under thermostatic control, a factor of 20 to 25 percent is usually sufficient for pick-up allowance.

In those installations, in mild climates where 100 percent outdoor air is used, furnaces should be of larger size in order to provide adequate capacity and quick pickup under intermittent heating conditions.

The factor to be allowed for loss of heat from piping will vary somewhat, the proportionate amount of piping installed being greater for small installations than for large ones. For selection factors to be added to installed radiation under thermostatic control, see Chapter 35.

Appliances used for heating with gas should bear the approval seal of the A.G.A. Testing Laboratories on the manufacturer's nameplate; together with the official input and output ratings. It is not permissible to operate a gas heating unit above its stated rating. It may be necessary to operate below this rating at elevations above 2000 ft. Installations should be made in accordance with recommendations of the American Gas Association, as shown in current *American Standard for Installation of Gas Piping and Appliances in Buildings*, ASA Z21.30, and in accordance with local regulations.

Sizing of Gas Piping

Piping for gas appliances should be of adequate size, and so installed as to provide a supply of gas sufficient to meet the maximum demand without undue loss of pressure between the point of supply (the meter) and the burner. The size of gas pipe required depends upon the following:

1. Maximum gas consumption to be provided.
2. Length of pipe and number of fittings.
3. Allowable loss in pressure from the outlet of the meter to the burner.
4. Specific gravity of the gas.

To obtain the cubic feet per hour of gas required by the burner, divide the Btu input rate at which the burner will be adjusted, by the average Btu heating value per cubic foot of the gas.

Capacities of different sizes and lengths of pipe, in cubic feet per hour, with a pressure drop of 0.3 in. of water column for a gas of 0.60 specific gravity, are shown in Table 3. In adopting a 0.3 in. pressure drop, due allowance for an ordinary number of fittings was made.

To convert the values given in Table 3 to capacities for another gas of different specific gravity, multiply the tabular values by the multipliers shown in Table 4.

Pipe capacities for gas flow up to 100,000 cfh for various lengths of pipe are given in Fig. 16 which is based on a specific gravity of 0.60. The capacity at a given pressure loss for a different specific gravity may be obtained by multiplying Fig. 6 capacities by a multiplier obtained from Table 4.

Table 3 .... Capacity of Gas Piping—Cu Ft Per Hr.
(At pressure drop of 0.3 in. water. Specific gravity = 0.60)

Length of Pipe Ft	Iron Pipe Size (IPS) Inches										Length of Pipe Ft
	½	¾	1	1¼	1½	2	3	4	6	8	
15	76	172	345	750	1220	2480	6500	13880	38700	79000	15
30	52	120	241	535	850	1780	4700	9700	27370	55850	30
45	43	99	199	435	700	1475	3900	7900	23350	45600	45
60	38	88	173	380	610	1290	3450	6800	19330	39500	60
75		77	155	345	545	1120	3000	6000	17310	35300	75
90		70	141	310	490	1000	2700	5500	15800	32250	90
105		65	131	285	450	920	2450	5100	14620	29850	105
120			120	270	420	860	2300	4800	13680	27920	120
150			109	242	380	780	2090	4350	12240	25000	150
180			100	225	350	720	1950	4000	11160	22800	180
210			92	205	320	660	1780	3700	10330	21100	210
240				190	300	620	1680	3490	9600	19740	240
270				178	285	580	1580	3250	9000	18610	270
300				170	270	545	1490	3000	8500	17660	300
450				140	226	450	1230	2500	7000	14420	450
600				119	192	390	1030	2130	6000	12480	600

From *American Standard Installation of Gas Piping and Gas Appliances in Buildings*, ASA Z21.30-1954.

Table 4 .... Multipliers for Various Specific Gravities
(For Use With Table 3 and Fig. 16)

Specific Gravity	Multiplier	Specific Gravity	Multiplier
0.35	1.31	1.00	0.775
0.40	1.23	1.10	0.740
0.45	1.16	1.20	0.707
0.50	1.10	1.30	0.680
0.55	1.04	1.40	0.655
0.60	1.00	1.50	0.633
0.65	0.962	1.60	0.612
0.70	0.926	1.70	0.594
0.75	0.895	1.80	0.577
0.80	0.867	1.90	0.565
0.85	0.841	2.00	0.547
0.90	0.817	2.10	0.535

COMMERCIAL AND INDUSTRIAL GAS EQUIPMENT

No attempt can be made, due to space limitation, to describe or classify the varied types of equipment used for commercial and industrial applications. Much of the equipment utilized is custom built to suit a particular process. Certain standard parts such as venturi and burner head sets, ring burners, proportional mixers, or compressors, are purchased for assembly in existing equipment, usually by the installer. Furnaces to provide particular atmospheres or for annealing, soaking pits, metal melting, etc. are usually engineered by industrial gas-equipment manufacturers.

Burners utilized in industrial equipment often operate at gas pressures higher than encountered in domestic usage. High-pressure gas provides flexibility of control required in the production of constant temperatures. In addition, air-gas ratios can be closely maintained under throttling conditions, and heat liberation per cubic foot of combustion space is considerably greater than can be achieved with lower pressures.

A few of the many basic types of burners used are as follows: single-pipe high-pressure, proportional mixing, premix, two-pipe high-pressure, and atmospheric.

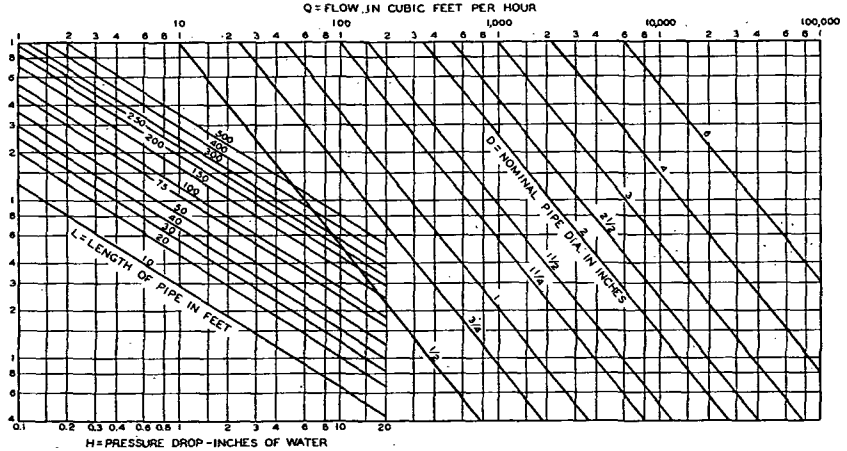
The *single-pipe high-pressure burner system* utilizes gas delivered to the orifice of a venturi at pressures from 10 to 50 psig. A constant pressure compressor, sometimes remotely placed, is occasionally used to increase the gas pressure. The velocity of the gas issuing from the orifice entrains primary air which is controlled by an air shutter on the venturi tube. An advantage of this system is that when the gas pressure is reduced (lowering the input) the air-gas ratio will remain constant over a fairly wide range.

In the *proportional mixing burner*, an enclosed type venturi is used and high-pressure air is delivered through an orifice. The resulting partial vacuum entrains the gas which is supplied through a zero pressure governor. When the air pressure is reduced, the amount of gas entrained is similarly reduced, thus maintaining a constant air-gas ratio over a wide fuel capacity range.

The *pre-mix burner system* embodies a compressor which mixes air and gas intimately and discharges the mixture through pipes to the burners where it is discharged through orifices and burned. The pre-mix compressor can be remotely installed and the mixture can be closely controlled.

The *two-pipe high-pressure burner* system utilizes both air and gas under pressure, both being delivered through orifices and mixed prior to delivery to the burner.

The *atmospheric burner* system is not commonly used. It consists of a burner supplied with gas at a pressure of 2



Based on Spitzglass Friction Factor and 0.60 Specific Gravity. To convert for other specific gravity see Table 4.

Fig. 16 .... Flow vs. Pressure Drop for Low Pressure Natural Gas

to 12 in. water. A portion of the air or primary air required for combustion is entrained from the atmosphere by the stream of gas issuing from an orifice, and the air-gas mixture is delivered through a venturi mixing tube to the burner head and parts. About 40 to 60 percent of the air required for combustion is in the form of primary air and the remainder, known as secondary air, is supplied from the air surrounding the flame.

Each system embodies characteristics essential for certain applications. It is important that proper and qualified engineering authorities be consulted prior to the purchase, installation, or operation of industrial gas equipment.

FUEL BURNING RATES

The burning rate for automatic fuel-burning devices is determined by the gross heat output required of the boiler, or furnace, to carry the net heating load, plus allowances for system losses and pickup. General values for these allowances have been given in preceding text. Detailed information for piping and pickup allowances for steam and hot water systems, is given in Chapter 35, and for warm air systems, in Chapter 18.

When the gross output, operating efficiency, and heat value of the fuel are known, the required rate of burning can be determined by means of Figs. 17, 18, and 19 for the several fuels. As the rate of fuel-burning is directly proportional to the load for a given efficiency, these charts can be extended by moving the decimal points the same number of digits in both vertical and horizontal scales.

The correct fuel-burning rate can be determined directly from the several charts for oil or gas burning installations, as these customarily operate on a strictly intermittent basis. These fuel burning devices usually introduce the fuel at a single fixed rate during the *on* periods, and this rate should be sufficient to carry the gross load. In the case of coal stokers, which are usually capable of variable rates of firing, it is desirable to operate at as low a rate as weather conditions will permit, but the maximum firing rate of the stoker should be sufficient to carry the gross load. This rate may be determined by the same method as used for oil or gas.

CONTROLS FOR AUTOMATIC FUEL-BURNING EQUIPMENT

Controls for the automatic fuel-burning equipment described in this chapter are outlined here. The basic requirements for oil burners, gas burners and coal burners (stokers) are included and the term *burner* refers to all three types of fuel-burning equipment. Controls for these burners can be classified as *operating controls, limit controls* and *primary controls*. The schematic diagrams shown in Fig. 20 indicate the relationships of the basic components. The *power supply* may be line voltage for each type of burner. In control systems for domestic burners the line voltage may be reduced by means of a transformer for those portions of the circuits indicated by dotted lines. Such transformers are usually built into the stoker and oil burner *primary controls*. Control systems for domestic gas burners frequently contain a separate transformer as shown. There are domestic gas burner control systems which obtain their electrical power supply from the conversion of heat to low-voltage electrical energy, in which case no transformer would be required. As the burner sizes increase, the tendency to use all line voltage controls increases in each of these general categories.

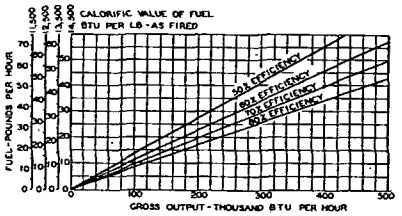
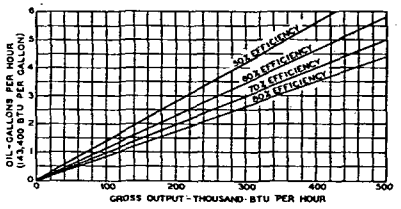


Fig. 17 .... Coal Fuel-Burning Rate Chart



* This chart is based upon No. 2 oil having a heat content of 143,400 Btu per gallon. If other grades of oil are used multiply the value obtained from this chart by the following factors: No. 1 oil (139,000 Btu per gallon) 1.032; No. 4 oil (144,500 Btu per gallon) 0.992; No. 5 oil (148,000 Btu per gallon) 0.982; and No. 6 oil (150,000 Btu per gallon) 0.956.

Fig. 18 .... Oil Fuel-Burning Rate Chart*

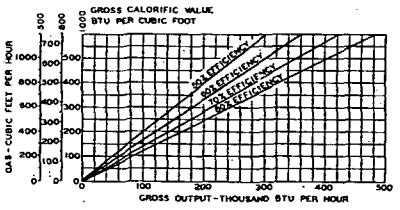
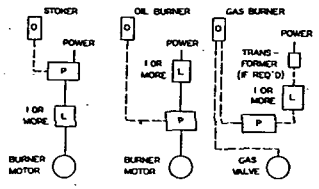


Fig. 19 .... Gas Fuel-Burning Rate Chart



(O = *Operating Control*; P = *Primary Control*; L = *Limit Control*)

Fig. 20 .... Schematic Diagrams of Some Typical Burner Control Systems

Operating Controls

Operating controls initiate the starting and stopping of the burner and on larger burners they usually vary the fuel and air input to the combustion chamber in response to varying load demands. For residential and small commercial heating plants with fuel inputs up to aproximately 500,000 Btuh, the *operating controller* is usually some form of room thermostat which automatically starts and stops the burner through the primary control as the *limit controls* permit. As the size of the heating system increases, the use of a room thermostat as the *operating controller* for the burner becomes less common. An insertion thermostat (for furnaces), immersion thermostat (for hot water boilers) or pressure controller (for steam boilers) may be applied as an *operating controller*. Also, such devices as submaster and compensated controllers (which are reset from outdoor temperature for governing the bonnet temperature in large furnaces or the water temperature in a boiler) come into more frequent use on the larger installations. The still larger burners are usually equipped to vary the rate of fuel and air input in response to varying load demands.

Limit Controls

Limit controls guard against unsafe temperature, pressure or water level to assure that safe conditions prevail for the operation of the burner. Regardless of the type of fuel burned, draft controls increase combustion efficiencies and reduce flue gas loss by maintaining proper draft conditions at all times. For fuel input requirements above approximately 1,000,000 Btuh, some additional factors become common to all types of burners. For example, the draft in the firebox is controlled at a suitable constant for all firing rates, including the *off* period, by measuring the draft conditions in the firebox and actuating a *damper operator* to position the damper at the boiler outlet. Where necessary, provision may be made to open the boiler outlet damper wide before permitting the burner to start and then to return this damper to automatic control after the burner has been started. Also, the firing rate of these larger burners is frequently automatically adjustable. Provisions are made for always starting the burner on its low firing rate, with facilities for increasing the firing rate slowly after the burner has been started. It is not recommended and codes seldom permit the use of one and the same instrument as both an *operating controller* and a *high limit control*. In such cases an *operating controller* will be adjusted to control at the desired set point while the *high limit control* will be adjusted to stop the burner at some higher point in the event that the *operating controller* fails to stop the burner before the higher shut-off point is reached. For example, to maintain a constant water temperature in a hot water boiler, the operating controller may be set to stop the burner when the boiler water temperature reaches 180 F. The *high limit control* should then be set to stop the burner at about 200 F.

Various interlocks or limit controls are usually included to assure safe operation of these larger burners. A latch-switch interlock prevents burner operation if the burner is not in its proper firing position. Another interlock must prove that the burner is in its low-fire position before the burner can be started. Safe draft conditions should be proved before the burner can be started or allowed to run, especially where induced draft fans are involved. A smoke detecting device may sound an alarm and stop the burner if excessive smoke is produced. Also, protection against electrical overloads must be provided.

Primary Controls

Primary controls provide for the safe start and operating procedure of the burner. They are somewhat different for each of the three types of fuel burners.

1. *Stokers.* Stokers are equipped with refueling, or *hold-fire*, controls. These *primary controls* cause the stoker to intermittently feed a predetermined amount of coal during the long *off* periods of the *operating controller* to maintain ignition and to reduce the possibility of having idle fuel beds damage stoker castings. In some cases the *primary control* system includes an *out-fire* feature which stops the feeding of coal into the firebox if the fire goes out.

2. *Oil Burners. Primary controls* required for oil burners depend on the size and type of burner. Some of the more common *primary controls* are discussed in following paragraphs a to d.

a. Domestic oil burners of the vaporizing type usually require a *primary control* that will (1) meter the correct amount of oil into the burner to maintain the pilot flame; (2) regulate the required amount of oil into the burner for its high flame operation; and (3) completely shut off the flow of oil in the event of a flame failure.

b. Domestic oil burners of the pressure-atomizing type require a *primary control* that will (1) energize the burner motor and electric ignition; (2) test for the establishment of main burner flame and stop the burner and ignition if flame has not been established; (3) cut off the ignition after a safe burner flame has been established (does not apply to constant ignition type burners); (4) monitor the flame continuously during burner operation and stop the burner in the event of flame failure (manual reset is necessary following a flame failure lock-out); and (5) stop the burner when either the *operating controller* or *limit control* requires it.

c. Commercial and industrial oil burners of the pressure atomizing type have *primary controls* that are essentially the same as those outlined for domestic oil burners of the pressure atomizing type. In addition, some models are equipped with a solenoid oil-shut-off valve to provide a pre-purge period, if desired, and a quick shut-off of the oil flow when the burner is stopped.

d. Commercial and industrial oil burners of the horizontal rotary-cup type are usually equipped with a more elaborate *primary control* system. The primary control is often described as a programming combustion safety control. Its most important operations include (1) starting the burner upon a call for heat; (2) providing a pre-purge period before activating the gas-electric ignition; (3) proving the presence of proper ignition before permitting the flow of oil to the main burner; (4) proving the presence of a stable main burner flame within a predetermined period of time; (5) shutting off the gas-electric ignition after a predetermined length of time has elapsed; (6) monitoring the main burner flame during its entire *on* period and stopping the burner quickly in the event of a flame failure (it is usually required that the flow of oil through the burner be stopped within four seconds following a flame failure); (7) stopping the flow of oil through the burner when either the *operating controller* or a *limit control* requires it; (8) operating the burner for a predetermined length of time after the main fuel valve has been closed; and (9) locking out on a flame failure shutdown (the programming control then must be manually reset prior to a restart).

When steam-, air-, or mechanical-atomizing burners are

automatically controlled, their sequence of control is similar to that described for horizontal rotary-cup burners.

3. *Gas Burners.* Some of the more common *primary controls* for gas burners are as follows.

a. Domestic gas burners of the atmospheric type require a *primary control*, to insure safe starting and operating conditions, which will (1) prevent opening of the gas valve unless the pilot is operating properly, and (2) stop the flow of gas through the burner if safe ignition conditions do not prevail at all times.

b. Commercial and industrial gas burners of the atmospheric type require a *primary control* whose function is similar to that for domestic sizes of these burners except in the rate of response to a pilot flame failure. The pilot flame burns constantly and, in general, the primary control system will (1) ensure operation only when ignition of the main burner will be safe; (2) start and stop the main burner when required to do so by either the *operating controller* or a limit control; and (3) close the safety shut-off valve in the gas supply line to the burner upon a flame failure or an electrical power failure.

c. Commercial and industrial gas burners of the power type equipped with electric ignition require a *primary control* whose most important operations include: (1) starting the burner motor upon a call for heat by the *operating control*; (2) providing a prepurge period before activating the gas-electric ignition; (3) proving the presence of proper ignition before allowing gas to flow to the main burner or shutting off the electric ignitor; (4) closing the safety shut-off valve in the event of a flame failure (it is usually required that the flow of gas be stopped within four seconds following a flame failure); (5) stopping the burner when either an *operating controller* or a *limit control* requires it; and (6) locking out on a flame failure shut-down (the programming control then must be manually reset prior to a restart).

REFERENCES

[1] *Domestic Burners for Pennsylvania Anthracite (Underfeed Type)* (U. S. Department of Commerce, *National Bureau of Standards Commercial Standard* No. CS48-40).

[2] Industry standards, recommended practices, technical information (*Stoker Manufacturers Association Manual*, Stoker Manufacturers Association).

[3] Code for determination of rated capacities of anthracite underfeed stokers, adopted June 1, 1944, and Code for determination of rated capacities of bituminous underfeed stokers, adopted May 3, 1944 (*Stoker Manufacturers Association Manual*, Stoker Manufacturers Association).

[4] *Automatic Mechanical Draft Oil Burners Designed for Domestic Installations* (U. S. Department of Commerce, *National Bureau of Standards Commercial Standard* No. CS75-42). *Flue Connected Oil Burning Space Heaters Equipped with Vaporizing Pot Type Burners* (U. S. Department of Commerce, *National Bureau of Standards Commercial Standard* No. CS101-43). *Warm-Air Furnaces Equipped with Vaporizing Pot-Type Oil Burners* (U. S. Department of Commerce, *National Bureau of Standards Commercial Standard* No. CS104-46). *Oil-Burning Floor Furnaces Equipped with Vaporizing Pot-Type Burners* (U. S. Department of Commerce, *National Bureau of Standards Commercial Standard* No. CS113-44).

BIBLIOGRAPHY

A. J. Johnson: Performance expectancy of domestic underfeed stokers for anthracite (*AIME Transactions*, Vol. 119, 1936).

R. A. Sherman, E. R. Kaiser, and H. R. Limbacher: *The Relation of the Size of Bituminous Coals to Their Performance on Small Underfeed Stokers—Burning Tests on Four Typical Coals* (Bituminous Coal Research, Inc., Technical Report No. 1—Part II, July 1937).

J. A. Moyer: *Oil Fuels and Burners* (McGraw-Hill Book Co., New York).

H. F. Tapp: *Handbook of Oil Burning* (American Oil Burner Association, superseded by OHI).

F. H. Faust and G. T. Kaufman (eds.): *Handbook of Oil Burning* (Oil Heat Institute of America).

C. H. Burkhardt: *Domestic Oil Burners, Installation and Servicing* (McGraw-Hill Book Co., New York).

A. H. Senner: *A Study of the Oil Burner as Applied to Domestic Heating (U. S. Department of Agriculture Technical Bulletin* 109).

R. J. Bender: Progress in domestic oil heating (*Mechanical Engineering*, October 1942).

L. E. Seeley and E. J. Tavanlar: ASHVE Research Report No. 907—Study of performance characteristics of oil burners and low pressure heating boilers (ASHVE Transactions, Vol. 37, 1931, p. 517).

L. E. Seeley and J. H. Powers: ASHVE Research Report No. 925—A study of intermittent operation of oil burners (ASHVE Transactions, Vol. 38, 1932, p. 317).

L. E. Seeley, J. H. Powers, and E. J. Tavanlar: Air supply and its effect on performance of oil burners and heating boilers (ASHVE Transactions, Vol. 39, 1933, p. 75).

L. E. Seeley and E. J. Tavanlar: Study of fuel burning rates and power requirements of oil burners in relation to excess air (ASHVE Transactions, Vol. 40, 1934, p. 319).

D. W. Nelson: Oil burning in residences (ASHVE Transactions, Vol. 41, 1935, p. 355).

R. C. Cross and W. R. Lyman: A study of oil-fired heating boilers (*Heating and Ventilating*, October 1931).

Gaseous Fuels (American Gas Association).

Comfort Heating (American Gas Association).

American Standard Approval Requirements for Central House Heating Gas Appliances (American Standards Association, ASA Z21.13).

Steam and Hot Water Boilers (Vol. I, Z21.13.1-1956 with addenda Z21.13.1a-1957).

Gravity and Forced Air Central Furnaces (Vol. II, Z21.13.2-1956 with addenda Z21.13.2a-1957).

Gravity and Fan Type Floor Furnaces (Vol. III, Z21.13.3-1956 with addenda Z21.13.3a-1957).

Gravity and Fan Type Vented Recessed Heaters (Vol. IV, Z21.13.4-1955 with addenda Z21.13.4a-1956 and Z21.13.4b-1957).

American Standard Requirements for Installation of Domestic Gas Conversion Burners (American Standards Association, Z21.8-1948).

American Standard Requirements for Installation of Gas Equipment in Large Boilers (American Standards Association, Z21.33-1950).

American Standard for Installation of Gas Piping and Gas Appliances in Buildings (American Standards Association, Z21.30-1954).

American Standard Approval Requirements for Gas Unit Heaters (ASA Z21.16-1957).

American Standard Approval Requirements for Gas-Fired Duct Furnaces (ASA Z21.34-1955, with addenda Z21.34a-1956 and addenda Z21.34b-1957).

American Standard Approval Requirements for Gas-Fired Room Heaters (ASA Z21.11-1956, with addenda Z21.11a-1957).

A.G.A. Industrial Gas School Lectures (American Gas Association, 1951).

J. D. Keller: Radiant gas burners (*A.G.A. Sales Conference*, American Gas Association, 1949).

W. Trinks: *Industrial Furnaces* (Vol. 1 and 2, John Wiley & Sons, New York, 1944).

R. C. Cross: A method for determining fuel burning rates in heating boilers fired by automatic devices (*Heating and Ventilating*, January 1932).

R. A. Sherman and R. C. Cross: Heat losses and efficiencies of fuels in residential heating (ASHVE Transactions, Vol. 43, 1937, p. 185).

CHAPTER 35

HEATING BOILERS, FURNACES, SPACE HEATERS

BOILERS: Construction, Types, Design Considerations, Rating and Testing Codes, Efficiency, Rating, Selection, Space Limitations, Connections and Fittings, Erection, Operation and Maintenance. FURNACES: Types, Materials and Construction, Ratings, Testing and Rating Codes, Efficiency, Design Considerations, Humidification Equipment. SPACE HEATERS: Types: Solid Fuel, Oil, Gas; Materials and Construction, Testing and Rating, Design Considerations, Installation

IN presenting the subject of Boilers, Furnaces, and Space Heaters this chapter is divided into three parts; the first dealing with boilers, the second treating warm air furnaces, and the third covering space heaters.

HEATING BOILERS

Steam and hot water boilers for low-pressure heating are built of steel or cast iron in a wide variety of types and sizes, many of which are illustrated in the Catalog Data Section.

The nationally recognized code governing the construction of low-pressure steel and cast-iron heating boilers is the *ASME Boiler Construction Code for Low Pressure Heating Boilers*. Some states and municipalities have their own codes which apply locally, but these are usually patterned after the ASME Code.

The maximum allowable working pressures are limited by the *ASME Code for Low Pressure Heating Boilers* to 15 psi for steam and 160 psi for hot water heating boilers, with a maximum temperature limitation of 250 F. Hot water boilers are generally designed for a working pressure of 30 psi and may be supplied for higher working pressures, for either heating purposes or for hot water supply, only when designed, tested, and stamped for the higher pressure.

TYPES OF HEATING BOILERS

Heating boilers are classified in a number of different ways, such as:

1. According to materials of construction. These are steel and cast iron. Very few nonferrous boilers are made.

2. According to the fuels for which the boilers are designed. These are coal, hand-fired or stoker-fired; oil; gas; or wood. Some boilers are designed specifically for one fuel, but many boilers are designed for more than one fuel.

3. According to the specific purpose or application for which the boiler is used, such as space heating or domestic hot water supply.

4. According to the design or construction of the boiler such as sectional, round, fire-tube, water-tube, magazine-feed, Scotch, *etc.*

Cast-Iron Boilers

Cast-iron boilers are generally classified as:

1. Square or rectangular boilers with vertical sections and rectangular grates, commonly known as sectional boilers.

2. Round boilers with horizontal pancake sections and circular grates.

Cast-iron boilers are usually shipped in sections, and assembled at the place of installation. However, some small boilers are shipped factory assembled. In the majority of boilers the sections are assembled with push nipples and tie rods but external headers and screw nipples are also used successfully. Many sectional boilers are provided with large push nipples at top to permit the circulation of water between adjacent sections at both the water line and bottom of the boiler, which is necessary to enable the use of an indirect water heater with the boiler for summer-winter hot water supply. Round and sectional boilers may be increased in size by the addition of sections and corresponding plate work.

Small sectional type boilers are available with *wet-base construction*, wherein the ashpit or combustion chamber sides and bottom are surrounded by extensions of the water legs of the boiler sections, and thus no separate base is required. This type of construction permits the boiler to be set directly on a wood or composition floor without danger of fire. The *wet-base* also provides some additional heating surface.

Capacities of cast-iron boilers range generally from capacities required for small residences up to about 12,000 sq ft of steam radiation but boilers are made with capacities up to approximately 32,000 sq ft EDR (steam).

Steel Boilers

Steel boilers may be of the fire-tube type, in which the gases of combustion pass through the tubes and the boiler water circulates around them, or of the water-tube type, in which the gases circulate around the tubes and the water passes through them. Either the fire-tube or water-tube type may be designed with integral water-jacketed furnaces, or arranged for refractory lined brick or refractory lined jacketed furnaces. Those with integral water-jacketed furnaces are called portable firebox boilers, and are the most commonly used type. They are usually shipped in one piece, ready for piping connections. Refractory furnaces are usually installed in refractory lined furnace boilers after they are set in place.

Capacities of steel boilers range from those required for small residences up to about 50,000 sq ft EDR (steam).

Boilers for Special Applications

One of these is known as the *magazine-feed boiler* developed for the burning of small sizes of anthracite and coke, and has a large fuel carrying capacity, which results in longer firing periods than would be the case with the standard types burning coal of buckwheat size. Special attention must be given to proper chimney sizes and connections in order to insure adequate draft.

Boilers for *hot water supply* are classified as direct, if the water heated passes through the boiler, and as indirect, if

the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains, and are tested at pressures from 200 to 300 psig. The life of direct heaters depends almost entirely on the scale-forming properties of the water supplied and the temperatures maintained. If low water temperatures are maintained, the life of the heater will be much longer due to decreased scale formation and minimized corrosion. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with heat exchangers of the coil or tube types which transmit the heat from the steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.
2. The boiler is protected from scale and corrosion.
3. The scale is formed in the heat exchanger in which the parts to which the scale is attached can be cleaned or replaced. The accumulation of scale does not affect efficiency, although it will affect the capacity of the heat exchanger.
4. Discoloration of water may be prevented if the water supply comes in contact with only nonferrous metal.

Where a steam or a forced circulation hot water heating system is installed, the domestic hot water may be heated by an indirect heater attached to the boiler. For most satisfactory performance in the steam system, this heater is placed just below the water line of the boiler. In a forced circulation hot water system, it should be located as high as practicable with respect to the boiler.

BOILER DESIGN CONSIDERATIONS

Furnace Design

Good efficiency and proper boiler performance are dependent on correct furnace design. There must be sufficient volume for burning the particular fuel which is used, and means to obtain a thorough mixing of air and gases at a high temperature and at a velocity low enough to permit complete combustion of all the volatiles. For hand-fired boilers, the furnace volume should be large enough to hold sufficient fuel for reasonably long firing periods. (See Chapters 33 and 34.)

Heating Surface

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. Heating surface on which the fire shines, is known as *direct* or radiant surface, and that in contact with hot gases only, as *indirect* or convection surface. The amount of heating surface, its distribution, and the temperatures on either side thereof, influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. The area of the gas passages must not be so small as to cause excessive resistance to the flow of gases, where natural draft is employed. Inserting baffles so that the heating surface is arranged in series with respect to the gas flow, increases boiler efficiency and reduces stack temperature, but increases the draft loss through the boiler.

Heat-Transfer Rate

Practical average overall heat transfer rates, expressed in Btu absorbed per square foot of surface per hour, will average about 3300 for hand-fired boilers, and 4000 for mechanically-fired boilers when operating at *design load*. When mechanically-fired boilers are operating at *maximum load*, as defined in this chapter under heading Selection of Boilers, these values will run between 5000 and 6000. Boilers operating under favorable conditions at these heat transfer rates, will give exit gas temperatures that are considered consistent with good practice, although there are boilers which have high efficiencies and also operate at higher transmission rates.

RATING AND TESTING CODES

Heating boilers are usually rated according to codes developed by the Steel Boiler Institute, the Institute of Boiler and Radiator Manufacturers, the American Gas Association, and the Mechanical Contractors Association of America, whose test codes have been prepared specifically for the purpose of obtaining information required for establishing acceptable ratings. The various current rating and testing codes will be discussed in following paragraphs in which limitations, methods of application, and resulting ratings will be described.

The Steel Boiler Institute has adopted a rating code for boilers designated *Table 1 Steel Boilers*, *Table 2 Steel Boilers*, and *Table 3 Steel Boilers*, and a testing code for *Oil-fired Table 2* and *Table 3 Steel Boilers* (*SBI Rating Code*, Eighth Edition, February 1958). Tables 1, 2, and 3 of this chapter show the SBI Gross Outputs and Net Ratings of these three classes of boilers.

The *Table 1* boilers (defined as those having 129 to 3571 sq ft of heating surface and formerly designated as *Commercial* boilers) are rated in square feet EDR design load (steam) on the basis of heating surface with limitations set for grate area, furnace volume, and furnace height. The *Table 2* boilers (defined as those having not more than 294 sq ft of heating surface and formerly designated as *Residential* boilers) are rated on the basis of heating surface with ratings confirmed by test. The *Table 3* boilers (defined as having no limits of minimum heating surface or furnace volume, and having Gross Outputs and Net Ratings listed in Table 3) are rated on the basis of performance under the provisions of Section III, Part 2, of the *SBI Rating Code*.

The minimum stack area dimensions to be cataloged for *Table 3* boilers are shown in Table 4. Required stack heights are found in the catalogs of the manufacturers.

Stoker-fired and gas-fired *Table 2* boilers are rated (SBI Net Rating) not in excess of the oil-fired rating. Hand-fired *Table 2* boilers are rated (SBI Net Rating) not greater than fourteen times the heating surface.

The Institute of Boiler and Radiator Manufacturers has adopted a code[1] for rating cast-iron heating boilers, based upon performance obtained under controlled test conditions. This code applies to all sectional cast-iron heating boilers except those of magazine-feed type.

The *Gross I = B = R Output* is obtained by test, and is subject to certain limiting factors. For hand-fired boilers, the number of boilers of a series to be tested, the minimum overall efficiency, the minimum time limit (the time an *Available Fuel Charge* will last when burned at a rate which will produce the *Gross I = B = R Output*), the chimney area and

Table 1 . . . . SBI Ratings for Table 1 Steel Boilers

Mechanically-Fired									Hand-Fired							(a) Tappings	
SBI Rating			SBI Net Rating			SBI Gross Output Mbh	(c) Minimum Furnace Volume Cu Ft	(b) Minimum Furnace Height In.	Minimum Heating Surface Sq Ft	SBI Rating			SBI Net Rating			(c) Minimum Grate Area Sq Ft	Steam Outlet IPS In.	(c) Steam Return IPS In.
Sq Ft Steam	Sq Ft Water	Mbh	Sq Ft Steam	Sq Ft Water	Mbh					Sq Ft Steam	Sq Ft Water	Mbh	Sq Ft Steam	Sq Ft Water	Mbh			
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
2,190	3,500	526	1,800	2,880	432	648	15.7	26	129	1,800	2,880	432	1,500	2,400	360	7.9	6	3*
2,080	4,280	643	2,200	3,520	528	792	19.2	28	158	2,200	3,520	528	1,830	2,930	439	8.9	6	3*
3,160	5,030	758	2,600	4,180	624	936	22.6	29¼	186	2,600	4,180	624	2,170	3,470	521	9.7	6	3*
3,650	5,840	876	3,000	4,800	720	1,080	26.1	29½	215	3,000	4,800	720	2,500	4,000	600	10.6	6	4*
4,250	6,800	1,020	3,500	5,600	840	1,260	30.4	30	250	3,500	5,600	840	2,920	4,670	700	11.4	6	4*
4,860	7,770	1,166	4,000	6,400	960	1,440	34.8	30½	286	4,000	6,400	960	3,330	5,330	800	12.2	6	4*
5,470	8,760	1,313	4,500	7,200	1,080	1,620	39.1	31¼	322	4,500	7,200	1,080	3,750	6,000	900	13.4	6	4*
6,080	9,720	1,459	5,000	8,000	1,200	1,800	43.5	31¾	358	5,000	8,000	1,200	4,170	6,670	1,000	14.5	6	4*
7,290	11,660	1,750	6,000	9,600	1,440	2,160	52.1	32¾	429	6,000	9,600	1,440	5,000	8,000	1,200	16.4	8	4*
8,500	13,600	2,040	7,000	11,200	1,680	2,520	60.8	34	500	7,000	11,200	1,680	5,830	9,330	1,400	18.1	8	4*
10,330	16,250	2,479	8,600	13,600	2,040	3,060	73.8	35½	608	8,500	13,600	2,040	7,080	11,330	1,700	20.5	8	4
12,150	19,440	2,916	10,000	16,000	2,400	3,600	86.8	37½	715	10,000	16,000	2,400	8,330	13,330	2,000	22.5	8	4
15,180	24,280	3,463	12,500	20,000	3,000	4,500	108.5	40½	893	12,500	20,000	3,000	10,420	16,700	2,500	25.6	8	4
18,220	29,150	4,373	15,000	24,000	3,600	5,400	130.2	43	1,072	15,000	24,000	3,600	12,500	20,000	3,000	28.4	8	4
21,250	34,000	5,100	17,500	28,000	4,200	6,300	151.8	46	1,250	17,500	28,000	4,200	14,580	23,330	3,500	30.9	8	4
24,290	38,860	5,830	20,000	32,000	4,800	7,200	173.5	48½	1,429	20,000	32,000	4,800	16,670	26,670	4,000	33.2	8	4
30,360	48,750	7,286	25,000	40,000	6,000	9,000	216.9	54½	1,786	25,000	40,000	6,000	20,830	33,330	5,000	37.4	10	6
36,430	58,280	8,743	30,000	48,000	7,200	10,800	260.3	60½	2,143	30,000	48,000	7,200	25,000	40,000	6,000	41.2	10	6
42,500	68,000	10,200	35,000	56,000	8,400	12,600	303.6	66	2,500	35,000	56,000	8,400	29,170	46,670	7,000	44.7	10	6
48,570	77,710	11,657	40,000	64,000	9,600	14,400	346.9	71½	2,857	40,000	64,000	9,600	33,330	53,300	8,000	48.0	12	6
54,640	87,420	13,114	45,000	72,000	10,800	16,200	390.3	77½	3,214	45,000	72,000	10,800	37,500	60,000	9,000	51.0	12	6
60,710	97,140	14,570	50,000	80,000	12,000	18,000	433.6	83½	3,571	50,000	80,000	12,000	41,670	66,700	10,000	54.0	12	6

(a)* The return tapping sizes shown for boilers having 129 to 600 sq ft heating surface, inclusive, are adequate for forced hot water.

(b) Bituminous Stoker-Fired. Does not apply to Scotch-Type Boilers.

(c) Oil, gas or bituminous stoker-fired coal. Minimum furnace volumes for anthracite, stoker-fired, are not specified in this code.

(d) Does not apply to Scotch-Type Boilers.

(e) Does not apply to high-pressure boilers.

Note: Scotch-Type Boilers may be cataloged in sizes having 60 sq ft minimum heating surface with 7.2 cu ft minimum furnace volume, and 70 sq ft minimum heating surface with 11.9 cu ft minimum furnace volume.

Table 2 .... SBI Net Rating Data for *Table 2* Steel Boilers—Oil-Fired[a]

SBI Net Rating			SBI Gross Output Btuh	Minimum Furnace Volume Cu Ft	Minimum Heating Surface Sq Ft
Steam		Water Btuh			
Sq Ft	Btuh				
(1)	(2)	(3)	(4)	(5)	(6)
275	66,000	74,000	99,000	2.5	16
320	77,000	87,000	115,000	2.9	19
400	96,000	108,000	144,000	3.6	24
550	132,000	149,000	198,000	5.0	32
700	168,000	189,000	252,000	6.4	41
900	216,000	243,000	324,000	8.2	53
1100	264,000	297,000	396,000	10.0	65
1300	312,000	351,000	468,000	11.8	77
1500	360,000	405,000	540,000	13.6	88
1800	432,000	486,000	648,000	16.4	106
2200	528,000	594,000	792,000	20.0	129
2600	624,000	702,000	936,000	23.6	153
3000	720,000	810,000	1,080,000	27.3	177
3500	840,000	945,000	1,260,000	31.8	206
4000	960,000	1,080,000	1,440,000	36.3	236
4500	1,080,000	1,215,000	1,620,000	40.9	265
5000	1,200,000	1,350,000	1,800,000	45.4	294

[a] Stoker-Fired and Gas-Fired SBI Net Rating not greater than oil-fired rating. Hand-Fired Net SBI Rating (Steam) not greater than 14 times the square feet of heating surface.

Table 3 .... SBI Gross Output and SBI Net Rating Data for *Table 3* Steel Boilers

SBI Gross Output Btuh	SBI Net Ratings		
	Water Btuh	Steam Btuh	Steam Sq Ft
99,000	74,000	66,000	275,000
115,000	87,000	77,000	320,000
144,000	108,000	96,000	400,000
198,000	149,000	132,000	550,000
252,000	189,000	168,000	700,000
324,000	243,000	216,000	900,000
396,000	297,000	264,000	1,100,000
504,000	378,000	336,000	1,400,000
648,000	486,000	432,000	1,800,000
792,000	594,000	528,000	2,200,000
1,008,000	756,000	672,000	2,800,000
1,260,000	945,000	840,000	3,500,000
1,512,000	1,134,000	1,008,000	4,200,000
1,800,000	1,350,000	1,200,000	5,000,000

Table 4 .... Minimum Stack Dimensions for *Table 3* Oil-Burning Boilers and Boiler-Burner Units

Rectangular			Round	
Firing Rate Not Over Gph	Nominal Dimensions Inches	Inside Dimensions of Liner Inches	Firing Rate Not Over Gph	Inside Dia of Liner Inches
2.1	8 × 8	6¾ × 6¾	1.3	6
3.5	8 × 12	6¾ × 10¾	1.8	7
5.3	12 × 12	9¾ × 9¾	2.5	8
7.7	12 × 16	9¾ × 13¾	4.7	10
11.5	16 × 16	13¾ × 13¾	5.6	11
15.0	16 × 20	13 × 17	7.0	12
21.0	20 × 20	16¾ × 16¾	12.0	15
26.0	20 × 24	16¾ × 20¾	19.0	18

height, and the draft in the stack are all subject to the limits established in the code. Tests are run using anthracite coal of standard specification. Bituminous coal and coke ratings are the same as for anthracite.

For automatically-fired boilers, the number of boilers of a series to be tested, the flue gas temperature and analysis, the minimum overall efficiency, the draft loss through the boiler, and the heat release in the combustion chamber are subjected to limitation by the code.[1] Oil-fired boiler ratings are established by tests using gun-type oil burners and commercial grade No. 2 fuel oil. Stoker-fired ratings are based on the *Gross I = B = R Output* obtained by oil-fired tests. Gas-fired ratings for boilers having gross outputs greater than 300,000 Btu per hour, where no *A.G.A. Rating* is published, are based on the *Gross I = B = R Output* obtained by oil-fired tests. Gas-fired ratings for boilers having gross outputs less than 300,000 Btu per hour, where no *A.G.A. Rating* is published, are established by gas-fired tests with atmospheric, single-port-type burners using natural or manufactured gas or a mixture of these two gases. The *A.G.A. Output Rating* of a gas-designed boiler may be used in determining the *Net I = B = R Rating*.

The *Net I = B = R Rating* is determined from the *Gross I = B = R Output* by applying specified *Piping and Pickup Factors* which range from 2.36 to 1.40 for hand-fired boilers, and from 1.56 to 1.288 for automatically-fired steam boilers, and from 1.333 to 1.288 for automatically-fired hot water boilers. In all cases, the factor decreases as the boiler size increases. Table 5 is abstracted from the 1951 I = B = R Boiler Rating Tables in the code and illustrates the relationship between *Net I = B = R Rating* and *Gross I = B = R Output*.

The American Gas Association rates gas designed boilers at 80 percent of the A.G.A. *Input Rating*. These ratings are determined by *performance tests* described in the *A.G.A. Approval Requirements for Central Heating Appliances*.

The Mechanical Contractors Association of America has adopted a method, based on their physical characteristics, for rating boilers that are not rated in accordance with the SBI or I = B = R Codes. Ratings are expressed on a Net Load basis in square feet of steam radiation.

The MCAA has also adopted a *Testing and Rating Code for Boiler-Burner Units*.[2] The purpose of this Code is to provide a basis of rating commercial sizes of steel heating boiler-burner units fired with oil or gas fuel. This code allows a higher rating than is permissible under the SBI Code. A gross output is established with certain limiting factors applying to flue gas temperature, carbon dioxide, efficiency, and quality of steam. This output is divided by 1.5 to determine the *net rating*.

BOILER EFFICIENCY

The term efficiency, as used for guarantee of boiler performance, is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate, to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace, and grate* is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired, to the calorific value of 1 lb of fuel as fired.

2. *Liquid and Gaseous Fuels.* The *combined* efficiency of boiler, furnace, and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound or cubic foot of fuel, to the calorific value of 1 lb or cubic foot of fuel, respectively.

The following efficiencies apply to current designs of boilers operated under favorable conditions at their gross output ratings. Some older boilers, designed primarily for hand firing, may have lower efficiencies when automatically-fired.

Anthracite, hand-fired	60 to 75 percent
Bituminous coal, hand-fired	50 to 65 percent
Stoker-fired	60 to 75 percent
Oil- and gas-fired	70 to 80 percent

Higher efficiencies for hand-fired bituminous coal may be obtained by careful firing of either a regular or a smokeless boiler.

RATING OF BOILERS

In referring to boiler rating, it is necessary to know the basis on which the rating has been established in order to understand the exact meaning of the term. The following example will illustrate the meaning of three ratings which might be established for the same boiler.

Assume that an installation has the following loads determined in accordance with the section Selection of Boilers:

Net Load	1000 sq ft of steam radiation
Piping Tax	200 sq ft of steam radiation
Design Load	1200 sq ft of steam radiation
Pickup Allowance	240 sq ft of steam radiation
Maximum or Gross Load	1440 sq ft of steam radiation

Table 5 .... I = B = R Boiler Rating Table[a]

	Hand-Fired						Automatic-Fired							
Gross I=B=R Output	Net I=B=R Rating		Piping and Pickup Factor	Time Available Fuel Will Last,	Maximum Stack Height[c]	Minimum Stack Area[b]	Gross I = B = R Output	Net I=B=R Rating			Piping and Pickup Factor		Minimum Stack Area[b]	Maximum Allowable Draft Loss
	Steam and water	Steam						Water	Steam		Water	Steam		
MBh	MBh	Sq Ft		Hr	Ft	Sq In.	MBh	MBh	MBh	Sq Ft			Sq In.	In. W.G.[b]
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
57	24	100	2.360	7.50	29.0	50	37	28	24	100	1.333	1.560	50	0.044
218	96	400	2.268	6.32	36.5	50	146	110	96	400	1.333	1.525	50	0.058
359	168	700	2.139	5.73	42.0	54	251	188	168	700	1.333	1.492	50	0.072
489	240	1000	2.039	5.32	46.0	82	352	264	240	1000	1.333	1.466	52	0.084
614	312	1300	1.967	4.99	49.0	110	451	338	312	1300	1.333	1.444	73	0.096
735	384	1600	1.914	4.73	52.0	135	547	410	384	1600	1.333	1.424	95	0.108
854	456	1900	1.872	4.49	54.5	160	642	482	456	1900	1.333	1.408	115	0.118
970	528	2200	1.837	4.34	56.5	185	736	552	528	2200	1.333	1.394	135	0.130
1083	600	2500	1.805	4.24	59.0	207	829	622	600	2500	1.333	1.382	155	0.141
1192	672	2800	1.774	4.15	61.0	231	920	690	672	2800	1.333	1.369	173	0.152
1298	744	3100	1.744	4.07	63.0	253	1011	758	744	3100	1.333	1.359	192	0.162
1401	816	3400	1.717	4.00	64.5	275	1101	826	816	3400	1.333	1.349	211	0.172
1502	888	3700	1.691	4.00	66.5	296	1189	892	888	3700	1.333	1.339	230	0.183
1599	960	4000	1.666	4.00	68.0	315	1278	960	960	4000	1.331	1.331	249	0.192
1726	1056	4400	1.634	4.00	70.0	337	1394	1056	1056	4400	1.320	1.320	274	0.206
1849	1152	4800	1.605	4.00	71.5	358	1509	1152	1152	4800	1.310	1.310	297	....
1968	1248	5200	1.577	4.00	73.0	376	1624	1248	1248	5200	1.301	1.301	318	....
2086	1344	5600	1.552	4.00	74.5	393	1739	1344	1344	5600	1.294	1.294	339	....
2197	1440	6000	1.528	4.00	76.0	409	1858	1440	1440	6000	1.290	1.290	359	....
2307	1536	6400	1.502	4.00	77.5	425	1978	1536	1536	6400	1.288	1.288	377	....
2415	1632	6800	1.482	4.00	79.0	438	2102	1632	1632	6800	1.288	1.288	395	....
2470	1680	7000	1.470	4.00	79.5	446	2164	1680	1680	7000	1.288	1.288	405	....
2605	1800	7500	1.447	4.00	81.5	464	2318	1800	1800	7500	1.288	1.288	426	....
2738	1920	8000	1.426	4.00	83.0	481	2473	1920	1920	8000	1.288	1.288	446	....
2874	2040	8500	1.409	4.00	85.0	497	2628	2040	2040	8500	1.288	1.288	467	....
3024	2160	9000	1.400	4.00	87.0	515	2782	2160	2160	9000	1.288	1.288	486	....
3192	2280	9500	1.400	4.00	89.0	535	2937	2280	2280	9500	1.288	1.288	504	....
3360	2400	10000	1.400	4.00	91.5	555	3091	2400	2400	10000	1.288	1.288	522	....
4032	2880	12000	1.400	4.00	99.0	634	3709	2880	2880	12000	1.288	1.288	596	....
4704	3360	14000	1.400	4.00	106.5	712	4328	3360	3360	14000	1.288	1.288	668	....
5376	3840	16000	1.400	4.00	112.5	789	4946	3840	3840	16000	1.288	1.288	740	....
6048	4320	18000	1.400	4.00	118.0	863	5564	4320	4320	18000	1.288	1.288	810	....
6720	4800	20000	1.400	4.00	120.0	900	6182	4800	4800	20000	1.288	1.288	877	....

[a] Extracted from Rating Table in 1951 edition of *I = B = R Testing and Rating Code for Low Pressure Cast Iron Heating Boilers.*
[b] To be specified in catalog. [c] Maximum stack height which may be shown in catalog.

A boiler that is just large enough to carry this system might be said to have a *net load* rating of 1000 sq ft, a *design load* rating of 1200 sq ft, or a *gross load* rating of 1440 sq ft, depending on the basis on which the boiler is rated.

On a *net load* basis the boiler would be rated 1000 sq ft of steam radiation and would have sufficient excess capacity to supply the *normal piping* and *pickup load*. Net I = B = R Ratings, SBI Net Ratings, and Net Load Ratings of the Mechanical Contractors Association of America are established on this basis.

On a *design load* basis the boiler would be rated 1200 sq ft of steam radiation and would have sufficient excess capacity to supply the *pickup load*. It would be of adequate size for a system in which the *sum* of the *net load* and the *piping heat loss* did not exceed 1200 sq ft of steam radiation. The SBI Ratings shown in columns 1, 2, 3, 10, 11, and 12 of Table 1 (not to be confused with SBI Net Rating) are established on a *design load* basis.

On a *gross output* basis of rating, the boiler would be rated 1440 sq ft of steam radiation and would be of adequate size for a system in which the sum of the *net load*, *piping load*, and *pickup load* did not exceed 1440 sq ft of steam radiation. Gross I = B = R Output and A.G.A. Ratings are established on a *gross output* basis.

In the determination of boiler ratings, the *Gross Output* is the quantity of heat available at the boiler nozzle, with the boiler normally insulated and when operating under limitations stipulated in the code or method by which the boiler is rated. The boiler may be capable of producing a greater nozzle output, but in doing so would exceed some of these limitations.

SELECTION OF BOILERS

General Factors

The *Maximum Load* or *Gross Load* on the boiler is the sum of the four following items.

The *Design Load* is the sum of items 1, 2, and 3.

The *Net Load* is the sum of items 1 and 2.

1. *Radiation Load*. The estimated heat emission in Btu per hour of the connected radiation (direct, indirect, or forced convection coils) to be installed.

The connected radiation is determined by calculating the heat losses for each room in accordance with data given in Chapters 9, 11, and 12. The sum of the calculated heat losses for all the rooms represents the total required heat emission of the connected radiation, expressed in Btu per hour. As practically all boilers are now rated on a Btu basis, it is unnecessary to convert the radiation load to square feet of equivalent direct radiation.

2. *Hot Water Supply Load*. The estimated maximum heat in Btu per hour required to heat water for domestic use.

I = B = R recommends that allowance for hot water supply load be made only for bathrooms in excess of two, as follows: Instantaneous Coil 12,000 Btu per hour, and for Storage Tank installation 120 Btu per (hour) (gallon of tank capacity). For instantaneous coil installations the boiler capacity should not be less than required to heat 2 to 3 gal of water, 100 deg per min. See also Chapter 56.

3. *Piping Tax*. The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

As the heating industry as a whole is not entirely agreed upon piping tax allowances for different sizes of installations, it is better to compute the heat emission from both bare and covered pipe surface in accordance with data in Chapter 32. In average house heating systems, it is common practice to consider the piping tax to be equal to 25 percent of the Net Load. In determining Net I = B = R Ratings from Gross I = B = R Output, the piping factor allowed varies from 30 percent for small boilers to 12 percent for larger boilers.

4. *Warming-Up or Pickup Allowance*. The estimated increase in the normal load in Btu per hour caused by the heating up of the cold system.

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature, and heating up cold radiators and piping.

Other items to be considered in boiler selection are as follows:

a. Efficiency with hard or soft coal, gas, or oil firing, as the case may be.

b. Grate area with hand-fired coal, or fuel-burning rate with stokers, oil, or gas.

c. Combustion space in the furnace.

d. Type of heat liberation, whether continuous or intermittent, or a combination of both.

e. Convenience in firing and cleaning.

f. Adaptability to changes in fuel and kind of attention.

g. Height of water line.

h. Miscellaneous items such as draft available, possibility of future extension, possibility of breakdown, and head room in the boiler room.

i. The most economical size of boiler is usually one that is just the right size for the load. Either larger or smaller boilers may be less economical.

Cast-Iron Boilers

Net load ratings of cast-iron boilers are usually available from manufacturers' catalogs. They may also be obtained conveniently from published tables of I = B = R ratings,[3] or from recommendations of the Mechanical Contractors Association of America,[4] and can be used in selection of boilers, unless the heating system contains an unusual amount of bare pipe, or the nature of the connected load is such that the normal allowances for pipe loss and pickup do not apply. In such a case, the selection must be based on the gross output.

Steel Heating Boilers

SBI catalog ratings, in accordance with the previously mentioned Steel Boiler Institute code, are intended to correspond with the estimated *design load*. When the heat emission of the piping is not known, the net load to be considered for the boiler may be determined from Tables 1 and 2. The difference between design load and net load represents an amount which is considered normal for piping loss of the ordinary heating system.

Boilers with less than 294 sq ft of heating surface and having SBI net ratings (steam) of not more than 5000 sq ft if mechanically-fired and 4120 sq ft if hand-fired, are classified as *Table 2 Steel Boilers*, formerly Residential boilers. An insulated *Table 2 boiler* for oil, gas, or stoker firing may carry a *net load* expressed in square feet of steam radiation of not more than seventeen times the square feet of heating surface in the boiler, provided the boiler has been tested in accordance with the SBI Code for testing oil-fired steel boilers at output rates of 125, 150, and 175 percent of the steam SBI net rating. The SBI net rating for a water boiler, automatically-fired, expressed in Btu per hour, is 270 times the steam SBI net rating in square feet. The net water rating is 12½ percent greater than the steam rating because a lower piping and pickup factor is generally required in a water system. The SBI net rating (sq ft steam) for hand-fired *Table 2 boilers* is not greater than fourteen times the heating surface. If the heat loss from the piping system exceeds 20 percent of the installed radiation, the excess is to be considered as part of the *net load*.

Gross Output ratings for *Table 1* and *Table 2*, as well as for *Table 3* (a new designation) *Steel Boilers* are shown in the seventh edition of the *SBI Rating Code*.

Heating Surface and Grate Area Basis

Where neither the *net load* nor *gross output* ratings based upon performance tests are available, a good general rule for conventionally designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per sq ft) (hr) represented by the design load. This is equivalent to allowing 10 sq ft of boiler heating surface per boiler horsepower. In this case it is assumed that the *maximum load* including the warming-up allowance will be provided for by operating the boiler in excess of the *design load*, that is, in excess of the 100 percent rating on a boiler-horsepower basis. SBI ratings for hand firing are based on 10 sq ft of heating surface per boiler horsepower.

Table 6 .... Practical Combustion Rates for Coal-Fired Heating Boilers Operating at Maximum Load on Natural Draft of From ⅛ In. to ½ In. Water*

Kind of Coal	Sq Ft Grate	Lb of Coal per (Sq Ft Grate) (Hr)
No. 1 Buckwheat Anthracite	Up to 4	3
	5 to 9	3½
	10 to 14	4
	15 to 19	4½
	20 to 25	5
Anthracite Pea	Up to 9	5
	10 to 19	5½
	20 to 25	6
Anthracite Nut and Larger	Up to 4	8
	5 to 9	9
	10 to 14	10
	15 to 19	11
	20 to 25	13
Bituminous	Up to 4	9.5
	5 to 14	12
	15 and above	15.5

* Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

Due to the wide variation which may be encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \qquad (1)$$

where

G = grate area, square feet.
H = required *gross output* of the boiler, Btu per hour (see Selection of Boilers).
C = desirable combustion rate for fuel selected, pounds of dry coal per (square foot of grate) (hour). (See Table 6.)
F = calorific value of fuel, Btu per pound.
E = efficiency of boiler, usually taken as 0.60.

Example 1: Determine the grate area for a required *gross output* of the boiler of 500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 percent.

$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

The boiler selected should have a grate area not less than that determined by Equation 1. With small boilers, where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 percent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Gas-Fired Boilers

After determining the *net load* for the installation, gas-designed boilers can usually be selected from manufacturers' tables of *net load ratings* which are based on piping and pickup allowances varying from 56 percent for small steam boilers and 33.3 percent for small hot water boilers to 28.8 percent for very large boilers. If the piping and pickup load or other factors create an unusual load, a boiler should be selected which has an *A.G.A. output rating* equal to the maximum output required. Detailed recommendations for selection of gas designed boilers are given in the A.G.A. publication, *Comfort Heating*.[5]

SPACE LIMITATIONS

Boiler rooms should, if possible, be situated at a central point with respect to the building, and should be designed for a maximum of natural light. The space in front of the boilers should be sufficient for firing, stoking, ash removal, and cleaning or renewal of flue tubes, and should be at least 3 ft greater than the length of the tubes.

A space of at least 3 ft should be allowed on at least one side of every boiler for convenience of erection and for accessibility to the various dampers, cleanouts, and trimmings. The space at the rear of the boiler should be ample for the chimney connection and for cleanouts. With large boilers the rear clearance should be at least 3 ft in width.

The boiler room height should be sufficient for the location of boiler accessories, and for proper installation of piping. In general, the ceiling height for small steam boilers should be at least 3 ft above the normal boiler water line. With vapor heating, especially, the height above the boiler water line is of vital importance.

CONNECTIONS AND FITTINGS

Steam outlet connections should be the full size of the manufacturers' tappings, in order to keep the velocity of flow through the outlet reasonably low, and to avoid fluctuation of the water line and undue entrainment of moisture, and should extend vertically to the maximum height available above the boiler. A steam velocity in boiler outlets not exceeding 25 to 30 fps at maximum load is recommended, unless data are available to show that a higher velocity is satisfactory. See further data on pipe connections to boilers in Chapters 26 and 28 and in the *ASME Boiler Construction Code for Low Pressure Heating Boilers*.

Where a *return header* is used on a cast-iron sectional boiler to distribute the condensate to both rear tappings, it is advisable to provide full size plugged tees instead of elbows where the branch connections enter the return tappings. This aids in cleaning of sludge from the bottom of the boiler sections through the large plugged openings. An equiva-

lent cleanout plug should be provided in the case of a single return connection.

Blowoff or drain connections should be made near the boiler, and so arranged that the entire system may be drained of water by opening the drain cock. In the case of two or more boilers separate blowoff connections must be provided for each boiler, on the boiler side of the stop valve on the main return connection.

Water service connections must be provided for both steam and water boilers, for refilling and for the addition of make-up water to boilers. This connection is usually of galvanized steel pipe, and is made to the return main near the boiler or boilers.

Fitting connections for pressure gage piping, water gage connections, and safety valves, should be made in accordance with the *ASME Boiler Construction Code for Low Pressure Heating Boilers*.

Smoke Breeching and Chimney Connections. The breeching or smoke pipe from the boiler outlet to the chimney should be airtight and as short and direct as possible, preference being given to long-radius and 45-deg instead of 90-deg bends. The breeching entering a brick chimney should not project beyond the flue lining, and where practicable it should be grouted from the inside of the chimney. A thimble or sleeve usually is provided where the breeching enters a brick chimney.

Where a battery of boilers is connected into a breeching, each boiler should be provided with a tight damper. The breeching for a battery of boilers should not be reduced in size as it goes to the more remote boilers. Good connections made to a good chimney will usually result in a rapid response by the boilers to demands for heat.

ERECTION, OPERATION, AND MAINTENANCE

The directions of the boiler manufacturer should always be read before the assembly or installation of any boiler is started, even though the contractor may be familiar with the boiler. All joints requiring boiler putty or cement, which cannot be reached after assembly is complete, must be finished as the assembly progresses.

Five precautions that should be taken in all installations to prevent damage to the boiler are:

1. There should be provided proper and convenient drainage connections for use if the boiler is not in operation during freezing weather.
2. Strains on the boiler, due to movement of piping during expansion, should be prevented by suitable anchoring of piping, and by proper provision for pipe expansion and contraction.
3. Direct impingement of too intense local heat upon any part of the boiler surface, as with oil burners, should be avoided by protecting the surface with firebrick or other refractory material.
4. Condensation in steam systems must flow back to the boiler as rapidly and uniformly as possible. Return connections should prevent the water from backing out of the boiler.
5. Automatic boiler feeders and low water cut-off devices, which shut off the source of heat if the water in the boiler falls below a safe level, are recommended for mechanically-fired boilers.

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft; (*b*) poor fuel; (*c*) inferior attention or firing; (*d*) boiler too small; (*e*) improper piping; (*f*) improper arrangement of sections; (*g*) heating surfaces covered with soot; (*h*) insufficient radiation installed; or (*i*) with mechanical firing, fuel-burning equipment too small.
2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler; (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; or (*c*) boiler operating at excessive rate of output.
3. *Water disappears from the gage glass.* This may be caused by: (*a*) priming due to grease and dirt in boiler; (*b*) too great pressure difference between supply and return piping preventing return of condensate; (*c*) valve closed in return line; (*d*) connection of bottom of water column to a very active section or thin waterway; or (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensate into boiler with lower pressure.
4. *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area; (*c*) outlet connections of too small area; (*d*) excessive rate of output; (*e*) water level carried higher than specified.
5. *Boiler is slow in response to operation of dampers.* This may be due to: (*a*) poor draft resulting from air leaks into chimney or breeching; (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; or (*e*) boiler too small for the load.
6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft; (*b*) smoky combustion; (*c*) too low a rate of combustion; or (*d*) too much excess air in firebox causing chilling of gases.
7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers; (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel; (*d*) dirty or clogged flues; or (*e*) improper reduction in breeching size.
8. *Low carbon dioxide.* This may be due in oil burning boilers to: (*a*) improper adjustment of the burner; (*b*) leakage through the boiler setting; (*c*) improper fire caused by a fouled nozzle; or (*d*) to an insufficient quantity of oil being burned.

Cleaning Boilers

All boilers are provided with flue cleanout openings through which the heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil-burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom of the boiler and form sludge. These impurities tend to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a surface blow connection of at least 1¼-in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in this line close to boiler. Bring the water line to center of outlet, raise steam pressure, and while fire is burning briskly open valve in blowoff line. When pressure drops, close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 psi, close blowoff, draw the fire or stop burner, and open drain valve. After boiler has cooled partly, fill and flush out several times before filling it to proper water level for normal service.

Where the outlined blowing-off procedure does not remove the grease and dirt and obtain clear boiler water, it may be necessary to use a detergent type of cleaner in which case the boiler manufacturer as well as cleaner manufacturer should be consulted regarding procedure to be followed.

When addition of water to a system becomes necessary it is advisable to operate the boiler or at least to raise the water temperature to the boiling point in order to permit escape of oxygen and other dissolved gases that would be corrosive if the boiler is out of operation.

Treatment of Boiler Water

Corrosion on the water side of a boiler can be prevented by proper treatment of the boiler water. Generally, treatment which will remove the dissolved oxygen and carbon dioxide and maintain a minimum pH of 11 (slightly alkaline water) will effectively prevent corrosion. A pH higher than 11 may be used although the boiler water is more likely to foam and cause wet steam with the greater concentration of alkali. A number of compounds are being marketed for this purpose. The Steel Boiler Institute adopted a conditioning compound for use with steel boilers after an extensive cooperative study and many members of the Institute either supply this compound with their boilers or have it available.

It should be emphasized that when the character of the water is such that treatment is required, the treatment should be administered immediately after the boiler and system have been cleaned. It is not good practice to fill a boiler with fresh water and allow it to stand idle as the dissolved gases in fresh water available in many localities can start corrosive action in a very short time. After refilling a system it is advisable to elevate the temperature close to the boiling point to drive off the dissolved gases immediately.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulfur in the soot with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash, and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.
2. All machined surfaces should be coated with oil or grease.
3. Connections to the chimney should be cleaned, and in case of small boilers, the pipe should be placed in a dry place after cleaning.
4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard that someone may inadvertently build a fire in a dry boiler, however, it is safer to keep the boiler filled with water, particularly in residential installations. Air can be excluded from a steam boiler by raising the water level into the steam outlets. A hot water system usually is left filled to the expansion tank.
5. The grates and ashpit should be cleaned.
6. Clean and repack the gage glass if necessary.
7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper. After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted.
8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings, and regulator parts.
9. If a system has been drained and refilled with fresh water, the boiler should be fired long enough to elevate the water temperature close to the boiling point to drive off dissolved oxygen and carbon dioxide. These gases can be responsible for excessive corrosion if allowed to remain in the system.

WARM AIR FURNACES

Warm air heating furnaces of a number of types and a wide range of sizes are listed and illustrated in the Catalog Data Section.

Warm air furnaces may be classified in several different ways:

1. According to method of heat distribution—these are either gravity or mechanical (blower) furnaces.
2. According to fuels for which the furnaces are designed—these are coal hand-fired or stoker-fired, oil, gas, or wood.
3. According to materials of construction—they are cast iron, low carbon steel, and occasionally high temperature steel alloys.
4. According to design or construction, such as drum and radiator, tubular, horizontal, etc.

Gravity Warm Air Furnaces

A gravity furnace is one in which the motive head producing air flow depends upon the difference in density between the heated air leaving the top of the casing and cooled air entering the bottom of the casing. Since this gravity head is relatively low, the furnace must have low internal resistance to the flow of air, and relatively large areas must be available for free circulation within the furnace casing. It is common practice to provide approximately 50 percent free-air area through gravity-type furnaces.

Furnaces for gravity-type systems are available in designs suitable for central heating, pipeless furnace, or unit floor-furnace installations. Booster fans are sometimes used, in conjunction with gravity systems, to increase air circulation. Where a fan is to be used with a furnace casing sized for gravity air flow, some form of baffling must be employed to restrict the free area within the casing and to force impingement of the air against the heating surfaces. Where square casings are used, the corners must be baffled.

Mechanical Warm Air Furnaces

Mechanical or forced warm air furnaces include fans or blowers as integral parts, for the purpose of circulating the air, and usually include air filters.

Centrifugal fans with either backward- or forward-curved blades are the type most commonly used. Motors may be mounted on the fan shaft or connected to the fan by a belt drive. Adjustable pulleys are desirable to provide means of regulating the quantity of air distributed to the heated spaces. Either the motor load or the noise considerations may limit the maximum operating fan speed. Two-speed motors have given successful operating results. Motors and mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with, the fan housing.

Filters

Several types of filters are available for mechanical warm air furnace applications, and are discussed in Chapter 24. For maximum efficiency and life under operating conditions, filters should not be subjected to a temperature in excess of 150 F. Filters should have at least 80 percent average efficiency on an 8-hr test at a maximum resistance of 0.25 in. of water. Filter resistance rises rapidly with the accumulation of dirt, and may reduce the air circulation over heating surfaces. In domestic furnaces, the maximum velocity, based on nominal filter area, should not exceed 300 fpm.

Fuel Utilization

A combustion rate of from 5 to 8 lb of coal per (square foot of grate) (hour) is recommended for residential furnaces. A higher combustion rate is permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes the ratio may be as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of a number of furnaces, using one or more fans.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. Furnaces for burning oil fuel are usually designed for blow-through installations so that the pressure in the air space is higher than that in the combustion chamber or flues. The National Warm Air Heating and Air Conditioning Association has prepared a *Tentative Code for Testing and Rating of Oil-Fired Furnaces*. Compact fan-furnace-burner units are available, suitable for basement, closet, or attic installations.

Gas-fired forced air furnaces should conform in construction and performance to A.G.A. Approval Requirements.

Heavy Duty Fan Furnaces

Fan furnaces for large commercial and industrial buildings, churches, schools, etc., are available in sizes ranging from 300,000 to 6,000,000 Btu per (hour) (unit). Heavy duty furnace heaters may be arranged in battery combinations of one or more units.

Most manufacturers of heavy duty furnaces rate their furnaces in Btu per hour, and also indicate the number of square feet of heating surface. Limitations on temperature of flue gases, heat exchanger, and casing as well as carbon monoxide and air temperature rise are also factors in establishing ratings for these units. These limits are shown in the *ASHAE Code for Testing and Rating Heavy Duty Furnaces and Direct-fired Unit Heaters*.

Control of temperature is secured through (1) controlling the quantity of heated air entering the room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 19. Ducts are designed by the method outlined in Chapter 21.

MATERIALS AND CONSTRUCTION

Both cast-iron and steel furnaces are made in capacities ranging from those for small insulated residence application with inputs of 40,000 Btu per hour or less, to capacities as large as 600,000 Btu per hour.

Cast-iron furnaces are usually constructed with a minimum sectional thickness of ¼ in., and effectively resist high temperatures and corrosion. They usually have a fairly large heat capacity because of their mass, which provides a distinct *fly wheel* or carry-over heating effect.

In steel furnaces welding, riveting, or both are used to join the formed metal. The use of steel castings, however, is rare, because of the cost, and because high stresses are not encountered in normal furnace construction. Steel furnaces have low heat capacities as a result of their relatively low mass and, therefore, deliver heat rapidly on demand.

FURNACE RATING

Rating Equations for Gravity Warm Air Furnaces[6]

Until a method of testing and rating gravity warm air furnaces has been developed, the following empirical rating equations are recommended by the National Warm Air Heating and Air Conditioning Association.

Gravity warm air furnaces of conventional design, having ratios (of heating surface to grate area) of 15 to 1 or greater, and having a ratio of casing area to face area not less than 0.4, are rated by the following equations:

1. *Hand-fired furnaces converted to Stoker, Gas, or Oil Firing.*

Bonnet Capacity in Btu per hour $= 1785 \times S \times 1.333$ (2)

2. *Hand-fired furnaces, with ratios of heating surface to grate area greater than 15 to 1 and less than 25 to 1.*

Bonnet Capacity in Btu per hour $= 1785 \times S \times 1.333$ (3)

3. *Hand-fired furnaces with ratios of heating surface to grate area in excess of 25 to 1.*

Bonnet Capacity in Btu per hour $= 1785 \times 25 \times G \times 1.333$ (4)

where

S = heating surface, in square feet.
G = actual grate area, in square feet.

The *Register Delivery Rating* is equal to 0.75 × (*Bonnet Capacity*). The *Leader Pipe Rating* in square inches, formerly used as a rating unit, may be found by dividing the *Register Delivery Rating* by 136.

Heating Surface of Furnace

Prime heating surface is defined[6] as surface above the top of the grate having *hot gases or live fuel on one side and circulating air over the other*, and in all cases is measured on the exterior or air side. The areas of the outer casing, the inner liner, and any radiation shields shall not be considered as heating surface.

In determining the amount of heating surface, extended surfaces are considered to be prime heating surface subject to the following limitations:

1. Extended heating surface may consist of fins, ribs, webs, lugs, or other projections from the prime heating surface. Projections less than ¼ in. thick at the base, and extending more than 1 in. from the prime surface are classified as fins.

2. Integral fins are continuously welded to, or cast as a part of, the prime heating surface. Both sides are included as heating surface, subject to the following allowances:

Distance from prime surface	1st in.	2nd in.	3rd in.	Over 3 in.
Ratio of effective area to total area	0.40	0.30	0.20	None

3. Non-integral fins are spot welded to, or otherwise held in line contact with, the prime heating surface. Both sides are included as heating surface, subject to the following allowances:

Distance from prime surface	1st in.	2nd in.	3rd in.	Over 3 in.
Ratio of effective area to total area	0.30	0.20	0.15	None

4. In the case of ribs, webs, or lugs more than ¼ in. thick at the base and extending less than 1 in. from the prime surface, the entire surface in contact with circulating air is included as heating surface.

5. In the case of ribs, webs, or lugs more than ¼ in. thick at the base and extending more than 1 in. from the prime heating surface, the areas of both sides of the first inch are included as prime heating surface. The portions projecting beyond 1 in. are treated as integral fins.

Grate Area

Grate area is defined[6] and treated for purpose of rating as follows:

1. The *nominal grate area* is defined as the total cross-sectional area of the bottom of the firepot. In steel furnaces the nominal grate area is the cross-sectional area inside the firebrick lining.

2. The *actual grate area*, used for calculating the ratios of heating surface to grate area, is the nominal grate area minus certain areas that cannot be considered as part of the grate itself. The following rules govern these deductions: (1) If a solid, continuous ledge extends around the grate and inside the firepot, any area of this ledge extending inside of a circle, the diameter of which is 1 in. less than the diameter of the bottom of the firepot, shall be deducted. (2) If separate, solid projections extend from the firepot towards the grate, the areas of any portions of these projections extending inside of a circle, the diameter of which is 3 in. less than the diameter of the bottom of the firepot, shall be deducted. (3) In the case of grates which are inclined, or are conical, the projected area is the same as the nominal grate area. The latter should, therefore, be used after making any necessary deductions.

Ratings for Forced Air Furnaces

For solid-fuel-burning, forced air furnaces having bonnet capacities between 80,000 and 250,000 Btu per hour, no standard method of test has been accepted, although eventually such codes will be developed. The National Warm Air Heating and Air Conditioning Association recommends the following empirical equations for use in rating solid fuel forced air furnaces:

1. *Hand-fired furnaces converted to Stoker, Gas, or Oil Firing.*

Bonnet Capacity in Btu per hour $= 2265 \times S \times 1.177$ (5)

2. *Hand-fired furnaces, with ratios of heating surface to grate area greater than 15 to 1 and less than 25 to 1.*

Bonnet Capacity in Btu per hour $= 2265 \times S \times 1.177$ (6)

3. *Hand-fired furnaces with ratios of heating surface to grate area in excess of 25 to 1.*

Bonnet Capacity in Btu per hour $= 2265 \times 25 \times G \times 1.177$ (7)

where

S = heating surface, in square feet.
G = actual grate area, in square feet.

The *Register Delivery Rating* is equal to 0.85 × (*Bonnet Capacity*).

The following testing and rating codes have been generally accepted in the industry:

Commercial Standard CS-109-44 for rating solid-fuel-burning, forced air furnaces having bonnet outputs of 80,000 Btu per hour or less. This provides a method of rating small coal-fired forced air furnaces by test.

A Tentative Code for Testing Oil-Fired Furnaces. This code has been adopted by the National Warm Air Heating and Air Conditioning Association for rating oil-fired furnaces by test.

The American Gas Association method of rating gas-fired furnaces on performance under tests. This is described in the *Approval Requirements for Central Heating Gas Appliances —Vol. II, Gravity and Forced Air Central Furnaces*, Z21.13.2.

Commercial Standard 113-51 is a method of rating oil-burning floor furnaces by test.

Commercial Standard CS 104-49 is a method of rating warm air furnaces equipped with pot-type oil burners by test.

Various codes covering the construction and performance of appliances as related to fire hazards have been developed by Underwriter Laboratories, Inc. In addition, there are many municipal codes[7] which regulate construction and installation of furnace equipment.

The yardstick of the National Warm Air Heating and Air Conditioning Association provides criteria for evaluating a furnace design and installation against industry accepted standards.

FURNACE EFFICIENCY

Rating formulas of the National Warm Air Heating and Air Conditioning Association are based on 55 percent efficiency for gravity coal furnaces and 65 percent efficiency for forced air coal furnaces. In the tentative *Oil Testing Code* the contemplated minimum efficiency is 70 percent for oil-fired forced air furnaces. Gravity gas furnaces approved by the American Gas Association are assigned a rating based on 75 percent efficiency. All *forced-air gas-fired furnaces* approved by American Gas Association are assigned a rating based on 80 percent efficiency.

DESIGN CONSIDERATIONS

Considerations of prime importance in the design of warm air furnaces and some general suggestions to be observed in connection with each, are as follows:

1. *Adequate heat transfer surface.*
   a. Heat transfer rates of 2000 to 4500 Btu per (hour) (square foot) of heating surface may be obtained without unduly high metal temperatures. With some types of design somewhat higher rates are acceptable.
   b. Fins, pins, and bosses are frequently used to add surface and to break down superficial gas films, both on gas-to-metal and metal-to-air surfaces.
   c. Surface and stack (flue gas) temperatures are good indications of the amount and effectiveness of the heating surfaces.
2. *Safe and efficient combustion of fuel.*
   a. Proper mixture of fuel and air is necessary for efficient combustion. This necessitates careful attention to the design of grates, nozzles, burners, air inlet areas and location, and combustion chamber baffling.
   b. Regulation of the quantity and the distribution of the air for combustion should be provided by use of check dampers, draft regulators, draft hoods, air shutters, and air orifices.
   c. Total draft loss through appliances should not exceed that available from chimneys which would normally be obtainable in the size of building which the appliance will supply with heat.
   d. The use of ignition safety devices such as safety pilots, hold-fire controls, and the like is recommended.
3. *Fuel capacity of appliance.*
   a. With solid fuels adequate coal capacity should be provided for at least 5 hr of operation at the maximum rated combustion rate.
4. *Adequate circulation of air over heating surface.*
   a. In gravity furnaces, free air space between casing and heat exchanger should be great enough to permit free flow over all surfaces.
   b. Forced air furnace design must include fans having proper capacity and suitable performance characteristics. Internal static pressures must be minimized without losing the advantages of high-velocity circulation over the heat exchanger surfaces.
   c. The air flow over the heating surface must be directed to obtain maximum efficiency and to eliminate hot spots and air noises.

d. Air velocities at bonnet should not be much in excess of 1000 fpm, and air temperature distribution at the furnace outlet should be uniform within approximately ± 30 deg.

5. *Durability.*

a. A minimum metal weight for gas-fired heat exchangers is established as No. 20 U.S. Gage for plain carbon steel by the *A.G.A. Approval Requirements for Central Heating Gas Appliances*, with some municipal codes specifying 18 gage. Cast-iron sectional thicknesses of ¼ in. to ⅜ in. are recommended.

b. Added strength and reinforced designs may be required to preclude damage in shipment, burning out from overfiring, or corrosion from condensation.

c. Maximum heat exchanger surface temperatures which may be used vary with the metal. The *American Gas Association Approval Requirements for Central Heating Gas Appliances* specify a maximum of 875 F for cast-iron or steel gas furnaces, and the *National Bureau of Standards* CS 109-44 *Code for Forced Air Solid Fuel-Burning Furnaces* specifies 1000 F as a maximum surface temperature. These temperatures define the range in which oxidation of non-alloy ferrous metal begins. The use of proper alloy additions increases the temperature resistance properties of metals.

d. Casing temperatures should be controlled so that they do not become hazards by burning those who touch them, or by creating fires.

6. *Serviceability.*

a. Those parts of the furnace which may be subject to soot, fly-ash, or condensation deposits should be accessible for cleaning.

b. Parts which may require adjustments or replacements, such as grates, baffles, liners, controls, should be removable.

c. Furnaces should be so designed that they can be installed with a minimum of difficulty.

7. *Control.*

a. Thermostatic controls of various types should be used to correlate space temperatures with unit operation.

b. Controls should be provided wherever possible, to prevent the occurrence of excessive temperatures or other conditions which might cause unsafe operation in any part of the unit.

8. *General design considerations.*

a. Furnace casings are normally constructed of formed and painted sheet steel or of galvanized iron. The casing should be protection from excessive radiation losses and temperatures by use of insulation or sheet steel air space liners. Liners should extend from the grate level to the top of the furnace and should be spaced from 1 in. to 1½ in. from the outer casing.

b. The hood or bonnet of the casing above the furnace should be as high as basement conditions will allow, to form a plenum chamber over the top of the furnace. This tends to equalize the pressure and temperature of the air leaving the bonnet through the various openings. It is generally considered advisable to take off the warm air pipes from the side of the bonnet near the top, as this method of take-off allows the use of a higher bonnet and thus provides a larger plenum chamber.

c. Warm air outlet and return air connections should be designed so that the ductwork may be easily attached. A ⅞ in. flange is normally used for this purpose.

d. Suitable provision should be made in appliances so that the controls and humidifiers may be installed in the proper location. When these auxiliary units are installed in the ductwork, detailed instructions should be provided to insure their proper location.

e. The flue connection should be of integral flue pipe size, and provision should be made to attach the flue pipe to the flue outlet of the furnace.

HUMIDIFICATION EQUIPMENT

Evaporating pans are usually located in the outlet air. There is a present trend toward heating the water. Equipment for doing this may make use of sprays, or it may take the form of water circulating coils placed within the combustion chamber, and connected by pipes to the humidifier pans where a constant water level is maintained by some separate float device. All humidifiers require provision for removal of dirt and lime.

SPACE HEATERS

Space heaters may be classified in several ways, such as:

1. By the type of fuel used as coal, wood, gas, and fuel oil.

2. According to the method of heat distribution as circulators or radiant types. A radiant heater is one in which the heat exchanger surface is exposed directly to the room atmosphere, and the generated heat is dissipated primarily by radiation. A circulating heater is essentially a jacketed radiant heater from which circulation of room air is promoted by the chimney effect caused by the movement of air passing upward between the jacket and heat exchanger surface.

3. According to method of design for particular fuel types, such as: (a) surface-fired and magazine-feed for solid fuels, (b) vaporizing pot-type and blue-flame heater for oil, and (c) vented and unvented heaters for gas. (The type of gas burner design, such as injection, yellow flame, power, and pressure, may also be mentioned.)

SOLID FUEL-FIRED HEATERS

Surface-fired heaters normally have a front firing door and are operated with relatively shallow fuel beds. A magazine-feed heater includes a deep reservoir of fuel to lengthen the attention intervals. In a true magazine-feed heater the rate of fuel ignition would be equal to the rate of burning; and self-feeding should operate to move the unburned fuel by gravity flow from the magazine section into the hearth area. However, the ideal balance between rate of ignition and rate of burning is virtually impossible to attain for any solid fuel under normal usage, although self-feeding may be obtained with wood and some free burning coals. Thus, a magazine-type space heater is essentially a deep surface-fired heater, its principal difference from other heaters being increased fuel capacity.

Materials and Construction

There is no accepted code governing the construction of solid-fuel-burning space heaters. In past years cast iron was used predominantly in the construction of coal and wood heaters, with the exception of the so-called *airtight* heaters designed as low-cost wood-burning units with little consideration for long life, and stoves were priced on a poundage basis. The present trend is toward fabricated steel parts and welded assemblies, although cast iron is still used for grates, firebox liners, and parts subject to high temperatures. Refractory firebox liners are also used quite extensively.

Formed sheet steel is used predominantly for the outer jacket of circulator heaters, although heaters with outer casings formed from cast iron are still readily available. Circulator cabinets normally have surfaces finished with a porcelain enamel, while the casing of a radiant heater is finished with an air-dried japan or a baked enamel.

Both welding and stove bolts are used in unit assembly, and stove cement is used on section joints to prevent air leakage. This latter is extremely important to obtain a low rate of combustion when desired.

Testing and Rating

There is no accepted code governing the method of testing and rating solid fuel space heaters. A tentative procedure, TS-3443, has been issued by the Division of Trade Standards, National Bureau of Standards, but is based on use of anthracite as a rating fuel, although bituminous coals are used predominantly as heating fuels. This procedure, which has been the basis of published ratings, consists of determining the heater output, expressed in Btu per hour, by the indirect method. The measurable heat losses: (1) loss due to moisture in the fuel, (2) loss due to heat in the dry flue gases, (3) loss due to unburned carbon monoxide, and (4) loss due to unburned combustible in the ash and refuse, are measured by test. The total of these four measurable losses, plus an assumed value for unaccounted for losses, are then subtracted from the heat input. The difference multiplied by the burning rate in pounds per hour is the heater output.

When using anthracite as the rating fuel, the unaccounted for loss has been assumed to be zero. It has been accepted practice to use a 20 percent allowance for the unaccounted for losses when burning a bituminous coal. The value of this factor has been under study and, although test work is incomplete, a value of 12 percent of the heat input has been determined as representing the losses due to smoke and unburned hydrocarbons for a surface-fired heater, when burning a high volatile bituminous coal.

No allowance is made for a radiation loss, as this is useful heat.

Design Considerations

Some important considerations in the design of solid-fuel space heaters are:

1. Suitable protection by baffling or insulation against overheating of floors and walls. Although there are no industry-accepted standards by which floor and wall temperatures may be determined, some indication of heater performance, with regard to overheating, may be found by the use of the corner booth test arrangement described in *National Bureau of Standards Commercial Standard* CS 103-43 and *Underwriters Laboratories Standard*, Subject 896, mentioned in following section on Oil Heaters.

2. Tight heater construction to prevent air leakage, and to enable maintenance of a suitably low minimum burning rate. This includes a ground, paper tight joint between the ashpit door and door frame.

3. Sufficient free air space, between the casing and heat exchanger of circulating heaters, to permit free air flow over all surfaces and maintain a suitably low casing temperature.

4. Protection of all metal parts from deterioration due to high temperature. This may be accomplished by: (a) fabrication of certain parts from cast iron or an alloy iron, (b) protection by a refractory liner, (c) use of high temperature enamel coatings, (d) directing air against hot spots.

5. Proper admission of secondary air to complete the combustion process. Care should be taken to prevent this air from also functioning as primary air.

6. Strength in assembly to prevent transportation and use damage.

Considerable work has been directed toward improvement of the performance of bituminous coal-fired space heaters, with particular reference to smokeless operation under conditions of normal operation such as obtained in homes. A smokeless coal heater has been developed by Bituminous Coal Research, Inc., wherein smokeless combustion is obtained by admitting secondary air through a narrow slot, extending from side to side, above the edge of the fuel bed where the gas leaves to enter a vertical gas passage.[4] Complete mixing of the volatile material, released from the coal in the magazine, with the secondary air supplied is obtained as both streams pass under the bottom of the arch. Complete combustion results from this intimate mixing in a region which maintains itself at high temperatures even during banking periods.

OIL HEATERS

Vaporizing pot-type oil heaters consist of: (1) a metal pot in the bottom of which the oil is vaporized, the vapors burning at or near the top of the pot and (2) a secondary combustion chamber, or heat exchanger, in which combustion is completed. The flue connection is made to this chamber or through a second heat exchanger which may be of the diving-flue type installed to increase efficiency. The burner may be designed for operation both with and without mechanical draft.

Blue-flame oil burners differ from the pot-type variety in that removable perforated sleeves are provided above an oil pan instead of a metal pot, and lighting rings or kindlers are used for easy lighting.

Both types of oil burners operate by the burning of the oil vapor rather than the oil itself, the oil being first fed to a chamber in which the oil is entirely vaporized, then mixed with air introduced through suitably located ports and burning at the top of the pot or perforated sleeves. Such heaters are designed to burn No. 1 oil (See Table 4, Chapter 33) or kerosine (coal oil). At no time should oil heavier than that for which the burner is designed be used, as heavier oils may cause excessive carbonization in the burner or fuel feed line.

Materials and Construction

Formed sheet steel and welded assemblies are used primarily in oil heater construction. Standards governing construction which have been generally accepted are:

1. *Commercial Standard for Flue-Connected Oil-Burning Space Heaters Equipped with Vaporizing Pot-Type Burners*, CS101-43 (National Bureau of Standards).

2. *Standard for Oil-burning Stoves*, Subject 896 (Underwriters' Laboratories, Inc.).

3. *Standard for Construction and Performance of Oil Burners for Installation in Stoves and Ranges*, Subject 865 (Underwriters' Laboratories, Inc.).

Some states or municipalities have codes which apply locally, but these usually apply primarily to installation, and the Underwriters' Laboratories label of approval is sufficient to cover acceptance of the unit.

Testing and Rating

Commercial Standard CS101-43 is intended to provide a uniform standard method for ascertaining the maximum practical heat output in Btu per hour of flue-connected oil-burning space heaters under approximately normal service conditions. This method is based upon the following equations:

$$H_r = A - B \qquad (8)$$

and

$$E = H_r/A \qquad (9)$$

where

A = total heat of fuel used.
B = heat lost in flue gases.
H_r = net heat delivered to the room.
E = unit efficiency.

The following minimum performance requirements are stipulated:

1. Adequate provision for ease of lighting and insurance against loss of ignition prior to heating of burner.
2. Ease of operation of controls.
3. Proper operation of burner without excessive carbonization with grades of oil recommended by the manufacturer.
4. The heater shall be capable of passing the 6 percent ICHAM smoke test.
5. The heater shall be capable of operating with an overall efficiency of not less than 70 percent under conditions of test, or at a lower stack draft recommended by the manufacturer.

Design Considerations

Some factors important in the design of oil-burning heaters are:

1. Proper pitch of oil lines from the sump to the burner, thus preventing vapor and air lock.
2. Proper positioning of the oil sump or constant-level valve to maintain the proper oil level in the burners, if factory assembled.
3. Tight construction, not only of oil lines, but of oil tank, sump, and burner to prevent a hazardous condition due to oil leakage.
4. Provision for leveling and aligning the entire heater for maintenance of proper operation. If a separate fuel tank is used, the heater should have provision for secure fastening to the floor to prevent excessive strain on oil supply line, and the consequent danger of leakage of oil.
5. Provision of a draft regulator to prevent abnormal draft fluctuations.
6. Proper shielding of an attached fuel oil supply tank to prevent excessive oil temperatures.
7. All metal parts subjected to the corrosive action of the oil shall be made of noncorrodible metal, or of metal suitably coated to resist corrosion.
8. The heater should have suitable baffling or insulation to prevent overheating of floors and walls.
9. Strength in assembly to prevent transportation and use damage.

GAS HEATERS

Vented gas heaters are defined as those capable of removing 90 percent of the flue gases through a single flue outlet. All heaters having a gas input rating in excess of 50,000 Btu per hour must be of the vented type in order to meet *ASA Approval Requirements for Gas-Fired Room Heaters*.[9]

Space heaters may be classified by burner type as follows:

1. *Injection Burner* type which employs the energy of a jet of gas to inject air for combustion into the burner and mix it with the gas.
2. *Yellow Flame Burner* type in which secondary air only is depended on for the combustion of the gas.

Materials and Construction

Standards covering materials and accessories used in the construction of gas heaters are described in *ASA Approval Requirements for Gas-Fired Room Heaters*[9] and in applicable Listing Requirements.[10]

Efficiency Requirement

Vented space heaters having input ratings in excess of 20,000 Btu per hour are required to have a heating efficiency of not less than 70 percent,[9] based on the total heating value of the gas. Vented space heaters having input ratings of 20,000 Btu per hour or less are required to have a heating efficiency of not less than 65 percent.[9] These efficiencies are based upon the following equation:

$$e_t = 100 - \frac{H_f}{q} \times 100 \qquad (10)$$

where

e_t = heating efficiency, percent.
H_f = heat above room temperature carried away by the flue products, Btu per hour.
q = hourly gas heat input, Btu per hour.

Radiant heaters are required to have a radiant efficiency of not less than 28 percent.

Design Considerations

Some factors important in the design of gas heaters are:

1. Proper design of the burner head, port sizes, and locations so that the flame will not lift, float, or flash back, and be excessively noisy in operation.
2. Proper venting of combustion chamber for relief of forces resulting from ignition of an explosive mixture of gas and air.
3. Protection of valve handles to prevent excessive temperature rise during operation.
4. Insulation and baffling of heater to prevent overheating of walls and floor.

INSTALLATION OF SPACE HEATERS

The two most important considerations involved in the installation of a space heater are safety and chimney draft. The items of chimney details and flue connections which should have special attention are outlined in Chapter 36, Chimneys and Draft Calculations. In all cases, it is recommended that installation be made in accordance with the current *National Building Code*.

REFERENCES

[1] *I = B = R Testing and Rating Code for Low Pressure Heating Boilers* (Institute of Boiler and Radiator Manufacturers, July 1952).

[2] Testing and rating code for boiler-burner units (*Engineering Standards Part II*, Mechanical Contractors Association of America, 1956).

[3] *I = B = R Ratings for Cast-Iron Boilers* (Institute of Boiler and Radiator Manufacturers, May 10, 1955).

[4] Net load recommendations for heating boilers (*Engineering Standards Part II*, Mechanical Contractors Association of America, 1956).

[5] *Comfort Heating* (American Gas Association, 1938, p. 35).

[6] *Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems* (National Warm Air Heating and Air Conditioning Association, Manual No. 5, 5th ed.).

[7] Recommended forms for municipal installation and fire codes are included in *Code and Manual for Design and Installation of Warm Air Winter Air Conditioning Systems* (National Warm Air Heating and Air Conditioning Association, Manual 7, 4th ed.).

[8] B. A. Landry and R. A. Sherman: Development of a design of smokeless stove for bituminous coal (ASME Annual Meeting, 1948).

[9] *American Standard Approval Requirements for Gas-Fired Room Heaters* (American Standards Association, Z21.11, 1954).

[10] *American Standard Listing Requirements for: Automatic Pilots* (Z21.20, 1951); *Gas Appliance Thermostats* (Z21.23, 1941); *Domestic Gas Appliance Pressure Regulators* (Z21.18, 1956); *Automatic Valves for Gas Appliances* (Z21.21, 1952); (American Standards Association).

CHAPTER 36

CHIMNEYS AND DRAFT CALCULATIONS

Draft Definitions, Theoretical Natural Draft, Factors Affecting Natural Draft, Available Draft, Required Draft; Industrial Chimneys, Available Draft for the Industrial Chimney, Determining Industrial Chimney Sizes; Residential Chimneys, Available Draft for Residential Chimneys, Short Chimneys, Determining Residential Chimney Sizes, Draft Requirements of Appliances, Chimneys for Gas Appliances; Recommendations of the National Board of Fire Underwriters, General Considerations

DRAFT is the pressure difference associated with the movement of flue gases through a flue or chimney. Natural draft is caused and measured by the difference in weight of a column of flue gas within the flue and a corresponding column of air of equal dimension outside the flue. Natural draft is always negative and is expressed in inches of water gage.

When the movement of air is supplied by a fan, the draft is said to be forced or induced. When the fan is located so as to push the flue gases through the flue, the draft is forced. When the fan is located so as to draw the flue gases through the flue, the draft is induced.

THEORETICAL NATURAL DRAFT

Theoretical natural draft is produced solely by the difference in weight of the column of flue gases within the flue and that of a corresponding column of air of equal dimensions outside the flue. In the determination of theoretical natural draft, no allowance is made for friction or other positive or negative effects. Theoretical natural draft is expressed by the following formula:

$$D_t = 2.96\, HB_a \left(\frac{\rho_a}{T_a} - \frac{\rho_s}{T_s} \right) \qquad (1)$$

where

D_t = theoretical draft, inches of water.
H = height of flue or chimney, feet.
B_a = barometric pressure, inches of mercury.
ρ_a = density of air at 0 F and 1 atmosphere pressure, pounds per cubic foot.
ρ_s = density of flue gas at 0 F and 1 atmosphere pressure, pounds per cubic foot.
T_a = temperature of air surrounding the chimney, Fahrenheit absolute.
T_s = temperature of the gases, average or effective, in the chimney, Fahrenheit absolute.

Theoretical natural draft may also be expressed by the following simplified formula in which it is assumed that the densities of the flue gas and of the air are equal at the same temperature and pressure:

$$D_t = \frac{\rho H (T_s - T_a)}{5.2 T_a} \qquad (2)$$

where

ρ = mean density of atmospheric air, pounds per cubic foot.

FACTORS AFFECTING NATURAL DRAFT

As indicated in both Equations 1 and 2, the theoretical natural draft is directly proportional to the height of the flue and is dependent upon the absolute mean temperature of the gases within the flue. It is also directly proportional to the density of the air outside the flue although the effect of this factor is minor.

The formulas for theoretical natural draft are based on mean or average temperature which is one half of the sum of the entering and exit temperature. Since the difference in exit and entering temperatures, or the temperature gradient in the flue, is due to the escape of heat through the walls of the flue and chimney, the heat transfer characteristics of the chimney construction affect draft. Heat transfer is also affected by the rate of flow of the flue gases through the flue so that draft is also affected directly.

The flow of flue gases through the flue is retarded by friction which varies directly with the friction coefficient of the flue surface, and with the rate of flue-gas flow. The friction loss may be estimated by means of one of the formulas for ducts such as the Fanning equation or it may be estimated with sufficient accuracy by the method used for air ducts. (See Chapter 21.)

The height of the flue is measured from the point of entrance of the flue gases to the top of the flue. With no wind to produce an aspirating effect and with no obstruction such as a rain cap at the top of the flue, the column of flue gases may extend upward above the top of the flue for several feet. The effect of this column may be sufficient to overcome the negative or retarding effect of friction and to cause the observed draft to exceed the theoretical draft. This added effect has been measured in some test work. A wind blowing across the top of the flue may also produce an aspirating effect sufficient to overcome the effect of friction and to cause the observed draft to exceed the theoretical. Both of these effects are transient and unreliable and should not be considered in determining the proper design of a chimney.

AVAILABLE DRAFT

The available draft produced by any chimney is equal to the theoretical natural draft minus the friction loss. Given the height, the heat transfer characteristics of the chimney construction, the entering temperature of the flue gases and their volume, and the friction coefficient of the flue and its dimensions, the available draft is readily calculable.

The efficiency of a chimney is defined as the ratio of the observed draft or available draft, produced by the chimney

for a given inlet temperature and flow rate, to the ideal draft that would be observed if the same quantity of flue gases traversed the chimney without cooling and without friction. The chimney efficiency may be calculated as follows:

$$\text{Efficiency} = \frac{\text{final measured draft}}{\text{ideal draft calculated from the inlet temperature}}$$

REQUIRED DRAFT

Before the proper chimney for a particular installation can be selected or designed, the required draft of the combustion unit must be known. The required draft is equal to the sum of all of the resistances to gas flow from the point at which combustion air enters the unit to and including the chimney connection. This is based on the assumption that air for combustion is available without restriction (such as would be caused by tightly enclosed building construction) at that

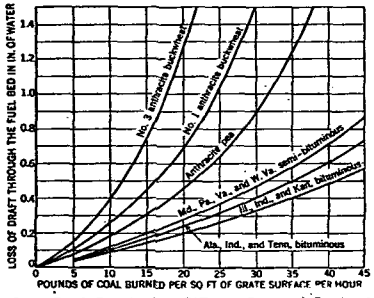


Fig. 1 .... Draft Required at Different Rates of Combustion for Various Kinds of Coal

point of entrance to the unit. Particularly in residences, unrestricted supply of combustion air must be assured.

Fig. 1 presents information on the fuel-bed draft loss for various kinds of coal burned at different rates. Rough generalizations can be given for the losses in the flue passages of boiler or furnace but, on account of the great differences in such devices, more reliable data on their flue-gas volume, temperature, and flue resistance should be obtained for design purposes from their respective manufacturers.

Flue gases encounter resistance to flow in breechings or smoke pipes, and this can probably be treated with sufficient accuracy by means of the method used for air ducts. (See Chapter 21.) The friction in straight ducts can be estimated by means of the last term of Equations 3 and 4.

Also, the temperature of flue gases falls during passage through breechings or flue pipes. For uninsulated surfaces this probably can be adequately estimated by assuming a loss of heat from the flue gas of 3 Btu per (hr) (sq ft) (Fahrenheit deg temperature difference between the gases and surrounding air).

INDUSTRIAL CHIMNEYS

Chimneys can be classified as residential and industrial, the chief difference being their sizes and the types of draft. Chimneys over approximately 1½ ft in diameter are in the industrial chimney class, and their requirements should be treated accordingly. The majority of industrial chimneys operate under induced or forced draft, resorting to natural draft operation only in the case of emergencies. They are built of brick, concrete, or steel, depending upon economy and the type of installation needed. Proper height is of importance because of removal of waste products, inasmuch as the products of combustion are often deflected downward around the chimney and, with the large amount of gases that are exhausted to the atmosphere through the industrial chimney, downwash can be very objectionable.[1]

AVAILABLE DRAFT FOR THE INDUSTRIAL CHIMNEY

The available draft, D_a, *for large chimneys and stacks* has been estimated with apparent satisfaction in the past by means of formulas which in effect deduct an estimated friction loss from a theoretical draft determined as in Equation 1. The friction loss can be estimated by means of one of the formulas available for ducts, such as the Fanning equation. This procedure results in formulas for the available draft as follows:

For a *cylindrical* stack:

$$D_a = 2.96\,HB_a\left(\frac{\rho_a}{T_a} - \frac{\rho_c}{T_c}\right) - \frac{0.00126W^2T_cfL}{d^5B_a\rho_c} \qquad (3)$$

and for a *rectangular* stack:

$$D_a = 2.96\,HB_a\left(\frac{\rho_a}{T_a} - \frac{\rho_c}{T_c}\right) - \frac{0.000388W^2T_cfL(x+y)}{(xy)^2B_a\rho_c} \qquad (4)$$

where

D_a = available draft, inches water gage.
H = height of chimney above inlet, feet.
B_a = existing barometric pressure, inches of mercury.
ρ_a = density of air at 0 F, 1 atmosphere pressure.
ρ_c = density of flue gas at 0 F, 1 atmosphere pressure.
T_a = temperature of atmosphere, Fahrenheit, absolute.
T_c = temperature of flue gas, Fahrenheit, absolute.
W = flue gas flow rate, pounds per second.
f = coefficient of friction.
L = length of friction duct (approximately equal to H), feet.
d = minimum diameter of round chimney, feet.
x and y = length and width of cross-section of rectangular chimney, feet.

The following notes facilitate the use of Equations 3 and 4.

1. The *barometric pressure*, represented by B_a, is the actual pressure at the site of the chimney and not the pressure reduced to sea level datum.

In general, the barometric pressure decreases approximately 0.1 in. Hg per 100 ft increase in elevation.

2. The *unit weight of a cubic foot of chimney gases* at 0 F and sea level barometric pressure is given by the equation:

$$\rho_c = 0.131CO_2 + 0.095O_2 + 0.083N_2 \qquad (5)$$

In this equation CO_2, O_2, and N_2 represent the percentages of the parts by volume of the carbon dioxide, oxygen, and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of ρ_c may be assumed at 0.09.

The effect on the chimney gas density, of superheated water vapor resulting from moisture and hydrogen in the fuel, or of any air infiltration in the chimney proper, is disregarded. Although water vapor content is not disclosed by Orsat analysis,

its presence tends to reduce the actual weight per cubic foot of chimney gases.

3. The *atmospheric temperature* is the actual observed temperature of the outdoor air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* decreases from the breeching connection to the top of the stack. This drop in temperature depends upon the material and construction of the stack, its tightness or freedom from leaks, its area, its height, and the velocity of the gases through it. The same chimney will suffer different temperature losses depending upon the capacity under which it is working, and the variable atmospheric conditions. No general equation covering all these variables has been suggested, but from observations on chimneys varying in diameter from 3 to 16 ft, and in height from 100 to 250 ft, Equation 6 was deduced:[2]

$$T_c = \frac{3.13T_t\left[\left(\frac{H_b}{3}\right)^{0.86} - 1\right]}{H_b - 3} \qquad (6)$$

where

T_t = temperature at the center of the connection from the breeching, Fahrenheit, absolute.
H_b = height of the stack above center line connection to breeching, feet.

5. The *coefficient of friction* between the chimney gases and a sooted surface has been taken by many workers in this field as a constant value of 0.016 for the conditions involved. This value, of course, would be less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys, regardless of the materials of construction, becomes covered with a layer of soot, and thus the coefficient of friction has been taken the same for all types of chimneys and generally constant for all conditions of operation. For reasons of simplicity and convenience to the reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

In important large chimney design, especially when the construction or the materials are unusual, it is recommended that use be made of the Reynolds number[3] in determining the friction factor, f.

The following problem illustrates the use of Equation 3:

Example 1: Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter, operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F; sea level atmospheric pressure, B_a = 29.92 in. Hg; atmospheric and chimney gas density, 0.0863 and 0.09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft. The chimney discharges 100 lb of gases per second.

Solution: Substituting these values in Equation 3 and reducing,

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960}\right) - \frac{0.00126 \times 100^2 \times 960 \times 0.016 \times 200}{10^5 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 2 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in *Example 1*. When the chimney is under static conditions and no gases are flowing, the available draft is equal to 1.27 in. of water, the theoretical intensity. As the amount of gases flowing increases, the available draft decreases until it becomes zero at a gas flow of 297 lb per second, at which point the draft loss, due to friction, is equal to the theoretical intensity. The point of maximum draft and

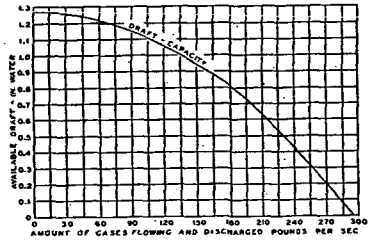
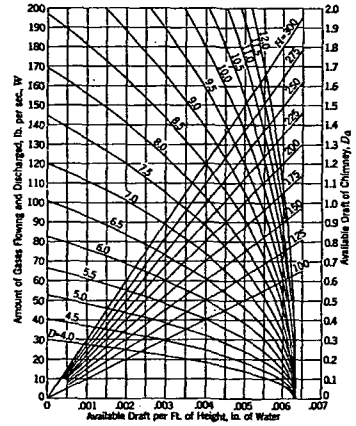


Fig. 2 .... Typical Set of Operating Characteristics of a Natural-Draft Chimney

zero capacity is called shut-off draft, or point of impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is called the wide open point, and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 3, and then plotting the results in the manner shown in Fig. 2.

Fig. 3 is a typical chimney performance chart giving the available draft for various gas flow rates and sizes of chimney. This chart is based on an atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas



To solve a typical example: Proceed horizontally from a Weight Flow Rate point to intersection with diameter line; from this intersection follow vertically to chimney height line; from this intersection follow horizontally to the right to Available Draft scale. Starting from a point of Available Draft, take steps in reverse order.

Fig. 3 .... Chimney Performance Chart

weight of 0.09 lb per cu ft, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be used for general operating conditions. For specific conditions, a new chart may be prepared from Equation 4.

DETERMINING INDUSTRIAL CHIMNEY SIZES

If the required performance for a proposed chimney is known, and *if a chimney-gas velocity is assumed*, Equation 3 can be transposed to yield the necessary height, and an equation can be developed for the required diameter. These operations result in the following equations:

$$H = \frac{D_r}{2.96B_a\left(\frac{\rho_a}{T_a} - \frac{\rho_c}{T_c}\right) - \frac{0.184 f \rho_c B_a V^2}{T_c d}} \qquad (7)$$

The weight of gas per second, $W = 12.075\ (dVB_a\rho_c/T_c)$

$$d = 0.288\sqrt{\frac{WT_c}{B_a\rho_c V}} \qquad (8)$$

where

H = required height of chimney above inlet, feet.
d = required minimum diameter of chimney, feet.
V = chimney gas velocity, feet per second.
D_r = total required draft, inches of water.

For large chimneys, it is usual to assume that total construction cost is least when the product Hd (height × diameter) is minimum. On this assumption, the product of Equations 7 and 8 can be differentiated, and the differential set equal to zero to find the minimum. Solution for velocity then yields the following equation:

$$V_e = \left(\frac{0.772T_c\left(\frac{\rho_a}{T_a} - \frac{\rho_c}{T_c}\right)\sqrt{\frac{WT_c}{B_a\rho_c}}}{f\rho_c}\right)^{2/5} \qquad (9)$$

where

V_e = economical chimney gas velocity, feet per second.

Equations 7, 8, and 9 can of course be simplified if values are assumed for some of the factors in it. Some typical values for boiler plants are:

Average chimney gas temperature 500 F ..... T_c = 960 F absolute
Average atmospheric temperature 62 F ..... T_a = 522 F absolute
Average coefficient of friction 0.016 ..... f = 0.016
Average chimney gas density, 0 F, 1 atmosphere ..... ρ_c = 0.09 lb per cu ft
Barometer reading, sea level ..... B_a = 29.92 in. Hg

When these values are substituted in Equations 7, 8, and 9, respectively, the results are:

$$H = 190D_r \qquad (10)$$

$$d = 1.5W^{2/5} \qquad (11)$$

$$V_e = 13.7W^{2/5} \qquad (12)$$

These equations should be used for general operating conditions only, or where the required data necessary for an exact determination are difficult or impossible to secure. Whenever it is possible to obtain accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 7, 8, and 9.

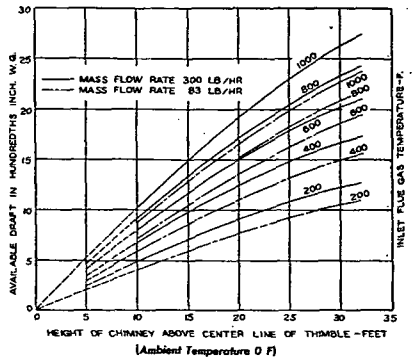


Fig. 4 .... Available Draft for 9″ x 9″ and 9″ x 13″ Masonry Chimneys

Additional construction data for large industrial chimneys, whether brick, concrete, or steel may be found in *Kent's Mechanical Engineers' Handbook*[4] or the *Handbook of Building Construction*.[5]

RESIDENTIAL CHIMNEYS

Since residential heating appliances depend to a large extent on the natural draft of residential chimneys for satisfactory performance, the chimneys must be of adequate height and of optimum cross-sectional area, reasonably gas tight, and as smooth as practicable internally. They must be of such construction as to create no fire hazard to the building. The height and location of a residential chimney are usually limited by architectural and structural considerations, and therefore the draft produced will sometimes be inadequate for satisfactory and efficient operation of the fuel-burning equipment. The basementless house, the one-story ranch or rambler type, and the flat-roofed structure are usually restrictive, particularly with regard to adequate chimney height. The limitations of chimney height should be carefully considered to determine whether or not the type of heating equipment specified will operate properly. If the available draft of the chimney is less than the draft required for the particular heating equipment specified, auxiliary means such as forced or induced draft should be included.

Until recently most chimneys for residences were constructed of masonry. The *National Building Code* of the National Board of Fire Underwriters specifies a minimum of 4 in. of masonry enclosing a fire clay flue liner. Within the past several years, however, the Underwriters' Laboratories have tested and listed as approved *Type A* flues, several chimneys of prefabricated construction. Their advantages, compared with masonry chimneys, are ease of installation, and usually lower cost and a greater degree of safety under abnormal firing conditions. Through the use of insulation or other means to reduce heat transfer through the chimney walls, these *Type A* flues usually operate at a higher mean

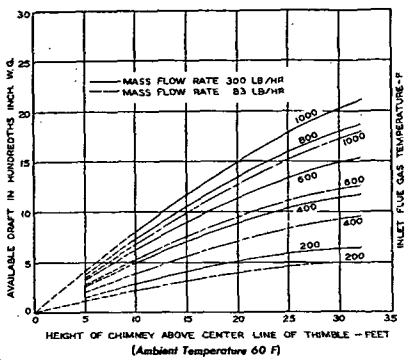


Fig. 5 .... Available Draft for 9″ x 9″ and 9″ x 13″ Masonry Chimneys

temperature for a given entering temperature and therefore produce slightly higher drafts.

AVAILABLE DRAFT FOR THE RESIDENTIAL CHIMNEY

Equations 3 and 4 cannot readily be used for computing the available draft for residential chimneys because of the relatively greater importance of friction losses, cooling of the gases, and soot deposits in small chimneys. Eddy currents and simultaneous flow both upward and downward can occur in a residential chimney for very low flue-gas velocities.

At present, it is best to rely on actual test data for determining the available draft of residential chimneys. Fig. 4 shows the available draft of nominal 9 x 9-in. and 9 x 13-in. masonry chimneys with an ambient temperature of 0 F for a range of effective heights from 5 to 32 ft, a range of entering flue-gas temperatures from 200 to 1000 F, and for mass flow rates of 83 and 300 lb per hr. Fig. 5 shows the same information for an ambient temperature of 60 F. The available drafts produced by 9 x 9-in. and 9 x 13-in. masonry chimneys are equal for practical purposes over the range of mass flow from 83 to 300 lb per hr.[6] In tests of these chimneys the smaller chimney produced slightly greater drafts in the lower end of the range of mass flow rates whereas the larger chimney produced slightly higher drafts in the upper end of the range.

The chimney height for heating plants that operate on an *on-off* or *high-fire low-fire* cycle should be selected to produce the desired draft from Fig. 5 since this class of heating system is required to operate at rated input for short periods when the outdoor temperature is 60 F. Heating plants whose fuel-burning rate is gradually increased as the outdoor temperature decreases are not required to operate at rated input except when outdoor temperatures approach design conditions. For such systems, the chimney height should be selected from Fig. 4 which shows the available draft for an outdoor temperature of 0 F since this is likely to be the more critical condition with respect to chimney draft. The available draft for outdoor temperatures between 0 F and 60 F can be obtained by interpolation from Figs. 4 and 5 with only slight error.

Fig. 6 is a graphical representation of the available draft for a 13-ft brick chimney with a nominal 8 x 8-in. flue liner[7] over a wide range of mass flow rates and for inlet flue-gas temperatures ranging from 200 to 1000 F. This family of curves is a typical group of performance curves showing that there is a certain mass flow rate that produces a maximum available draft for any flue-gas temperature.

The following approximate method may alternately be used to determine the available draft for small residential chimneys, from 10 to 25 ft in height and with internal cross-section areas from 35 to 55 sq in., with a maximum probable error of ±15 percent at the same flow and temperature conditions. This method is based on the chimney efficiencies[8] shown in Fig. 7 and the ideal draft computed from the chimney inlet temperature. The available draft may be expressed as:

$$D_a = \eta_c D_i \qquad (13)$$

where

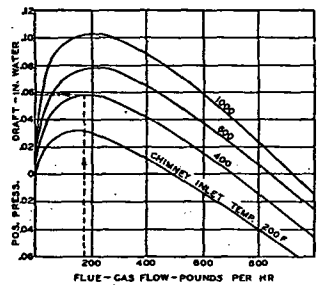
η_c = chimney efficiency taken from Fig. 7 at the desired conditions of temperature and flow.
D_i = ideal draft, calculated from Equation 14, assuming that the barometric pressure is 29.92 in. Hg, and the ambient temperature is 60 F.

$$D_i = 0.2554B_aH\left(\frac{1}{T_a} - \frac{1}{T_i}\right) \qquad (14)$$

where

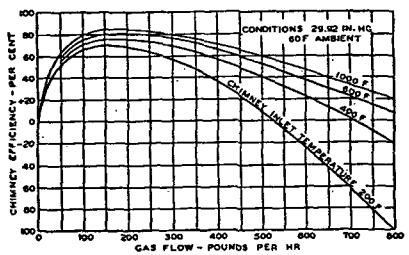
B_a = barometric pressure, inches of mercury.
H = chimney height, feet.
T_i = chimney inlet temperature, Fahrenheit, absolute.
T_a = ambient air temperature, Fahrenheit, absolute.

The clay-lined brick chimney is the most commonly used chimney, but recently other building materials have been used for reasons of economy or convenience. Investigations have established that the results shown in Figs. 4 and 5 for brick



[a] Square Flue Liner 6¾ x 6¾ in. inside.
[b] Barometric Pressure 29.92 in. Hg. Air Temperature 60 F.

Fig. 6 .... Available Chimney Draft for 13 Ft Brick Chimney[a, b]



[a] Derived from temperature plots. Liner 8 x 8 in. outside, 6¾ x 6¾ inside. Height 13 ft.

Fig. 7.... Effect of Gas Flow on Chimney Efficiency[a]

chimneys can be used, with slight error, for chimneys made of shale tile, concrete block, or cinder block.[6]

SHORT CHIMNEYS

The application of heating systems to one-story houses without basements requires special consideration because the chimneys in such houses are often made so low in height for architectural reasons that they produce insufficient draft for some kinds of heating systems. Gas-burning devices do not necessarily require a chimney for proper combustion of the gas, but the chimney must be of adequate size and height to carry the flue gases out of the building since it is undesirable and contrary to American Standards to have the products of combustion discharged in the living space. If a natural-draft oil-burning device is operated with insufficient draft because of a short chimney, smoky combustion usually occurs, pulsations are possible in the combustion chamber, the chimney or smokepipe may become blocked with soot, and the capacity of the heating equipment may be reduced to the point of inadequacy. Insufficient draft is not ordinarily the cause of smoky combustion in coal-burning heaters, but too short a chimney can cause slow pickup of the fire and insufficient heating capacity of the device.

Based on the amount of draft usually required for present-day heating systems at rated output, any chimney with an effective height of less than 15 ft above the center line of the thimble should be regarded as a short chimney. In short chimneys every precaution should be taken to attain the highest possible average temperature in the chimney and the lowest practicable friction loss. Short chimneys of conventional construction are likely to produce from 60 to 90 percent of the ideal draft and, therefore, the draft would only be increased from 10 to 40 percent if all friction and cooling could be eliminated.

The available draft for short chimneys can perhaps be determined most readily by using Equation 13, Fig. 8, and Table 1. Fig. 8 shows the relationship between effective chimney height, inlet flue-gas temperature, and the ideal draft that would be produced for two values of outdoor temperature if there were neither friction nor cooling of the gases as they traversed the chimney. After obtaining the ideal draft for a given chimney height and entering flue-gas temperature from Fig. 8 and selecting the appropriate efficiency value from Table 1, the available draft can be computed from Equation 13. Fig. 8 can be used in reverse in conjunction with

Table 1.... Efficiency of Short Chimneys

A. Masonry Chimneys

Height h ft	Flue Gas Flow Rate lb/hr	Internal Liner Size, in.		
		7 (dia)	7 x 7	7 x 11 and 10 (dia)
		Efficiency, Percent		
5 to 15	90	75–82	65–76	65–80
	200	83–84	67–85	82–88
	315	75–89	67–86	81–89
		(Inlet Flue Gas Temp. 200 to 1000 F)		

B. Uninsulated Metal Chimneys

Height h ft	Flue Gas Flow Rate lb/hr	Internal Diameter, in.	
		6	8
		Efficiency, Percent	
4 to 8	90	90–100	85–95
	200	"	83–88
	315	"	70–80
		(Inlet Flue Gas Temp. 200 to 600 F)	

C. Metal Chimneys[b]

Height h ft	Uninsulated	1-in. insulation with dead air space around chimney	1-in. insulation with 1-in. open air space around chimney
	Efficiency, Percent		
8 to 14	76	85	81

[a] Six-inch chimney not recommended for these flow rates.
[b] Internal dia 8-in.; flue gas flow rate 50 to 125 lb/hr; inlet flue gas temp. 200 to 1000 F.

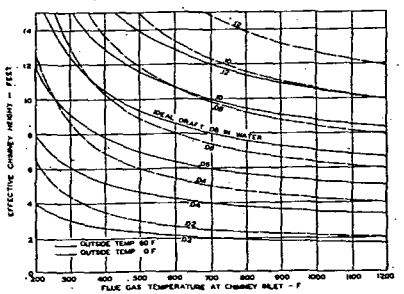


Fig. 8.... Ideal Draft for Short Chimneys at Outside Temperatures of 0 F and 60 F

efficiency values from Table 1 to determine what chimney height is required to produce the required draft at rated output for a given heating appliance.

Fig. 7 shows that the efficiency of a given chimney varies with the mass flow rate of gases and with the entering flue-gas temperature. The efficiency also varies with the height for a given construction and depends somewhat on the method used to connect the smokepipe to the chimney and the amount of exposure to the outdoor air. Experimental data are not sufficiently complete to tabulate efficiencies that take into account all of these variables. However, recommended values of efficiency for several sizes of domestic chimneys of masonry materials and of metal have been selected from the literature[4, 7, 8, 9] and summarized in Table 1 for use with Fig. 8 in determining the available draft produced by short chimneys of 15 ft in height or lower. In parts A and B of Table 1 the value of efficiency chosen from the range cited should be increased as the chimney inlet temperature increases over the applicable range. In part C of Table 1 the efficiency values listed will yield the available draft for the range of mass flows, inlet flue gas temperatures, and chimney height shown within 10 percent.

In the application of short chimneys, observance of the following precautions will assist in obtaining the highest practicable draft and may cure unsatisfactory operation of the chimney and heating plant in certain instances:

1. Use a minimum length of horizontal smokepipe between the heater and chimney.
2. Insulate the smokepipe and the chimney itself, if made of metal. Insulation of the chimney liner in masonry chimneys reduces the heat loss from the flue gases and the infiltration of cold air.
3. Do not use an oversized chimney because larger chimneys produce greater cooling of the flue gases.
4. Avoid downdrafts by proper construction above the roof or by the use of a suitable chimney cap.
5. Avoid the use of barometric dampers in the smokepipe of natural-draft heaters since most barometric dampers permit enough cold air to leak into the chimney to reduce the draft appreciably.
6. Avoid air leaks in the smokepipe and chimney.
7. Avoid connecting more than one heating device to a single chimney.
8. Use construction materials of low heat capacity if the chimney draft must increase quickly for an intermittently-fired heating device.
9. Do not operate a kitchen exhaust fan unless an air intake of ample area is used to prevent lowering the house pressure.
10. Avoid tight utility closets so the chimney draft will not be required to do the added work of drawing combustion air into the closet from the surroundings.

DETERMINING RESIDENTIAL CHIMNEY SIZES

The flue sizes for small residential chimneys are governed by the *National Building Code* of the National Board of Fire Underwriters for gas-burning appliances. Chimney areas for liquid- and solid-fuel-burning devices are selected primarily to meet the requirements of local building codes, but these requirements are not determined by any rigorous formula based on physical principles.

By calculating the available draft for the chimney in question, and comparing it with the performance values of the heating unit (either natural or forced draft type) at the desired output, it is possible to determine whether the chimney is adequate in height for the particular heating unit it serves.

For calculations where the fuel rate and the percentage CO_2 are the only known factors, the flue gas rate can easily be determined for coal, oil, and gas from Fig. 9. By entering Fig. 9 at the percentage CO_2, moving vertically to the curve for the type of fuel, and then moving horizontally to the fuel rate, the flue gas rate in pounds per hour may be determined for any fuel.[7]

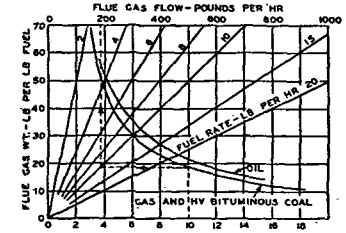
Any of the described methods of determining available draft may be used, but a graphical solution to the problem may be had for the 8 x 8-in. chimney from Figs. 6, 7, and 9. This solution can be best explained by a numerical example.

Example 5: Determine whether a 13-ft, 8 x 8-in. nominal-size flue is sufficient for a coal-heating unit rated at 0.03 in. of water draft at 400 F inlet temperature, fuel rate being 10 lb per hr of bituminous coal, with 10 percent CO_2.

Solution: From Fig. 9, a flue gas rate of approximately 180 lb per hr is obtained. The available draft for a 180 lb per hr fuel rate, and an inlet temperature of 400 F, obtained from Fig. 6, is 0.056 in. of water. This indicates that the chimney is adequate.

The selection of chimney areas for liquid- and solid-fuel-burning devices is difficult because of the variability in efficiency of different models, the possibility that soot on the lining will restrict the chimney area, and the variation in combustion air requirements of different solid fuels. Figs. 6 and 7 show that a given chimney produces a maximum available draft and a maximum efficiency for some intermediate mass flow rate for any selected inlet flue gas temperature. For mass flow rates lower than the optimum the greater cooling of the gases in the chimney causes lower available draft, whereas for mass flow rates above the optimum the greater friction losses reduce the available draft. A chimney for a given heating system should probably be designed to operate at its point of maximum efficiency and maximum available draft for its full rated output. A chimney would have an accelerating effect[10] on the combustion rate of a solid-fuel burning device if it were operating to the left of the optimum point in Fig. 6 because an additional increment in mass flow rate would increase the available draft a small amount and tend to increase the mass flow still more. On the other hand, a chimney operating to the right of the optimum point in Fig. 6 would tend to decelerate the combustion rate for any small increase in mass flow rate.

Data are not complete for the selection of proper chimney areas for heating plants of different capacities, but some information on the effect of cross-section area on the capacity of masonry chimneys is provided by tests[8, 11] on several chimneys with liners having nominal outside dimensions: 9-in.



[a] Average density gas 0.0845 lb per cu ft, 62 F. Average weight oil 7.08 lb per gal, 60 F.

Fig. 9.... Graphical Evaluation of Rate of Flue Gas Flow from Percent CO_2 and Fuel Rate[a]

Table 2 .... Approximate Flue Gas Flow Rates for Maximum Available Draft in Masonry Chimneys

Nominal External Liner Dimensions, In.	Internal Area of Liner Sq. In.	Flue Gas Temperature at Chimney Inlet, F: Mass Flow Rate, Lb/Hr — 200	600	1000	Flue Gas Velocity at Chimney Inlet, Fpm — 200	600	1000
9 (dia)	38.5	170	150	130	175	250	300
9 x 9	49	215	306	334	175	400	600
12 (dia)	78.5	290	Above 320	Above 320	150	—	—
9 x 13	77	295	Above 320	Above 320	150	—	—

diameter, 12-in. diameter, 9 x 9-in., and 9 x 13-in. These tests showed that for flue gas rates up to 200 lb per hr, and entering flue gas temperatures from 200 to 1000 F, the 9-in. round liner provided an available draft equal to, or greater than, that produced by the other three larger liners. When the flue gas rate was increased to 320 lb per hr, the three larger liners produced a little more draft than the smallest one for entering flue gas temperatures above 600 F. Other data[8] show that metal chimneys with an internal diameter of 6 in. should not be used for mass flow rates greater than 130 lb per hr.

Table 2 shows approximate values of the mass flow rates and flue-gas velocities at the chimney inlet that produce the maximum available draft for masonry chimneys[8] of several conventional sizes and with an effective height of 15 ft. This table shows that a 9-in. round chimney is best suited to mass flow rates from 130 to 170 lb per hr, a 9 x 9-in. chimney performs best for flow rates from about 200 to 300 lb per hr, and 9 x 13-in. and 12-in. round chimneys are best suited to flow rates above 300 lb per hr. These results were obtained with clean chimneys, so that conclusions about chimney areas require some modification if soot deposits are taken into consideration.

Soot deposits in chimneys reduce the effective area of the liner and may in some cases entirely close the passage. Soot deposits are likely to be greater in the horizontal passages in the heating plant, the breeching, and the smokepipe, than in the vertical chimney liner. An increase in the liner diameter of one inch above that required for a clean chimney will probably make adequate allowance for soot deposits in all but the worst cases. Where smoky combustion is likely to occur and the mass flow rates on a clean chimney basis approach those listed in Table 2, the next larger commercial size liner should be used. Smoky combustion with oil-burning devices, particularly with vaporizing oil burners, is likely to be caused by inadequate draft which is often due to insufficient chimney height. For coal, inadequate chimney height or chimney area may be a contributing cause, but smoky combustion with coal is related to fuel characteristics and firing methods. Therefore, an increase in chimney area will usually not cure smoky combustion of a coal-burning device, but can be expected to lengthen the interval between cleanings.

New standard sizes of clay flue linings were recently developed by the industry and approved by the American Standards Association, in order to effect economies that can be derived from coordination of the dimensions of building materials. These linings are known as modular clay flue linings, and the dimensions and tolerances are summarized in *ASA Standard* A62.4-47. As the effective areas of these liners are somewhat smaller than those of the corresponding linings used previously, they cannot be used in certain municipalities where building codes specify minimum areas based on the older dimensions.

DRAFT REQUIREMENTS OF APPLIANCES

Typical flue-gas temperatures and drafts required at rated output for several kinds of domestic heating appliances[28] are contained in Table 3. Chimney height and chimney area for cast-iron boilers are specified in the *I = B = R Testing and Rating Code* of the Institute of Boiler and Radiator Manufacturers.

Mechanically-fired devices such as oil burners and stokers are equipped with blowers, and therefore, the chimney is not required to overcome the resistance of the fuel bed or burner. Nevertheless, a draft in the firebox, of about 0.03 in. of water, is desirable so that leakage, if any, will be into the firebox rather than outward from it.

Insufficient draft for natural-draft appliances is likely with installations in attics or in one-story basementless houses. Automatic oil-burning space heaters, floor furnaces, and warm air furnaces employing natural-draft vaporizing burners require a draft of 0.06 to 0.08 in. water for outdoor temperatures of 60 F. Coal-burning heaters and furnaces which attain design rating at 0 F outdoor temperature would require about the same draft. For such requirements, data from Figure 8 and Table 1 show that an effective chimney height, above chimney inlet, should be a minimum of 8 to 12 ft. Because temperature drop and friction loss in the flue pipe cannot be calculated accurately,[7] and the resistance of various chimney connections is uncertain, the minimum calculated heights should have a generous factor of safety applied to allow for these and other uncertainties. Where ample height cannot be provided, forced or induced draft should be provided. Extreme care should be used in the application of forced draft to avoid producing a pressure in combustion

Table 3 .... Drafts Required by Typical Residential Heating Devices or Appliances

Device	Draft, In. Water	Stack Temperature[a] F
Space Heater, Oil-Burning, Pot Burner	0.06 to 0.08	1000
Warm Air Furnace, Oil-Burning, Pot Burner	0.06	860
Warm Air Furnace, Hand-Fired	0.06[b]	900
Floor Furnace, Oil-Burning, Pot Burner	0.06	860
Mechanical Oil Burner, Less than 5 gph	0.03[a]	—
Mechanical Oil Burner, More than 5 gph	0.05[a] or less	—
Cooking Stove, Solid-Fuel	0.04[b]	400
Space Heater, Coal-Burning	0.06[b]	900

[a] Draft in firebox. [b] For chestnut sized anthracite.
[a] 18 in. from heater.

chambers, because they are not usually designed to prevent the outward leakage of toxic gases which might result.

Induced draft by means of motor-driven fans or blowers will solve low-draft problems, but would usually be employed for residential systems only as a last resort. Operation of a blower entails some cost, introduces maintenance requirements, and may be dangerous in event of power failure unless suitable protective devices are used. Induced draft fans designed to resist effects of high temperature and corrosion are available. Fans not carefully designed or modified to meet these conditions should not be applied to this service.

CHIMNEYS FOR GAS APPLIANCES

The Approval Requirements of the American Gas Association are such that a gas-fired appliance must be able to operate at its rated input (plus 10 or 15 percent) without a chimney connection, and without producing carbon monoxide. The primary functions of the chimney for the gas-fired appliance are to remove the products of combustion from the living space and to assure safety to the building. The chimney therefore becomes essentially a vent and, in fact, the vents tested and listed by Underwriters' Laboratories for use only with gas appliances were for some time designated as *Type B* Gas Appliance Vents. Their use is limited to appliances approved by the American Gas Association and operating with flue-gas temperatures not in excess of 550 F at the outlet of the draft hood. A draft hood is always required in connection with a *Type B* flue. The *Type B-W* gas flue is further limited to the venting of wall heaters having an input of not more than 50,000 Btu per hr.

Because of the low temperatures to which gas flues are limited care should be taken to ascertain that the appliance

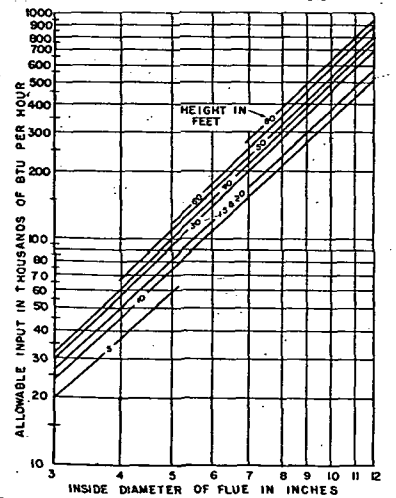


Fig. 10 .... Capacity in Btu per Hour for Gas Appliance Flues or Vents

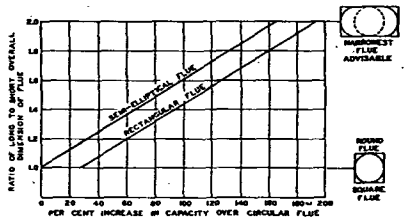


Fig. 11 .... Capacity of a Rectangular Flue or a Semi-Elliptical Flue, with Semi-Circular Ends Having Its Minimum Width Equal to the Diameter of a Circular Flue, Compared with the Capacity of the Circular Flue

is not convertible to fuels other than gas and that the appliance is approved by the American Gas Association. It is well also to be certain that gas fuel will continue to be available to the building.

Chimneys for venting appliances designed for burning gas, can therefore be low in height, but must have adequate area. The height is usually established by the building height. Chimney sizes are usually selected on the basis of heat input of the appliance. The minimum flue or vent sizes adopted by the American Standards Association and the National Board of Fire Underwriters in 1950 for gas appliances[29] are shown in Fig. 10.

Additional provisions of *ASA Standard* 221.30-1950 relating to chimney size are:

1. In no case shall the vent area be less than the area of a 3-in. diameter pipe.
2. When more than one appliance vents into a flue or vent, the flue or vent area shall be not less than the area of the largest flue or vent connector plus 50 percent of the areas of the additional flue or vent connectors.
3. Any-shaped flue or vent may be used provided its flue gas venting capacity is equal to the capacity of the round pipe for which it is substituted.

Since Fig. 10 has been prepared for circular flues, relative capacities for rectangular and semi-elliptical flues[14] are shown in Fig. 11.

Heating appliances designed to burn gas, as well as appliances converted to gas burning, except those equipped with power-type burners and excepting conversion burner installations in excess of 400,000 Btu per hour input in large steel boilers, are always equipped with a draft hood attached to the flue outlet of the appliance. This draft hood is required if the appliance is to meet the approval requirements of the American Gas Association and the American Standards Association, and is essential for safe operation. It is designed to prevent excessive chimney draft which would lower appliance efficiency, to prevent a blocked flue or a downdraft in the chimney from impairing combustion, to provide a relief opening for the products of combustion during downdraft or blocked-flue conditions, and to prevent spillage of the products of combustion to the space surrounding the appliance, if there is a chimney draft equivalent to that provided by a 3-ft chimney. As the draft hood is designed without moving parts, the relief opening is always open, and consequently some air is drawn into the chimney. This air lowers

the gas temperature in the chimney, but it also lowers the dew point of the gases and tends to prevent condensation.

The installation of conversion burner equipment in large boilers is usually made in accordance with regulations of the local gas company. In such installations a definite chimney draft may be required for proper combustion, and consequently the foregoing reference to the use of draft hoods would not apply.

The products of complete combustion of gas are water vapor and carbon dioxide. In the case of manufactured gas, the presence of organic sulfur compounds, generally between 3 and 15 grains per hundred cubic feet, gives rise to minute percentages of sulfur dioxide and sulfur trioxide. The volume of water vapor in the flue products from natural or coke oven gas is about twice the volume of carbon dioxide. It is extremely important that the chimney be tight and resistant to corrosion, not only from moisture, but also from dilute sulfur trioxide.

Clay linings with joints which prevent retention of moisture, and linings made of noncorrosive materials, are advantageous. The interior protection of unlined chimneys by applying an asphalt-chromate emulsion has been investigated by some gas companies but the material is not of current interest because insurance companies do not favor potentially combustible coatings.[14]

Table 4 .... Results of Field Survey on Chimney Damage

Type of Construction	Number of Chimneys	Number Reported Damaged	Percent Damaged
Chimneys without liners	4720	562	11.9
Chimneys with sewer tile construction	8410	104	1.2
Wilder metal coated with asphalt chromate emulsion	5185	291	5.6
Steel coated with plastic	9200	1928	21.0
Vitreous enamel on steel	614	No record	—
Aluminum	50,000	228	0.45

An investigation of damage occurring in 78,129 chimneys used for venting gas appliances in a severe climate has been made by the Minneapolis Gas Company with the results shown in Table 4 taken from Reference 15.

Advice regarding recommended practice and materials for flue connections and chimney linings can usually be obtained from the local gas company, and should be given careful consideration.

RECOMMENDATIONS OF THE NATIONAL BOARD OF FIRE UNDERWRITERS

For general data on the construction of chimneys, reference should be made to the *National Building Code*, recommended by the National Board of Fire Underwriters, Article X, Section 1001 to 1006, in which the following are some of the important provisions listed in the 1951 edition:

1. *Flue Connections Required.* Every heating apparatus or heat producing appliance requiring a flue connection shall be connected with a flue conforming to the provisions of this article. This shall not include electric appliances; gas appliances, except as specifically required in this article; or oil fired appliances especially designed for use without flue connection.

2. *Use of Nonconforming Flues.* Flues not conforming to the requirements of this article for chimneys, metal smokestacks or vents for gas appliances, shall not be used unless listed by Underwriters' Laboratories, Inc., installed in full compliance with the listing and the manufacturer's instructions, and approved for such use by the building official.

3. *Smoke Pipe Connections.*

a. No flue shall have smoke pipe connections in more than one story of a building, unless provision is made for effectively closing smoke pipe openings with devices made of noncombustible materials whenever their use is discontinued temporarily, and completely closing them with masonry when discontinued permanently.

b. Two or more smoke pipes shall not be joined for a single connection, unless the smoke pipes and flue are of sufficient size to serve all the appliances thus connected.

c. The smoke pipe of a heating appliance shall not be connected into the flue of an incinerator which has the rubbish chute identical with the smoke flue.

4. *Construction of Chimneys.*

a. Chimneys hereafter erected within or attached to a structure shall be constructed in compliance with the provisions of this section.

b. Chimneys shall extend at least 3 ft above the highest point where they pass through the roof of the building, and at least 2 ft higher than any ridge within 10 ft of such chimney.

c. Chimneys shall be wholly supported on masonry or self-supporting fireproof construction.

d. No chimney shall be corbeled from a wall more than 6 in.; nor shall a chimney be corbeled from a wall which is less than 12 in. in thickness, unless it projects equally on each side of the wall; provided that in the second story of 2-story dwellings corbeling of chimneys on the exterior of the enclosing walls may equal the wall thickness. In every case the corbeling shall not exceed 1 in. projection for each course of brick projected.

e. No change in the size or shape of a chimney, where the chimney passes through the roof, shall be made within a distance of 6 in. above or below the roof joists or rafters.

5. *Chimneys for Heating Appliances, Low Heat Industrial Appliances and Portable Type Incinerators.*

a. Chimneys for stoves, cooking ranges, warm air, hot water and low pressure steam heating furnaces, fireplaces, and low heat industrial appliances, other than chimneys for incinerators of nonportable type, shall be constructed of solid masonry units or of reinforced concrete. The walls shall be properly bonded or tied with corrosion-resistant metal anchors. In dwellings and buildings of like heating requirements, the thickness of the chimney shall be not less than 4 in. In other buildings the thickness shall be not less than 8 in., except that rubble stone masonry shall be not less than 12 in. thick.

b. Every such chimney hereafter erected or altered shall be lined with a flue lining conforming to the requirements below.

c. Flue linings shall be made of fire clay or other refractory clay which will withstand the action of flue gases and resist, without softening or cracking, the temperatures to which they will be subjected, but not less than 2000 F. Flue linings may be of cast iron of approved quality, form and construction.

d. Required clay flue linings shall be not less than ⅝ in. thick for the smaller flues, and shall increase in thickness for the larger flues.

e. Flue linings shall be installed ahead of the construction of the chimney as it is carried up, carefully bedded one on the other in Type A, Type B, or fire clay mortar with close fitting joints left smooth on the inside.

f. Flue linings shall start from a point not less than 8 in. below the intake, or, in the case of fireplaces, from the throat of the fireplace. They shall extend, as nearly vertically as possible, for the entire height of the chimney, and be extended 4 in. above the top of cap of the chimney.

g. Cleanouts for flues or fireplaces shall be equipped with cast-iron doors and frames arranged to remain tightly closed when not in use.

h. When two or more flues are contained in the same chimney, at least every third flue shall be separated by masonry at least 4 in. thick bonded into the masonry wall of the chimney. Where flue linings are not so separated, the joints of adjacent flue linings shall be staggered at least 7 in.

6. *Sizes of Flues.*

a. The cross-sectional area of smoke flues shall be designed and proportioned to meet the conditions of temperature, within and without flue, thickness of masonry, exposure, shape and material of flue, and other influences.

The *National Building Code* specifies lined chimneys and metal smoke stacks for all gas appliances which may be converted readily to the use of solid or liquid fuel, and also for all boilers and furnaces, except those having a flue-gas temperature not exceeding 550 F at the outlet of the draft hood when burning gas at the manufacturer's rating and which may, therefore, be connected to *Type B* vent piping. Approved *Type B* vent piping is noncombustible, corrosion-resistant piping of adequate strength and heat-insulating value, and having bell and spigot or other acceptable joints. Fig. 10 may be used for selection of vent-pipe size.

Important points to be considered in the use of *Type B* vent piping are:

1. *Type B flues* must be plainly and permanently marked at the point where the vent connection enters the flue: *For Use on Gas Appliances Only.*

2. *Type B* gas vents shall be installed with a clearance to combustible material or construction, whether plastered or unplastered, of not less than one inch, provided that for vents of floor furnaces, such clearance shall be not less than 3 ft from the outlet of the draft hood, measured along the center line of the vent piping.

Other important points that should be considered for flues and vents for gas appliances are as follows:

1. Clearances from combustible material to gas appliance vent piping other than approved *Type B* gas vents shall be in accordance with the *Building Code Standards* of the National Board of Fire Underwriters for the Installation of Heat Producing Appliances, Heating, Ventilating, Air-Conditioning, Blower, and Exhaust Systems.

2. Every flue-connected appliance, except an incinerator, unless its construction serves the same purpose, shall be equipped with an effective draft hood which either (*a*) has been approved as part of the appliance or (*b*) complies with nationally recognized standards for draft hoods. The draft hood shall be attached to the flue collar of the appliance as conditions permit, and in a position for which it is designed with reference to horizontal and vertical planes. The draft hood shall be so located that the relief opening is not obstructed by any part of the appliance or adjacent construction.

3. No vent pipe from a gas appliance shall be interconnected with any other vent pipe, smoke pipe, or flue, unless such gas appliance is equipped with an automatic device to prevent the escape of unburned gas at the main burner or burners. Where a gas appliance vent pipe is joined with a smoke pipe from an appliance burning some other type of fuel, for connection into a single flue opening, they shall be joined by a Y fitting located as close as practicable to the chimney. With liquefied petroleum gases, the automatic device to prevent the escape of unburned gas shall shut off the pilot light, as well as the main burner or burners.

Recent tests of masonry chimneys[16] made of a variety of materials have developed additional recommendations regarding the construction of masonry chimneys that will decrease the hazard to surrounding combustible materials.

GENERAL CONSIDERATIONS FOR CHIMNEYS

The draft of domestic chimneys may be subject to a variety of influences not usually encountered in power chimneys[17] because of the low available draft often supplied by a short chimney. Horizontal winds have an aspirating effect as they cross the chimney and are an aid to draft. However, surrounding objects, such as trees or other buildings, may affect the direction of the wind at the chimney top, and may even direct it down the chimney, tending to reduce the draft or even to cause it to change to a positive pressure.

It is not to be assumed that increasing the cross-sectional area of a chimney will always effect a cure for poor draft. The opposite result may occur because of the cooling effect of the larger area, and the effect of recirculation of the flue gases. The flow of gases into the chimney top has been observed at low rates, and recirculation in small residential chimneys has been noted throughout the entire length of a chimney and smoke pipe, with the greater amount of recirculation occurring at the thimble. The effect of recirculation decreases with chimney height and the increase in flue-gas velocity.

It is also important to consider the course of the air supply for proper combustion. The boiler or furnace is usually located in the basement. In the majority of cases, the furnace room has windows and doors opening to the outdoors on two or more sides of the house. Usually enough air leaks into the furnace room through these openings to sustain combustion. In some cases, however, windows and doors are so tight as to restrict the flow of combustion air, and thereby affect the correct operation of the chimney. If the boiler room is fairly tight and is open to the outdoors on only one side of the house, the draft will be affected in windy weather even with windows or doors open. If the wind is blowing toward the boiler room, the draft will be increased, but if blowing in the opposite direction, the draft may be decreased.

The surrounding of a heating appliance with a restrictive enclosure that will limit entrance of combustion air is more likely to occur in a small utility closet installation on the first floor than in a basement installation. An opening with a free area approximately twice the area of the smokepipe should therefore, be provided between the utility closet and the living space or the outdoors. If the utility closet is connected to the outdoors, greater difficulty with wind pressures will be encountered. Where a draft regulator is used it should have ample communication with the same space from which the combustion air is taken. Where forced warm air furnaces are enclosed in utility closets, care should be used to make the return air connection inside the utility closet airtight so that the blower cannot reduce the pressure in the closet and cause a downdraft in the chimney.

Two or more chimneys, either large or small, should never be connected. If connected at the bottom, hot gases in the U-tube thus formed would be in unstable equilibrium. Cold air from the top would descend through one such chimney and drive the hot gases out of the other, thus annulling the draft.

More than one device can be served by one chimney. Batteries of boilers are commonly connected to a single chimney in power plants. However, if two or more chimneys are used, each chimney should be used separately for part of the boilers, and not connected in manifold with another chimney, in order to avoid the difficulty described previously.

In domestic installations it is sometimes necessary to serve a space heater or cooking stove and a water heater with the same chimney flue. This is not desirable, especially for low chimneys, since doors left open on one device, while it is unfired, will tend to annul the draft on another device. Gas-burning devices, with their draft hoods and lack of draft dampers, are especially bad in this respect. The traditional method of avoiding this with brick chimneys has been to construct multiple-flue chimneys, so that each fuel-burning device could be served by a separate opening. If two devices must be served by one flue-opening in a chimney, their connections to the chimney should not be located opposite each

other. The connection from the larger device should be reasonably low, and that from the smaller should be up near the ceiling, so that each device can be serviced as well as possible, regardless of the treatment of the other.

Even where such precautions are taken, there is, under unusual conditions, some possibility of flow of combustible gases from one appliance into another. Reverse flow of cooled gases has been demonstrated in a chimney at very low rates of flow.[7] Under certain similar quiescent conditions, accidental discharge of combustible gases from a defective device into a chimney could result in flow of these gases counter current into the combustion chamber of another device attached to the same chimney. If ignition occurred in this second device, an explosion could result.

Excessive height in a chimney does no harm, but means for controlling the draft are more than ordinarily essential if the chimney is too tall. Coal-burning devices often have air leaks around the firebox, and the draft doors sometimes fit so poorly that the fire cannot be controlled at a low rate. The simplest remedy for such cases is the barometric damper which admits air into the flue pipe and thus reduces draft.

Where a chimney serves a fireplace, it is important that no other heating device be connected to it unless the fireplace is effectively sealed.

REFERENCES

[1] R. H. Sherlock and B. A. Stalker: *A Study of Flow Phenomena in the Wake of Smokestacks* (University of Michigan, *Department of Engineering Research Bulletin* No. 29, 1941).

[2] E. F. Miller and James Holt: *Notes on Power Plant Design* (Massachusetts Institute of Technology, 1930).

[3] L. F. Moody: Friction factors for pipe flow (*ASME Transactions*, Vol. 66, 1944, p. 671).

[4] R. T. Kent (ed.): *Mechanical Engineers' Handbook* (John Wiley & Sons, New York).

[5] G. A. Hool and N. C. Johnson: *Handbook of Building Construction* (McGraw-Hill Book Co., New York, 1929).

[6] R. S. Dill, P. R. Achenbach, and J. T. Duck: Observed performance of some experimental chimneys (ASHVE Transactions, Vol. 48, 1942, p. 351).

[7] L. B. Schmitt and R. B. Engdahl: Performance of residential chimneys (ASHVE Transactions, Vol. 55, 1949, p. 241).

[8] P. R. Achenbach and S. D. Cole: Performance of fourteen masonry chimneys under steady state conditions (ASHVE Transactions, Vol. 55, 1949, p. 129).

[9] R. D. Thulman and W. H. Shenkle: A theoretical and experimental investigation of the performance of some short flues under steady-state conditions (Thesis at Massachusetts Institute of Technology, June 1951).

[10] P. R. Achenbach: Physics of chimneys (*Physics Today*, Vol. 2, December 1949).

[11] R. K. Thulman: *Performance of Masonry Chimneys for Houses* (*Housing and Home Finance Agency Technical Paper* No. 13, August 1949).

[12] CS101-43 *Oil-Burning Space Heaters Equipped with Vaporizing Pot-Type Burners*, CS75-42 *Automatic Mechanical Oil Burners Designed for Domestic Installations*, CS(E)104-43 *Warm Air Furnaces Equipped with Vaporizing Pot-Type Burners*, CS109-44 *Solid-Fuel Burning Forced Air Furnaces*, CS113-44 *Oil-Burning Floor Furnaces Equipped with Vaporizing Pot-Type Burners* (National Bureau of Standards Commercial Standards).

[13] *American Standard Installation of Gas Piping and Gas Appliances in Buildings* (American Standards Association, Z21.30-1950).

[14] *Comfort Heating* (American Gas Association, 1938, p. 71).

[15] C. G. Segeler: *Sizing of Chimneys and Flues Used for Gas Appliances* (American Gas Association; Sixth International Gas Conference, 1955).

[16] N. D. Mitchell: Fire hazard tests with masonry chimneys (*National Fire Protection Association Quarterly*, October 1949).

[17] S. Konzo: Chimneys and draft (*Winter Air Conditioning*, National Warm Air Heating and Air Conditioning Association, 1939, Chapter 32).

CHAPTER 37

ESTIMATING FUEL CONSUMPTION FOR SPACE HEATING

Bases of Fuel Estimates; Efficiency of Utilization; Calculated Heat Loss Method: Computation and Application, Short Methods for Estimating Heat Loss; Degree-Day Method: Computation and Application, Unit Fuel Consumption per Degree Day, Estimating Consumption for Various Fuels, Degree Day as an Operating Unit; Industrial Degree Days; Maximum Demand and Load Factors

IT IS often necessary to estimate the anticipated heat requirements and fuel consumptions of heating plants for either short or long terms of operation. These quantities can be much more difficult to calculate than design heat loss or required system capacity, since they involve essentially the summing up of the net result over the period in question of the influence of many factors which may vary greatly with time. It will seldom be possible to predict with any great accuracy the way in which all the factors involved will vary throughout the prediction period. In addition to this, the calculations required to take all such variations into account become very involved. For these and other reasons records of past operating experience, when these are available, provide the most reliable and usually the most accurate basis for the prediction of future requirements.

Records of past heat requirements or of fuel consumption of a particular building are a better basis for estimates than are averages of records from similar buildings. In the absence of past records for a particular building the data from similar plants in the same locality may have to be used. Averages of consumption figures taken from many types of plants in many types of buildings in various localities can only produce an average estimate which may prove to be inaccurate when applied to a particular building.

Where unusual operating conditions exist due to factors such as excessive ventilation, abnormal inside temperatures, and heat gains from external sources, or where, in the case of proposed buildings of unusual design, no information is available regarding former consumption, it is necessary to estimate fuel consumption from the computed heat losses.

In preparing fuel consumption estimates it is well to realize that any estimating method used will produce a more reliable result over a long period operation than over a short period. Nearly all of the methods in common use will give trustworthy results over a full *annual* heating season, and in some cases such estimates will prove consistent within themselves for monthly periods. As the period of the estimate is shortened, there is more chance that some factor not taken into account directly in the estimating method will deviate from its long-time average value, and thus lead to serious error in the predicted heat requirement.

EFFICIENCY OF UTILIZATION

Any method of estimating fuel requirements based primarily on calculation of heat losses must, of necessity, also take into account considerations of the efficiency with which the heat in the fuel is used. Efficiency can be defined in a variety of ways for various purposes, and the values in any given case can vary widely depending on the conditions. For estimating purposes, as will be shown later, it is the *efficiency of utilization* of the fuel over the calculation period which is wanted. This is distinct from *furnace efficiency* which expresses the heat delivered by a furnace unit as a percentage of the heat provided in the fuel and which is the efficiency normally used in describing the performance of a unit under rated, or stated, load conditions. It has been shown, at the University of Illinois for example,[1] that in the case of a dwelling with an inside chimney, as much as 35 percent of the heat delivered to the chimney which is normally considered to be a loss so far as efficiency of the furnace is concerned may be delivered to the house through the heated chimney walls. This recovered heat amounted to as much as 12 percent of the heat required by a house. Much of this recovered heat may be available for heating. It has been said in connection with the same study that at times only about 50 percent of the heat utilized by a house was supplied by radiators, while the other 50 percent of the heat utilized was supplied from such sources as chimney, piping, boiler jacket and smoke pipe, people, lights, etc. A summary[2, 3] of many tests in two research residences at the University of Illinois using many fuels and systems give values of 67 to 90 percent for overall house efficiency (the ratio of the heat loss from the structure to the heat input to the unit for an average day).

The efficiency of utilization of the fuel burned can at times be relatively very high for houses. It can also vary quite widely from one house to another because of variations in chimney location and design, in the utilization of heat lost to the basement from the heating unit and smoke pipe, and for other reasons. The efficiency of utilization may also vary with time of year as the average load on the furnace varies. Some of the reasons for this in the case of intermittently-fired oil furnaces have been demonstrated.[4]

In buildings other than houses the efficiency of utilization of the heat supplied in the fuel may also vary widely. There will, in general, be less tendency in many larger buildings toward effective use for heating purposes of the heat which is normally lost from chimney, furnace, and piping surfaces, and efficiency is seldom likely to be as high as in the case of many houses.

The average fuel consumption of various types of approved gas-fired equipment has been obtained from a large number of heating systems.[5] Corresponding efficiencies of

utilization over the heating season can be calculated from these data. They show a variation from approximately 72 to 88 percent depending upon the type and size of system. Laboratory tests on a gas conversion burner in a heating boiler operated with *on* and *off* cycles gave about 72 percent efficiency.[4] Other tests on coal-fired room heaters indicated, for the heating season, efficiencies of 65 to 75 percent, when the heat from the flue pipe was included.[7] A survey of 30 residences in one locality showed a variation of 45 to 75 percent in utilization efficiency, depending upon the condition of the equipment and the fuel used.[8]

Experience at the University of Illinois was used in establishing overall efficiencies for a tabulation of the comparative costs of heating the same building with various fuels to be found in a publication of the Small Homes Council.[9]

The efficiencies of utilization over the heating season shown in Table 1 are suggested as a guide. Considerable judgment may have to be exercised in selection of appropriate values for any given case of fuel consumption calculations.

Table 1 . . . . Efficiency of Utilization Over the Heating Season

Type of Fuel-Burning Unit	Efficiency, Percent
Gas, designed unit	75–80
Gas, conversion unit	60–80
Oil, designed unit	65–80
Oil, conversion unit	60–80
Bituminous coal, hand-fired with controls	50–65
Bituminous coal, hand-fired without controls	40–60
Bituminous coal, stoker-fired	50–70
Anthracite, hand-fired with controls	60–80
Anthracite, hand-fired without controls	50–65
Anthracite, stoker-fired	60–80
Coke, hand-fired with controls	60–80
Coke, hand-fired without controls	50–65
Direct electric heating	100

CALCULATED HEAT LOSS METHOD

The quantity of fuel or energy required for heating may be found by the Calculated Heat Loss Method from the equation

$$F = \frac{XN}{EC} \tag{1}$$

where

F = quantity of fuel or energy required (in the units in which C is expressed).
X = average heat requirement for the period under consideration, Btu per hour.
N = number of heating hours in estimate period (for an Oct. 1–May 1 heating season, 212 days × 24 hr = 5088).
E = efficiency of utilization of the fuel over the period, expressed as a decimal; not the efficiency at peak or rated load condition.
C = heating value of one unit of fuel or energy.

The value of X is usually found from the equation

$$X = \frac{H(t - t_o)}{t_i - t_o} \tag{2}$$

where

H = calculated heat loss including infiltration loss, Btu per hour, based on t_o and t_i.
t = average inside temperature maintained during heating period, Fahrenheit.
t_o = average outside temperature through estimate period, Fahrenheit (for U. S. cities with an Oct. 1–May 1 heating season—see Table 2).
t_i = *indoor design* temperature, Fahrenheit (usually 70 F).
t_o = outdoor design temperature, Fahrenheit (see Table 2).

so that the value of F becomes

$$F = \frac{H(t - t_o)N}{E(t_i - t_o)C} \tag{3}$$

The calculation of X, the average heat requirement, Btu per hour, over the period under consideration is given only approximately by Equation 2, when the value of H used is the calculated maximum load, or design heating load for the building, as computed by the methods outlined in Chapters 9, 11, and 12. This value is commonly used since it is usually available from the heating system design calculations. This procedure will usually lead to values of X which are too high. In the first place, the design wind speed on which such calculations are based is usually higher than the average wind speed for the period so that infiltration losses may be overestimated. In addition there will usually be many sources of heat not taken into account in the estimation of maximum heat demand which may actually contribute to the heating and thus may decrease the fuel or energy required for heating without necessarily affecting the maximum heat demand. The more important sources are solar radiation, people, lights, and equipment.

Improved estimates of fuel requirements will result if H can be recalculated so as to be more representative of *average* conditions over the heating period rather than the *maximum* conditions used in maximum heating load estimation. Further improvement may be obtained at the expense of more involved calculations by making estimates of the average heat gains, and deducting these from the calculated heat losses. There is a practical limit to the refinement in calculations which should be attempted since there will always be some factors which are only approximately known. The weather, for example, very largely determines the heat losses, but will normally be described only approximately in terms of average values of a limited number of weather elements. Wind speed, and sunshine will not necessarily be distributed uniformly over periods of different outdoor temperatures. In addition there is a limit to the accuracy of prediction of the weather elements over the period of interest which is always in the future. For these and other reasons, including such unpredictable factors as the way in which windows and doors will be opened and closed, the determination of H or of X must always be in the nature of an estimate only.

Values of the average outdoor temperature, t_o, are required for the calculation of X by means of Equation 2. Such values for the period October 1 to May 1 are given for U. S. cities in Table 2. The assumption of an Oct. 1–May 1 heating season is reasonably accurate in the well-populated New York-Chicago zone but heating may be required over a longer period for points as far north as Minneapolis, and over a shorter period south of Washington, D. C. Consequently values taken from Table 2 may have to be adjusted

(*Text continued on p. 534.*)

Table 2 . . . . Average Monthly and Yearly Degree Days for Cities in the United States and Canada[a, b, c] (Base 65 F)

State	Station		Years	No. of Seasons	Avg. Winter Temp.[d]	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total
Ala.	Anniston	A			52.3	0	0	17	118	438	614	614	485	381	128	25	0	2820
	Birmingham	A			51.8	0	0	13	123	396	598	623	491	378	128	30	0	2780
	Mobile	A				0	0	0	28	219	376	416	304	222	47	0	0	1612
	Mobile	C			58.9	0	0	0	23	198	357	412	290	209	40	0	0	1529
	Montgomery	A				0	0	0	69	304	491	517	388	288	80	0	0	2137
	Montgomery	C			56.4	0	0	0	55	267	458	483	360	265	66	0	0	1954
Ariz.	Flagstaff	A			35.9	49	78	243	586	876	1135	1231	1014	949	687	465	212	7525
	Phoenix	A			59.5	0	0	0	22	223	400	474	309	196	74	0	0	1698
	Phoenix	C			57.7	0	0	0	13	182	360	425	275	175	62	0	0	1492
	Yuma	A			62.5	0	0	0	0	105	259	318	167	88	14	0	0	951
Ark.	Bentonville		06/07–40/41	35		1	1	38	216	516	810	879	716	519	247	86	7	4038
	Fort Smith	A				0	0	9	131	435	698	775	571	418	127	24	0	3188
	Little Rock	A				0	0	10	110	405	654	719	543	401	122	18	0	2982
Calif.	Eureka	C			49.3	267	248	264	335	411	508	552	465	493	432	375	282	4632
	Fresno	A				0	0	0	86	345	580	629	400	304	145	43	0	2532
	Independence		98/99–40/41	43		0	0	23	216	512	778	799	619	477	267	120	18	3834
	Los Angeles	A				31	22	56	87	200	301	378	305	273	185	121	56	2015
	Los Angeles	C			59.3	0	0	17	41	140	253	328	244	212	129	68	19	1451
	Needles		17/18–38/39	22		0	0	0	19	217	416	447	243	124	26	3	0	1495
	Point Reyes		98/99–40/41	43		350	336	263	282	317	425	467	406	437	413	415	363	4474
	Red Bluff	A			52.9	0	0	0	59	319	564	617	423	336	177	51	0	2546
	Sacramento	A				0	0	22	98	357	595	642	428	348	222	103	7	2822
	Sacramento	C			53.0	0	0	17	75	321	567	614	403	317	196	85	5	2600
	San Diego					11	7	24	59	147	255	317	247	223	151	97	43	1574
	San Francisco	A				144	136	101	174	318	487	530	398	378	327	264	164	3421
	San Francisco	C			54.2	189	177	110	128	237	406	462	336	317	279	248	180	3069
	San Jose	C			53.5	7	11	26	97	270	450	487	342	308	229	137	46	2410
Colo.	Denver	A			37.0	5	11	120	425	771	1032	1125	924	843	525	286	65	6132
	Denver	C			37.9	0	5	103	385	711	958	1042	854	797	492	266	60	5673
	Durango		04/05–40/41	37		25	37	201	535	861	1204	1271	1002	859	615	394	139	7143
	Grand Junction	A			39.9	0	0	36	333	792	1132	1271	924	738	402	145	23	5796
	Leadville		07/08–40/41	34		280	332	509	841	1139	1413	1470	1285	1245	990	740	434	10678
	Pueblo	A				0	0	74	383	771	1051	1104	865	775	456	203	27	5709
Conn.	Hartford	A				0	14	101	384	699	1082	1178	1050	871	528	201	31	6139
	New Haven	A				0	18	93	363	663	1026	1113	1005	865	567	261	52	6026
D. C.	Washington	A				0	0	37	237	519	837	893	781	619	323	87	0	4333
	Washington	C			43.4	0	0	32	231	510	831	884	770	606	314	80	0	4258
Fla.	Apalachicola	C			60.9	0	0	0	17	154	304	352	263	184	33	0	0	1307
	Jacksonville	A			60.6	0	0	0	16	148	309	331	247	169	23	0	0	1243
	Jacksonville	C			62.0	0	0	0	11	129	276	303	226	154	14	0	0	1113
	Key West	A			72.5	0	0	0	0	0	22	34	25	8	0	0	0	89
	Key West	C			73.1	0	0	0	0	0	18	28	24	7	0	0	0	77
	Miami	A				0	0	0	0	8	52	58	48	12	0	0	0	178
	Miami	C			71.4	0	0	0	0	5	48	57	48	15	0	0	0	173
	Pensacola	C			59.7	0	0	0	18	177	334	383	275	203	45	0	0	1435
	Tampa	A				0	0	0	0	60	163	201	148	102	0	0	0	674
Ga.	Atlanta	A				0	0	8	110	393	614	632	512	404	133	20	0	2826
	Atlanta	C				0	0	8	107	387	611	632	515	392	135	24	0	2811
	Augusta	A				0	0	0	59	252	494	521	412	308	62	0	0	2138
	Macon	A				0	0	0	63	280	481	497	391	275	62	0	0	2049
	Savannah	A			56.7	0	0	0	38	225	412	424	330	238	43	0	0	1710
	Thomasville		05/06–40/41	36		0	0	2	48	208	361	359	299	178	52	5	1	1513
Idaho	Boise	A			39.8	0	0	135	389	762	1054	1169	868	719	453	249	92	5890
	Lewiston	A				0	0	133	406	747	961	1060	815	663	408	222	68	5483
	Pocatello	A			35.0	0	0	183	487	873	1184	1333	1022	880	561	317	136	6976
Ill.	Cairo	C			46.4	0	0	28	161	492	784	856	683	523	182	47	0	3756
	Chicago	A			35.1	0	0	90	350	765	1147	1243	1053	868	507	229	58	6310
	Peoria	A			37.3	0	11	86	339	759	1128	1240	1028	828	435	192	41	6087
	Springfield	A			37.7	0	6	83	315	723	1066	1166	958	769	404	171	32	5693
	Springfield	C			39.8	0	0	56	259	666	1017	1116	907	713	350	127	14	5225

[a] Data for United States cities from a publication of the United States Weather Bureau, *Monthly Normal Temperatures, Precipitation and Degree Days*, 1954, are for the period 1921 to 1950 inclusive. Those United States cities for which years are given in Column 3 are not listed in the above publication and the data are those which were computed by the United States Weather Bureau in 1946 and 1947 in accordance with the requirements of the National Joint Committee on Weather Statistics.

[b] Data for airport stations, A, and city stations, C, are both given where available.

[c] Data for Canadian cities were computed by the Meteorological Division, Department of Transport from normal monthly mean temperatures, adjusted for the summer months by a method described by H. C. S. Thom, The Rational Relationship between Heating Degree Days and Temperature, *Monthly Weather Review* Vol. 82, No. 1, January 1954.

[d] For period October to April, inclusive.

Table 2 .... Average Monthly and Yearly Degree Days for Cities in the United States and Canada (Base 65F) (*Continued*)

State	Station	Years	No. of Seasons	Avg. Winter Temp.[4]	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total
Ind....	Evansville.....A			45.1	0	0	59	215	570	871	939	770	589	251	90	6	4360
	Fort Wayne.....A			37.6	0	17	107	377	759	1122	1200	1036	874	516	226	53	6287
	Indianapolis....A			39.0	0	0	79	306	705	1051	1122	938	772	432	176	30	5611
	Indianapolis....C			39.6	0	0	59	247	642	986	1051	893	725	375	140	16	5134
	Royal Center.....	18/19-31/32	14		11	19	116	373	740	1104	1239	976	860	502	245	54	6239
	Terre Haute.....A				0	5	77	295	681	1023	1107	913	725	371	145	24	5366
Iowa...	Charles City.....C			31.2	17	30	151	444	912	1352	1494	1240	1001	537	256	70	7504
	Davenport.....C			37.0	0	7	79	320	756	1147	1262	1044	834	432	175	35	6091
	Des Moines.....A			35.4	5	12	99	355	798	1203	1330	1092	868	438	201	45	6446
	Des Moines.....C			36.4	0	6	89	346	777	1178	1308	1072	849	425	183	41	6274
	Dubuque.....A			34.6	8	28	149	444	882	1290	1414	1187	983	543	267	76	7271
	Keokuk.....	98/99-41/42	44	39.3	1	3	71	303	680	1077	1191	1025	761	397	136	18	5663
	Sioux City.....A				8	17	128	405	885	1290	1423	1170	930	474	228	54	7012
Kan....	Concordia.....C			40.7	0	0	55	277	687	1029	1144	899	725	341	146	20	5323
	Dodge City.....A				0	0	40	262	669	980	1076	840	694	347	135	15	5058
	Iola.....	05/06-40/41	36		0	1	40	236	579	930	1026	817	599	282	98	8	4616
	Topeka.....A				0	8	59	271	672	1017	1125	885	694	326	137	15	5209
	Topeka.....C			42.1	0	0	42	242	630	977	1088	851	669	295	112	13	4919
	Wichita.....A				0	0	32	219	597	915	1023	778	619	280	101	7	4571
Ky.....	Louisville.....A				0	0	51	232	579	871	933	778	611	285	94	5	4439
	Louisville.....C			45.1	0	0	41	206	549	849	911	762	605	270	86	0	4279
	Lexington.....A				0	0	56	259	636	933	1008	854	710	368	140	15	4979
La.....	New Orleans.....A			60.6	0	0	0	7	169	308	364	248	190	31	0	0	1317
	New Orleans.....C			61.6	0	0	0	5	141	283	341	223	163	19	0	0	1175
	Shreveport.....A				0	0	0	53	305	490	550	386	272	61	0	0	2117
Me.....	Eastport.....C			31.5	141	136	261	521	798	1206	1333	1201	1063	774	524	288	8246
	Greenville.....	07/08-40/41															
		42/43-45/46	38		69	113	315	643	1012	1464	1625	1443	1251	842	468	194	9439
	Portland.....A			33.0	15	56	199	515	825	1237	1373	1218	1039	693	394	117	7681
Md.....	Baltimore.....A			44.1	0	0	50	278	582	908	955	840	676	378	115	5	4787
	Baltimore.....C			44.3	0	0	29	207	489	812	880	776	611	326	73	0	4203
Mass...	Boston.....A				0	7	77	315	618	998	1113	1002	849	534	236	42	5791
	Fitchburg.....	98/99-40/41	43		12	29	144	432	774	1139	1240	1137	940	572	254	70	6743
	Nantucket.....A				22	34	111	372	615	924	1020	949	880	642	394	139	6102
Mich...	Alpena.....C			29.6	50	85	215	530	864	1218	1358	1263	1156	762	437	135	8073
	Detroit Willow Run.....A				0	10	96	393	759	1125	1231	1089	915	552	244	55	6469
	Detroit City.....A				0	8	96	381	747	1101	1203	1072	927	558	251	60	6404
	Escanaba.....C			27.3	62	95	247	555	933	1321	1473	1327	1203	804	471	166	8657
	Grand Rapids....A				14	29	144	462	822	1169	1287	1154	1008	606	301	79	7075
	Grand Rapids....C			36.0	0	20	105	394	756	1107	1215	1086	939	546	248	58	6474
	Houghton.....	00/01-40/41															
		42/43-45/46	45		70	94	268	582	965	1355	1535	1421	1251	820	474	195	9030
	Lansing.....A				13	33	140	455	813	1175	1277	1142	986	591	287	70	6982
	Ludington.....	12/13-40/41	29		41	55	182	472	794	1135	1271	1183	1056	698	418	153	7458
	Marquette.....C			28.3	69	87	236	543	933	1299	1435	1291	1181	789	477	189	8529
	Sault Ste. Marie.A			26.0	109	126	298	639	1005	1398	1587	1442	1302	846	499	224	9475
Minn...	Duluth.....A				56	91	298	651	1140	1606	1758	1512	1327	846	474	178	9937
	Duluth.....C			24.3	66	91	277	614	1092	1550	1696	1448	1252	801	487	200	9574
	Minneapolis.....A				8	17	157	459	960	1414	1562	1310	1057	570	252	80	7853
	Moorhead.....	98/99-40/41	43		20	47	240	607	1105	1609	1815	1555	1225	679	327	98	9327
	Saint Paul.....A				12	21	154	459	951	1401	1553	1305	1051	564	256	77	7804
Miss....	Corinth.....	09/10-40/41	32		0	1	13	142	418	669	696	570	396	149	32	1	3087
	Meridian.....A				0	0	0	90	338	528	561	413	309	85	9	0	2333
	Vicksburg.....C			56.8	0	0	0	51	268	456	507	374	273	71	0	0	2000
Mo.....	Columbia.....A			41.1	0	6	62	262	654	989	1091	876	698	326	135	14	5113
	Hannibal.....	98/99-40/41	43		1	3	66	288	652	1037	1139	980	710	374	128	15	5393
	Kansas City.....A				0	0	44	240	621	970	1085	851	666	292	111	8	4888
	Saint Louis.....A			42.3	0	0	45	233	600	927	1017	820	648	297	101	11	4699
	Saint Louis.....C			43.6	0	0	38	202	570	893	983	792	620	270	94	7	4469
	Springfield.....A				0	8	61	249	615	908	1001	790	632	295	118	16	4693
Mont...	Billings.....A			34.9	8	20	194	497	876	1172	1305	1089	958	564	304	119	7106
	Havre.....C			28.4	20	38	270	564	1023	1383	1513	1291	1076	597	313	125	8213
	Helena.....A				36	66	320	617	999	1311	1469	1165	1017	654	299	197	8250
	Helena.....C			31.6	51	78	359	598	969	1215	1438	1114	992	660	427	225	8126
	Kalispell.....A			31.6	47	83	326	639	990	1249	1386	1120	970	639	391	215	8055

Table 2 .... Average Monthly and Yearly Degree Days for Cities in the United States and Canada (Base 65F) (*Continued*)

State	Station	Years	No. of Seasons	Avg. Winter Temp.[4]	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total
Mont...	Miles City.....A			27.6	6	11	187	525	966	1373	1516	1257	1048	570	285	106	7850
	Missoula.....A				22	57	292	623	993	1283	1414	1100	939	609	365	176	7873
Neb....	Drexel.....	15/16-25/26	11		4	6	95	405	788	1271	1353	1096	843	493	219	38	6611
	Lincoln.....A			35.6	0	12	82	340	774	1144	1271	1030	822	401	190	38	6104
	Lincoln.....C			37.0	0	7	79	310	741	1113	1240	1000	794	377	172	32	5865
	North Platte.....A				7	11	120	425	846	1172	1271	1016	887	489	243	59	6546
	Omaha.....A				0	5	88	331	783	1166	1302	1058	831	389	175	32	6160
	Valentine.....C			33.6	11	10	145	461	891	1212	1361	1100	970	543	288	83	7075
Nev....	Reno.....A				27	61	165	443	744	986	1048	804	756	519	318	165	6036
	Tonopah.....				0	5	96	422	723	995	1082	860	763	504	272	91	5813
	Winnemucca.....A			38.0	0	17	180	508	822	1085	1153	854	794	546	299	111	6369
N. H...	Concord.....A				11	57	192	527	849	1271	1392	1226	1029	660	316	82	7612
N. J....	Atlantic City....C			42.3	0	0	29	230	507	831	905	829	729	468	189	24	4741
	Cape May.....	98/99-31/32	34		1	2	38	221	527	852	936	876	737	459	188	33	4870
	Newark.....A				0	0	47	301	603	961	1039	932	760	450	148	11	5252
	Sandy Hook.....	15/16-40/41	26	41.2	1	2	40	268	579	921	1016	973	833	499	206	31	5369
	Trenton.....C			42.0	0	0	55	285	582	930	1004	904	735	429	133	11	5068
N. M...	Albuquerque.....A				0	0	10	218	630	899	970	714	589	289	70	0	4389
	Roswell.....A			49.1	0	0	8	156	501	750	787	566	443	185	28	0	3424
	Santa Fe.....	98/99-45/46	43		12	15	129	451	772	1071	1094	892	786	544	297	60	6123
N. Y...	Albany.....A				0	24	139	443	780	1197	1318	1179	989	597	246	50	6962
	Albany.....C			35.2	0	6	98	388	708	1113	1234	1103	905	531	202	31	6319
	Binghamton.....A				16	63	192	518	834	1228	1342	1215	1051	672	318	88	7537
	Binghamton.....C			34.7	0	36	141	428	735	1113	1218	1100	927	570	240	48	6556
	Buffalo.....A				16	30	122	433	753	1116	1225	1128	992	636	315	72	6838
	Canton.....	06/07-45/46	40	29.5	27	61	219	550	898	1368	1516	1385	1139	695	340	107	8305
	Ithaca.....	99/00-42/43	44	34.9	17	40	156	451	770	1129	1236	1156	978	606	292	83	6914
	New York La Guardia....A				0	0	28	250	546	908	992	907	760	447	141	10	4989
	New York.....C			41.1	0	0	39	263	561	908	995	901	753	456	153	18	5050
	New York Central Park Obs.....				0	0	31	250	552	902	1001	910	747	435	130	7	4965
	Oswego.....C			34.4	20	39	139	430	738	1132	1249	1134	995	654	355	90	6975
	Rochester.....A				9	34	133	440	759	1141	1249	1148	992	615	289	54	6863
	Syracuse.....A				0	29	117	396	714	1113	1225	1117	955	570	247	37	6520
N. C...	Asheville.....C			46.1	0	0	50	262	552	769	794	678	572	285	105	5	4072
	Charlotte.....A				0	0	7	147	438	682	704	577	449	172	29	0	3205
	Hatteras.....C				0	0	0	63	244	451	527	487	394	171	25	0	2392
	Manteo.....	04/05-28/29	25		0	0	7	113	358	595	642	594	469	240	75	7	3109
	Raleigh.....A				0	0	16	149	438	701	732	613	477	202	41	0	3389
	Raleigh.....C			50.0	0	0	10	118	387	651	691	577	440	172	29	0	3075
	Wilmington.....A			54.6	0	0	0	73	288	508	533	463	347	104	7	0	2323
N. D...	Bismarck.....A			22.9	29	37	227	598	1098	1535	1730	1464	1187	657	355	116	9033
	Devils Lake.....C			21.7	47	61	276	654	1197	1668	1866	1576	1314	750	394	137	9940
	Grand Forks.....	12/13-40/41															
		42/43-45/46	33		32	60	274	663	1160	1681	1895	1608	1298	718	359	123	9871
	Williston.....C			24.5	29	42	261	605	1101	1528	1705	1442	1194	663	360	138	9068
Ohio...	Cincinnati.....A				0	6	77	295	648	973	1029	871	732	392	149	23	5195
	Cincinnati.....C			43.0	0	0	42	222	567	880	942	812	645	314	108	0	4532
	Cincinnati Abbe Obs.....				0	0	56	263	612	930	989	846	682	347	132	13	4870
	Cleveland.....A				0	10	75	340	699	1057	1132	1019	874	531	223	46	6006
	Cleveland.....C			37.2	0	9	60	311	636	995	1101	977	846	510	223	49	5717
	Columbus.....A			38.2	0	8	69	337	693	1032	1094	946	781	444	180	31	5615
	Columbus.....C			40.4	0	0	59	299	654	983	1051	907	741	408	153	22	5277
	Dayton.....A				0	5	74	324	693	1032	1094	941	781	435	179	39	5597
	Sandusky.....C			38.0	0	0	66	327	684	1039	1122	997	853	513	217	41	5859
	Toledo.....A			35.7	0	12	102	387	756	1119	1197	1056	905	555	245	60	6394
Okla...	Broken Arrow.....	18/19-30/31	13		0	0	28	169	513	805	881	646	506	212	61	5	3826
	Oklahoma City..A				0	0	14	154	480	769	865	650	490	182	40	03	644
	Oklahoma City..C			47.9	0	0	12	149	459	747	843	630	472	169	38	0	3519
Ore.....	Baker.....C			35.2	25	47	255	518	852	1138	1268	972	837	591	384	200	7087
	Medford.....A				0	0	77	326	624	822	862	627	552	381	207	69	4547

Table 2 . . . . Average Monthly and Yearly Degree Days for Cities in the United States and Canada (Base 65F) (Continued)

State	Station	Years	No. of Seasons	Avg. Winter Temp.[4]	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total
Ore.	Portland A			44.3	25	22	116	319	585	750	856	658	570	396	242	93	4632
	Portland C			46.1	13	14	85	280	534	701	791	594	515	347	199	70	4143
	Roseburg C			46.7	14	10	98	288	531	694	744	563	508	366	223	83	4122
Pa.	Erie C			37.3	0	17	76	352	672	1020	1128	1039	911	573	273	55	6116
	Harrisburg A				0	0	69	308	630	964	1051	921	750	423	128	14	5258
	Philadelphia A			41.4	0	0	47	269	573	902	986	879	704	402	104	0	4866
	Philadelphia C			42.7	0	0	33	219	516	856	932	837	667	369	93	0	4523
	Pittsburg Allegheny A				0	6	78	336	678	1004	1073	955	784	447	167	27	5555
	Pittsburgh Greater Pittsburgh A			38.7	0	20	94	377	720	1057	1116	986	818	486	195	36	5905
	Pittsburgh C			40.9	0	0	56	298	612	924	992	879	735	402	137	13	5048
	Reading C			41.2	0	5	57	285	588	936	1017	902	725	411	123	11	5060
	Scranton C			37.7	0	18	115	389	693	1057	1141	1028	859	516	196	35	6047
R. I.	Block Island A			40.1	6	21	88	330	591	927	1026	955	865	603	335	96	5843
	Narragansett Pier	98/99–17/18	20		1	26	121	366	691	1012	1113	1074	916	622	342	113	6397
	Providence A				0	26	107	381	672	1035	1125	1019	874	570	258	58	6125
	Providence C			37.5	0	7	68	330	624	986	1076	972	809	507	197	31	5607
S. C.	Charleston A			55.0	0	0	0	52	270	456	472	379	281	63	0	0	1973
	Charleston C			57.4	0	0	0	34	214	410	445	363	260	43	0	0	1769
	Columbia A				0	0	0	82	338	558	566	468	340	83	0	0	2435
	Columbia C			54.4	0	0	0	76	308	524	538	443	318	77	0	0	2284
	Due West	21/22–31/32	11		0	0	9	142	393	594	651	491	411	158	39	2	2890
	Greenville A			49.2	0	0	10	131	411	648	673	552	442	161	32	0	3060
S. D.	Huron A				10	16	149	472	975	1407	1597	1327	1032	558	279	80	7902
	Pierre	98/99–40/41, 42/43–45/46	47		4	11	136	438	887	1317	1460	1253	971	516	238	52	7283
	Rapid City A				32	24	193	500	891	1218	1361	1151	1045	615	357	148	7535
Tenn.	Chattanooga A			47.8	0	0	24	169	477	710	725	588	467	179	45	0	3384
	Knoxville A			46.8	0	0	33	179	498	744	760	630	500	196	50	0	3590
	Memphis A			50.0	0	0	17	126	432	673	725	574	427	139	24	0	3137
	Memphis C			51.1	0	0	13	98	392	639	716	574	423	131	20	0	3006
	Nashville A				0	0	22	154	471	725	778	636	498	186	43	0	3513
Texas	Abilene A				0	0	5	98	350	595	673	479	344	113	0	0	2657
	Amarillo A				0	0	37	240	594	859	921	711	586	298	99	0	4345
	Austin A				0	0	0	30	214	402	484	322	211	50	0	0	1713
	Brownsville A				0	0	0	0	59	159	219	106	74	0	0	0	617
	Corpus Christi A				0	0	0	0	113	252	330	192	118	6	0	0	1011
	Dallas A				0	0	0	53	299	518	607	432	288	75	0	0	2272
	Del Rio A				0	0	0	26	188	371	419	235	147	21	0	0	1407
	El Paso A				0	0	0	70	390	628	670	445	330	110	0	0	2641
	Fort Worth A				0	0	0	58	299	533	622	446	308	90	5	0	2361
	Fort Worth Amon Carter Fld.				0	0	0	57	299	524	619	432	326	81	0	0	2338
	Galveston A				0	0	0	0	132	286	362	249	176	28	0	0	1233
	Galveston C			61.7	0	0	0	0	131	271	356	247	176	30	0	0	1211
	Houston A			60.1	0	0	0	7	181	321	394	265	184	36	0	0	1388
	Houston C			61.0	0	0	0	0	162	303	378	240	166	27	0	0	1278
	Palestine C			57.1	0	0	0	45	260	440	531	363	265	71	0	0	1980
	Port Arthur A				0	0	0	20	218	349	406	274	211	39	0	0	1517
	Port Arthur C			61.2	0	0	0	8	170	315	381	258	181	27	0	0	1340
	San Antonio A				0	0	0	25	201	374	462	293	190	34	0	0	1579
	Taylor	01/02–40/41	40		0	0	2	56	234	462	494	375	214	64	8	0	1909
Utah	Modena	00/01–45/46	46	38.3	6	11	156	499	832	1142	1190	944	816	567	338	97	6598
	Salt Lake City A			38.3	0	0	88	381	771	1039	1194	885	741	453	233	81	5866
	Salt Lake City C			40.0	0	0	61	330	714	995	1119	857	701	414	208	64	5463
Vt.	Burlington A				19	47	172	521	858	1308	1460	1313	1107	681	307	72	7865
	Northfield	98/99–42/43	45		62	112	283	602	947	1389	1524	1384	1176	754	405	166	8804
Va.	Cape Henry C			49.2	0	0	0	120	366	648	698	636	512	267	60	0	3307
	Lynchburg A				0	0	49	236	531	809	846	722	584	289	82	5	4153
	Norfolk A				0	0	9	152	408	688	729	644	500	265	59	0	3454
	Norfolk C			49.3	0	0	5	118	354	636	679	602	464	220	41	0	3119
	Richmond A				0	0	33	210	498	791	828	706	550	271	66	0	3955
	Richmond C			47.0	0	0	31	181	456	750	787	675	529	254	57	0	3720
	Wytheville	02/03–40/41	39		7	13	82	332	662	918	945	836	677	410	168	35	5103

Table 2 . . . . Average Monthly and Yearly Degree Days for Cities in the United States and Canada (Base 65F) (Concluded)

State or Province	Station	Years	No. of Seasons	Avg. Winter Temp.[4]	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total
Wash.	North Head L.H. Reservation			46.4	239	205	234	341	486	638	704	685	598	492	406	285	5211
	Seattle C			46.3	49	45	134	329	540	679	753	602	558	396	246	107	4438
	Seattle-Tacoma A			45.1	75	70	192	412	633	781	862	675	636	477	307	155	5275
	Spokane A				17	28	205	508	879	1113	1243	988	834	561	330	146	6852
	Tacoma C			44.9	66	62	177	375	579	719	797	636	595	435	282	143	4866
	Tatoosh Island C			45.4	295	288	315	406	528	648	713	610	629	525	437	330	5724
	Walla Walla C				0	0	93	308	675	890	1023	748	564	338	171	38	4848
	Yakima A				0	7	150	446	807	1066	1181	862	660	408	205	53	5845
W. Va.	Elkins A			39.4	9	31	122	412	726	995	1017	910	797	477	224	53	5773
	Parkersburg C			42.9	0	0	56	272	600	896	949	826	672	347	119	13	4750
Wis.	Green Bay A			29.8	32	58	183	515	945	1392	1516	1336	1132	696	347	107	8259
	La Crosse A			30.5	11	20	152	447	921	1380	1528	1280	1035	552	250	74	7650
	Madison A				13	31	150	459	891	1302	1423	1207	1008	579	272	82	7417
	Madison C			31.4	10	30	137	419	884	1287	1417	1207	1011	573	266	79	7300
	Milwaukee A			29.0	20	32	134	428	831	1218	1336	1142	983	621	351	109	7205
	Milwaukee C			33.4	11	24	112	397	795	1184	1302	1117	961	606	335	100	6944
	Wausau	15/16–40/41	26		26	58	216	568	952	1427	1594	1381	1147	680	315	100	8494
Wyo.	Cheyenne A				33	39	241	577	897	1125	1225	1044	1029	717	462	173	7562
	Lander A				7	23	244	632	1050	1383	1494	1179	1045	687	396	163	8303
	Yellowstone Park	04/05–40/41	37		125	173	424	759	1079	1388	1484	1252	1165	841	603	334	9605
Alta.	Calgary	1921–50	30		110	170	410	710	1110	1430	1530	1350	1200	770	460	270	9520
	Edmonton	1921–50	30		90	180	440	750	1220	1660	1780	1520	1290	760	410	220	10320
	Lethbridge	1921–50	30		60	100	350	620	1030	1330	1450	1290	1120	690	400	210	8650
	Medicine Hat	1921–50	30		20	50	300	600	1070	1440	1590	1380	1130	620	320	130	8650
B.C.	Prince George	1921–50	30		180	220	440	730	1100	1450	1560	1290	1070	730	470	280	9500
	Prince Rupert	1921–50	30		270	240	340	510	680	860	910	810	790	650	500	350	6910
	Vancouver	1921–45	25		60	60	200	410	620	790	850	710	650	460	290	130	5230
	Victoria	1921–50	30		160	150	230	410	600	730	800	660	620	470	350	230	5410
Man.	Brandon	1921–50	30		60	100	350	730	1290	1810	2010	1730	1440	820	420	170	10930
	Churchill	1926–50	25		310	370	650	1110	1750	2300	2520	2270	2150	1590	1100	690	16810
	Winnipeg	1921–50	30		40	70	300	690	1250	1770	2000	1710	1440	810	400	150	10630
N.B.	Fredericton	1921–50	30		50	70	250	600	940	1410	1570	1410	1180	780	420	150	8830
	Moncton	1939–50	12		40	80	240	580	890	1340	1510	1340	1180	830	460	210	8700
	Saint John	1921–50	30		110	110	250	530	830	1250	1400	1270	1100	780	500	250	8380
Nfld.	Corner Brook	1933–50	18		90	140	320	640	890	1230	1410	1360	1240	900	640	350	9210
	Gander	1937–50	14		130	160	320	660	920	1230	1430	1320	1270	970	650	380	9440
	Goose Bay	1941–51	10		130	220	440	840	1220	1740	2020	1710	1530	1110	770	410	12140
	St. John's		19		170	160	320	580	820	1100	1270	1230	1160	910	680	380	8780
N.W.T.	Aklavik	1926–50	25		280	460	800	1400	2040	2500	2580	2310	2280	1690	1050	480	17870
	Fort Norman		25		170	350	700	1220	1940	2460	2550	2190	2040	1390	730	280	16020
N.S.	Halifax	1921–50	30		50	70	180	470	740	1120	1260	1180	1050	760	480	210	7570
	Sydney	1921–50	30		60	80	220	510	780	1130	1310	1280	1160	850	570	270	8220
	Yarmouth	1921–50	30		110	120	230	480	720	1040	1180	1100	1010	750	510	270	7520
Ont.	Fort William	1921–50	30		90	120	350	690	1140	1590	1780	1550	1360	890	550	240	10350
	Hamilton		21		10	10	120	430	760	1130	1270	1160	1000	620	320	60	6890
	Kapuskasing	1921–50	30		110	170	410	730	1270	1780	2060	1770	1570	1030	600	240	11790
	Kingston	1921–50	30		30	40	160	500	820	1250	1420	1290	1110	710	380	100	7810
	London	1921–50	30		20	40	150	490	840	1200	1320	1210	1040	650	330	90	7380
	Ottawa	1921–50	30		30	60	210	590	960	1450	1640	1480	1230	730	340	80	8830
	Toronto	1921–50	30		20	30	140	460	770	1130	1260	1160	1020	640	320	70	7020
P.E.I.	Charlottetown	1921–50	30		40	60	200	510	820	1230	1430	1340	1180	840	510	220	8380
Que.	Arvida	1931–50	20		60	110	330	690	1100	1670	1880	1660	1410	880	480	170	10440
	Montreal	1921–50	30		10	40	180	530	890	1370	1540	1370	1150	700	300	50	8130
	Quebec City	1921–50	30		20	70	250	610	990	1470	1640	1460	1250	810	400	100	9070
	Sherbrooke	1921–50	30		20	70	240	590	920	1400	1560	1410	1190	750	370	90	8610
Sask.	Prince Albert	1921–50	30		70	140	410	780	1350	1870	2060	1750	1500	850	440	210	11430
	Regina	1921–50	30		70	110	370	750	1290	1740	1940	1680	1420	790	420	190	10770
	Saskatoon	1921–50	30		60	110	380	760	1320	1790	1990	1710	1440	800	420	180	10960
	Swift Current	1921–50	30		50	90	340	680	1170	1550	1710	1490	1260	730	400	190	9660
Y. T.	Dawson	1921–50	30		170	320	660	1170	1890	2410	2510	2160	1830	1100	570	250	15040

appropriately. When data on degree days are available these may be used in the calculation of average outdoor temperature for use in Equation 2 or 3. (See definition of degree day, Chapter 4.)

It may be noted that a portion of Equation 3, $(t - t_o)N$ will, when divided by 24, correspond exactly to the degree days for the period, provided that $t = 65$ F, the usual basis for calculating degree days (see discussion of degree days in following sections). Some engineers substitute degree days for the period, multiplied by 24, for the value of $(t - t_o)N$ in Equation 3. Since this amounts to the basis for degree-day calculations, assuming that $t = 65$ F, some allowance is thus made for the fact that F will frequently be overestimated, when H is taken as the maximum, or design, heat load.

Example 1: A residence building is to be heated to 70 F from 6 a.m. to 10 p.m. and 65 F from 10 p.m. to 6 a.m. The calculated hourly heat loss is 120,000 Btu per hour based on 70 F inside at −10 F outdoors. If the building is to be heated by metered steam, how many pounds would be required during an average heating season?

Solution: The heating value of steam may be taken as 1000 Btu per lb, and since it is purchased steam, the efficiency can be assumed as 100 percent. Assume average outdoor temperature as 36.4 F. The average inside temperature is:

$$\frac{(16 \times 70) + (8 \times 65)}{24} = 68.3 \text{ F.}$$

Substituting in Equation 1

$$F = \frac{120{,}000(68.3 - 36.4)5088}{1.00[70 - (-10)]1000} = 243{,}000 \text{ lb.}$$

Example 2: What would be the fuel cost to heat the building in *Example 1* during an average heating season, using stoker-fired bituminous coal at $14.00 per ton having a calorific value of 13,000 Btu per lb, assuming that the seasonal efficiency of the system is 65 percent?

Solution: Substituting in Equation 1

$$F = \frac{120{,}000(68.3 - 36.4)5088}{0.65[70 - (-10)]13{,}000} = 28{,}800 \text{ lb.}$$

The fuel cost is then $(28{,}800 \div 2000)\ (14) = \$201.00$

Example 3: What will be the estimated fuel cost per year of heating a building with gas, assuming that the calculated hourly heat loss is 92,000 Btu based on 0 F, which includes 26,000 Btu for infiltration? The design temperatures are 0 F and 72 F. The normal heating season is 210 days, and the average outdoor temperature during the heating season is 36.4 F. The seasonal efficiency will be 80 percent. The heating plant will be thermostatically controlled, and a temperature of 65 F will be maintained from 11 p.m. to 7 a.m. Assume that the price of gas is 7.5 cents per 100,000 Btu of fuel consumption, and disregard the loss of heat through open windows and doors.

Solution: The average hourly temperature is

$$t_o = \frac{(72 \times 16) + (65 \times 8)}{24} = 69.7 \text{ F.}$$

The maximum hourly heat loss is

$$H = 92{,}000 \text{ Btu.}$$

The seasonal heat loss is

$$M = \frac{92{,}000(69.7 - 36.4) \times 24 \times 210}{100{,}000 \times 0.80 \times (72 - 0)}$$

$$= 2697 \text{ hundred thousand Btu.}$$

The estimated seasonal fuel cost will be

$$2697 \times \$0.075 = \$202.00$$

It should be noted that savings from night setback may not result as calculated. Room temperature may not decrease and combustion efficiency may be poor during morning pickup.[10]

Several time-saving procedures have been devised for quickly estimating the hourly Btu loss of one- and two-story residences in order that fuel estimates can be predicted more quickly from Equation 1. A graphical method of calculating heat losses has been developed[11] which makes possible a quick solution if the gross wall, ceiling, or floor areas and respective transmission coefficients are known.

The Federal Housing Administration has originated a short-cut formula for residential heat loss determinations which makes use of the floor area and three selected *transmission coefficients*. The formula was developed to apply to detached houses approximately rectangular in shape with total exterior door and window areas equal to about 25 percent of the floor area, and with a floor area not greater than about 1500 sq ft. Equation 4 is for a one-story residence, and Equation 5 is intended for two-story structures.

$$H_1 = A(G + U_w + U_c + U_f)(t_i - t_o) \quad (4)$$

$$H_2 = A(G + 1.2U_w + 0.5U_c + 0.5U_f)(t_i - t_o) \quad (5)$$

where

H_1 = heat loss from one-story residence, Btu per hour.
H_2 = heat loss from two-story residence, Btu per hour.
A = floor area, square feet, measured to the inside faces of enclosing walls and is the sum of the following areas: (1) all the area on each principal floor level; (2) the area of all finished habitable attic rooms, including bathrooms, toilet compartments, closets, and halls; (3) all other areas intended to be heated and not located in the basement.
G = glass and infiltration factor for ordinary construction: (0.45 for no weatherstripping or storm windows), (0.40 for weatherstripping), (0.30 for storm windows with or without weatherstripping).
U_w = coefficient of transmission for outside wall.
U_c = coefficient of *transmission for ceiling.*
U_f = coefficient of transmission for floor.
t_i = indoor design temperature, Fahrenheit degrees.
t_o = outdoor design temperature, Fahrenheit degrees.

Notes for application of Equations 4 and 5.

1. The calculation of heat loss from heated spaces into adjacent spaces such as attics, basementless areas, and heated or unheated garages shall be based on the assumption that the temperature of such adjacent spaces is the same as the outdoor design temperature.

2. For all floors over basements or other warmed spaces assume $U_f = 0$.

3. For structures having concrete slab floors laid on the ground a modified application of the formula may be made. Assume $U_f = 0$ and calculate the heat loss in accordance with the check formula. Then add the slab loss determined in accordance with the procedure developed by the National Bureau of Standards and described in *Building Materials and Standards Report* 103.

4. No basement area is to be included in the formula calculation. If finished habitable rooms in the basement are to be heated, the additional heat loss should be calculated separately and added to the amount obtained by the formula.

Both the graphical method[11] and FHA formulas, when used within the limitations established, have been found to give reasonably accurate results for the average residence, but if precise estimates are required, the heat loss should be calculated as outlined in Chapter 12.

In the case of gravity warm air heating installations, the load was formerly expressed in square inches of leader pipe which can be converted into Btu per hour by multiplying the square inches of leader area by 111, 167, and 200 for first, second, and third floors, respectively.

DEGREE-DAY METHOD

This method is based on consumption data which have been taken from buildings in operation, and the results have been computed on a degree-day basis. While this method may not be as theoretically precise as the Calculated Heat Loss Method, it is considered by many to be of more value for practical use.

The amount of heat required in a building depends upon the outdoor temperature, if other variables are eliminated. Theoretically it is proportional to the difference between the outdoor and indoor temperatures. The American Gas Association[12] determined from records in the heating of residences that the gas consumption varied directly as the degree days, or as the difference between 65 F and the mean outdoor temperature. In other words, on a day when the mean temperature was 20 deg below 65 F, twice as much gas was consumed as on a day when the temperature was 10 deg below 65 F. For any one day, when the mean temperature is less than 65 F, there are as many degree days as there are degrees difference in temperature between the mean temperature for the day and 65 F. Degree days may be calculated on other than the 65 F base for use mainly for warehouse and other industrial spaces in which temperatures to be maintained are considerably below the 68 to 72 F range. They are listed in a later section of this chapter.

Studies made by the National District Heating Association[13] of the metered steam consumption of 163 buildings located in 22 different cities, and served with steam from a district heating company, substantiate the approximate correctness of the 65 F base chosen by the gas industry.

Table 2 lists the average number of degree days that have occurred over a long period of years, by months, and the yearly totals for various cities in the United States and Canada. The number of degree days for United States cities were calculated by taking the difference between 65 F and the daily mean temperature computed as half the total of the daily maximum and the daily minimum temperatures. The monthly averages were obtained by adding daily degree days for each month each year and dividing by the number of days in the month; then totaling the respective calendar monthly averages for the number of years indicated and dividing by the number of years. *The total or long term yearly* average degree-day value is the summation of the 12 monthly averages. Degree days for Canadian cities were supplied by the Canadian Meteorological Division of the Department of Transport, and were computed from the mean temperature normals on record for the various stations. (See Note c, Table 2.)

Any attempt to apply the degree-day method of estimating fuel consumption for less than one month would be of very little value. It should be noted that this method of calculation is based on a long term average and cannot be expected to coincide with any single year in calculating fuel requirement. Individual yearly degree-day calculations may vary as much as 20 percent above and below the long term average.

If the degree days occurring each day are totaled for a reasonably long period, the fuel consumption during that period as compared with another period may be assumed to be in direct proportion to the number of degree days in the two periods. Consequently, for a given installation, the fuel consumption can be calculated in terms of fuel used per degree day for any sufficiently long period, and compared with similar ratios for other periods to determine the relative operating efficiencies with the outdoor temperature variable eliminated.

Such results should be used with some reservation since it is possible to have wide variations, for example, as between early and late winter periods.[14]

Computation and Application

The general equation for calculating the probable fuel consumption by the degree-day method is:

$$F = U \times N \times D \times C_f \quad (6)$$

where

F = fuel consumption for the estimate period.
U = unit fuel consumption, or quantity of fuel used per (degree day) (*building load unit*).
N = number of *building load units* (when available, use calculated hourly heat loss instead of actual amount of radiation installed).
D = number of degree days for the estimate period.
C_f = temperature-correction factor from Table 3.

Table 3 .... Correction Factors for Outdoor Design Temperatures[a]

Outdoor Design Temp F....	−20	−10	0	+10	+20
Correction factor.....	0.778	0.875	1.000	1.167	1.400

[a] The multipliers in Table 3, which are high for mild climates and low for cold regions, are not in error as might appear. The unit figures in Tables 4, 5, and 6 are per square foot of radiator or thousand Btu heat loss per degree day. For equivalent buildings and heating seasons, those in warm climates have lower design heat losses and smaller radiator quantities than those in cold cities. Consequently, the unit figure in quantity of fuel per (square foot of radiator) (degree day), is larger for warm localities than for colder regions. Since the northern cities have more radiator surface per given building and a higher seasonal degree-day total than cities in the south, the total fuel per season will be larger for the northern city.

Values of N depend on the particular building for which the estimate is being prepared and must be found by surveying plans, by observation, or by measurement of the building. Values of U for use in this equation are the unit fuel consumptions per degree day, obtained as a result of the collection of operating information and listed in Tables 4, 5, and 6. Attention is directed to the nature of these units in the next following sections.

Unit Fuel Consumptions per Degree Day

The quantity of fuel used per degree day in a given heating plant can be reduced to a unit basis in terms of quantity of fuel or steam per degree day per thousand Btu hourly heat loss at design conditions. A less frequently used basis is quantity of fuel per (degree day) (square foot of floor area). In fact any convenient unit can be used to relate the consumption *to the degree day and to the building.*

The choice of these units requires explanation, and some discrimination and judgment. If the volume basis is used, the net heated space is preferable to the gross building cubage, since gross cubage includes outer walls and certain portions of attic and basement space that are usually unheated. In the absence of data on net heated volume, a value of 80 percent of the gross volume may be used to obtain the estimated net heated volume. The volume basis has been rather widely used primarily because it is simple to apply. In industrial buildings it is usually easier to obtain the correct

Table 4 .... Unit Fuel Consumption Constants (U) for Gas[a]

Based on 0 F Outdoor Temperature, 70 F Indoor Temperature

Heating Value of Gas Btu per Cu Ft	Hot Water			Steam			Warm Air	
	Cu Ft Gas per Degree Day per Sq Ft EDR			Cu Ft Gas per Degree Day per Sq Ft EDR			Cu Ft Gas per Degree Day per 1000 Btu Hourly Design Heat Loss	
	Up to 500 Sq Ft	500 to 1200 Sq Ft	Over 1200 Sq Ft	Up to 300 Sq Ft	300 to 700 Sq Ft	Over 700 Sq Ft	Gravity	Fan Systems
500	0.149	0.142	0.134	0.254	0.242	0.231	0.896	0.861
535	0.139	0.132	0.126	0.237	0.226	0.216	0.840	0.805
800	0.094	0.087	0.085	0.159	0.151	0.144	0.560	0.538
1000	0.075	0.071	0.068	0.127	0.121	0.116	0.449	0.431
	Gas Consumption in Therms per Degree Day							
1 Therm = 100,000 Btu	0.000743	0.000709	0.000675	0.00127	0.00121	0.00116	0.00450	0.00430

[a] Abstracted from *Comfort Heating*, American Gas Association, 1938, and 5 percent added for operation without night reduction of temperature.

volume of a given building than to measure and evaluate the heating capacity of its heating system, or calculate its maximum hourly Btu loss. The comparison of buildings on a straight volume basis does not allow for variation in exposure, type of construction, ratio of exposed area to cubical contents, and type of occupancy. It is inaccurate for estimating purposes unless the buildings are of very similar nature.

The calculated heat loss or the heating capacity of the installed radiation may be used as the unit. The use of the heating capacity of the installed radiation is of questionable value when referring to heat-transfer surfaces used in warm air furnace or central air-conditioning systems. Where steam or hot water radiation is already installed, care should be exercised when using the installed radiation as the basis for estimating, since actual installed radiation may differ considerably from the exact radiation requirements. In view of all these considerations, it is believed that the unit based on *thousands of Btu of hourly calculated heat loss for the design hour* is probably the most desirable.

Estimating Gas Consumption

Values of the Unit Fuel Consumption Constant U for gas are given in Table 4 for various gas heating values, and different types and sizes of heating systems. They are based on an indoor design temperature of 70 F and an outdoor design temperature of 0 F, and apply only to these conditions. For other outdoor design conditions corrections must be made by applying factors given in Table 3.

The factors in Table 4, as corrected if necessary, are satisfactory for regions having 3500 to 6500 degree days per heating season. In regions with less than 3500 degree days the unit gas consumption is higher than given; where over 6500, the unit is less than given. Ten percent addition or deduction in these cases is recommended by A.G.A. publications. This table cannot be used for making estimates for industrial buildings where low inside temperatures are maintained.

For gas heat values other than those given in Table 4, simply interpolate or extrapolate. It will also be noted that Table 4 applies only to small installations. Since the unit gas consumption becomes smaller as the installation becomes larger, the values in the table should be used with care, if at all, for large gas-burning installations.

Example 4: Estimate the gas required to heat a building located in Chicago, Ill., where the heating season has 6310 degree days and the gas heating value is 800 Btu per cu ft. The calculated heating requirements are 1000 sq ft of hot water radiation based on design temperature of −10 F and 70 F.

Solution: From Table 4, the fuel consumption for a design temperature of 0 F with 800 Btu gas is found to be 0.087 cu ft of gas per (degree day) (square foot of hot water radiation).

Table 5 .... Unit Fuel Consumption[a] Constants (U) for Oil[b]

Based on 0 F Outside Temperature, 70 F Inside Temperature

Unit[c]	Efficiency in Percent	
	70	80
Gal oil per sq ft steam radiator	0.00105	0.00092
Gal oil per sq ft hot water radiator	0.00066	0.00058
Gal oil per 1000 Btuh heat loss	0.00437	0.00383

[a] Based on a heating value of 141,000 Btu per gallon.

[b] Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss. Seven percent added for operation without night reduction of temperature and change to heating value of 141,000 Btu per gallon.

[c] Per degree day.

Table 6 .... Unit Fuel Consumption[a] Constants (U) for Coal[b]

Based on 0 F Outdoor Temperature, 70 F Inside Temperature

Unit[c]	Efficiency in Percent				
	40	50	60	70	80
Lb coal per sq ft steam radiator	0.0216	0.0172	0.0143	0.0123	0.0108
Lb coal per sq ft hot water radiator	0.0135	0.0108	0.0091	0.0078	0.0068
Lb coal per 1000 Btuh heat loss	0.0889	0.0717	0.0592	0.0507	0.0444

[a] Based on a heating value of 12,000 Btu per pound.

[b] Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss. Eight percent added for operation without night reduction of temperature.

[c] Per degree day.

From Table 3, the correction factor is 0.875 for −10 F outdoor design temperature, hence, $0.875 \times 0.087 = 0.076$. By Equation 4,

$$F = 0.076 \times 1000 \times 6310 = 480{,}000 \text{ cu ft.}$$

Estimating Oil Consumption

Table 5 gives unit fuel consumption factors for oil, similar to those given for gas in Table 4.

The factors in Table 5 apply only to an indoor design temperature of 70 F and an outdoor design temperature of 0 F. For other outdoor design temperatures, the constants in Table 5 must be multiplied by the values in Table 3 as explained under Estimating Gas Consumption.

Values given in Table 5 are based on use of oil with a heating value of *141,000 Btu per gallon*. For other heating values, multiply the values in Table 5 by the ratio of 141,000 divided by the heating value per gallon of fuel being used.

Example 5: Estimate the seasonal oil consumption of a boiler, designed for oil firing, in a building located in Toledo, Ohio. The building has a calculated heat loss of 240,000 Btu per hr. The oil heat value is 144,000 Btu per gal, and the assumed seasonal efficiency is 80 percent. The outdoor design temperature for Toledo is −10 F, and the indoor design temperature is 70 F.

Solution: From Table 5, under 80 percent efficiency and in the bottom line, the value of U is found to be 0.00383 gal per 1000 Btu hourly heat loss for 0 F outdoor temperature. The correction factor for −10 F outdoor design temperature from Table 3 is 0.875 which makes the correction $0.875 \times 0.00383 = 0.00335$. Making a further correction for the heating value, $0.00335 \times (141{,}000/144{,}000) = 0.00328$ gal per 1000 Btu per hr calculated heat loss per degree day.

From Table 2, *the total normal degree days for Toledo are* 6394. Since U is expressed in 1000 Btu, N is equal to 240. Substituting in Equation 4

$$F = 0.00328 \times 6394 \times 240 = 5030 \text{ gal.}$$

Estimating Coal or Coke Consumption

Coal or coke consumption estimates are made by following *exactly the same procedure as for oil*. Values of U are given in Table 6 which only apply to an indoor design temperature of 70 F and an outdoor design temperature of 0 F. A correction must be made for other conditions by use of the multiplying factors in Table 3. Data in Table 6 are based on 12,000 Btu per lb coal, and for other heating values of coal they must be multiplied by the ratio of 12,000 divided by the heating value of fuel used.

Example 6: A building in Scranton, Pa. has a calculated heat loss of 240,000 Btu per hr based on an indoor design temperature of 70 F and an outdoor design temperature of −10 F. What will be the estimated normal seasonal anthracite or coke consumption for heating if 13,000 Btu per lb fuel is burned in a hand-fired boiler, with automatic control, at a seasonal efficiency of 80 percent, and what part of the total will be used during November, December, and January?

Solution: From Table 6, U is 0.0444 of coal or coke per 1000 Btu per hr heat loss per degree day. Correcting for the outdoor design temperature of −10 F from Table 3, the value of U is $0.875 \times 0.0444 = 0.0389$. From Table 2, D is 6218 and from the problem, N is 240.

Substituting in Equation 4,

$$F = 0.0389 \times 240 \times 6047 = 56{,}500 \text{ lb.}$$

Fuel used over any period is, according to the theory of the degree day, proportional to the number of degree days during the period. From Table 2, the average numbers of degree days for November, December, and January in Scranton are 693, 1057, and 1141 respectively, a total of 2891. The yearly total is 6047, so that during these three months the estimated consumption is

$$\frac{2891}{6047} \times 56{,}500 = 27{,}000 \text{ lb.}$$

Estimating Steam Consumption

In estimating steam consumption the efficiency is generally assumed at 100 percent. If for low-pressure steam an average heating value of 1000 Btu per pound of steam is used, no correction is necessary. In comparing values from different cities, correction should be made for design temperature (see Table 3) when the unit figures are in terms of heat loss but not when the values are in terms of building volume or floor space.

Where the heat loss is calculated in Btu per (hour) (*degree difference in temperature*) the simple Equation 7 may be used:

$$F = \frac{H \times 24 \times D}{1000} \tag{7}$$

F = pounds of steam required for estimate period.
H = calculated heat loss, Btu per (hour) (degree difference).
24 = hours in one day.
D = number of degree days for the period of estimation.
1000 = Btu delivered per pound of steam condensed.

In this method the number of degree days automatically takes care of average indoor and outdoor temperature difference. *When degree days are taken from* Table 2, an av-

Table 7 .... Steam Consumption of Buildings with Various Types of Occupancy[a]

Type of Building	No. Bldgs	Average Volume Heated Space 1000 Cu Ft	Steam for Heating Lb per DD per 1000 Cu Ft	Average Hours of Occupancy
Office	334	2160	0.685	12.1
Office and Bank	49	3000	0.577	13.1
Office and Printing	8	1895	1.230	17.7
Office and Theater	7	4950	0.412	12.9
Office and Stores or Shops	26	1615	0.617	13.2
Bank	16	806	0.786	11.7
Department Store	63	3400	0.385	11.1
Stores	73	310	0.624	10.4
Loft	63	865	0.588	10.0
Warehouse	24	2230	0.459	9.4
Hotel and Club	73	1795	0.990	22.3
Apartment or Residence	51	1425	0.962	21.8
Theater	22	1240	0.482	12.9
Garage	13	1540	0.202	21.4
Manufacturing	19	1350	0.808	9.5
Church	9	656	0.532	7.9
Hospital	4	3306	1.194	22.0
School	8	1115	0.592	11.5
Municipal or Federal	15	3215	0.587	15.6
Lodge, Gym, Hall, or Auditorium	12	880	0.390	12.4
Miscellaneous	7	1387	0.479	21.4

[a] *Principles of Economical Heating*, National Association of Building Owners and Managers.

Table 8 .... Degree Days for Industrial Heating in Various Cities of the United States*

Annual Totals for a Normal Heating Season

State	City	Degree Days 45F Base	Degree Days 55F Base
Ala.	Birmingham	6	853
	Mobile	....	245
	Montgomery	....	504
Ariz.	Phoenix	....	229
	Yuma	....	37
Ark.	Fort Smith	329	1478
	Little Rock	276	1188
Cal.	Eureka	....	1328
	Fresno	3	718
	Independence	519	1930
	Los Angeles	....	17
	Red Bluff	24	822
	Sacramento	3	803
	San Diego	....	26
	San Francisco	....	384
	San Luis Obispo	....	230
Col.	Denver	1548	3440
	Grand Junction	1757	3433
	Pueblo	1499	3261
Conn.	Meriden	....	734
	New Haven	1769	3237
D.C.	Washington	1041	2487
Fla.	Pensacola	....	127
Ga.	Atlanta	109	1165
	Augusta	....	661
	Macon	....	711
	Savannah	....	231
Idaho	Boise	1045	2814
	Lewiston	1034	2688
	Pocatello	2161	4140
Ill.	Cairo	749	2119
	Chicago	1969	3743
	Springfield	1677	3289
Ind.	Evansville	799	2335
	Indianapolis	1397	2829
Ia.	Charles City	3370	5293
	Davenport	2296	4142
	Des Moines	2440	4180
	Dubuque	1628	4468
	Keokuk	1926	3579
	Sioux City	2894	4732
Kan.	Concordia	1652	2690
	Dodge City	1385	2962
	Topeka	1518	1811
	Wichita	1152	2587
Ky.	Lexington	....	2557
	Louisville	1073	2294
La.	New Orleans	....	30
	Shreveport	....	565
Me.	Eastport	2956	5236
	Portland	2530	4572
Md.	Baltimore	986	2491
Mass.	Boston	1787	3603
	Nantucket	1514	3419
Mich.	Alpena	3131	5499
	Detroit	2240	4089
	Escanaba	3699	5918
	Grand Haven	2405	3435
	Grand Rapids	2332	4177
	Houghton	4229	6112
	Lansing	2537	4444
	Marquette	3713	5842
	Port Huron	2630	4275
	Saginaw	2312	4552
	Sault Ste. Marie	4049	6575
Minn.	Duluth	4419	6774
	Minneapolis	3309	5417
	Moorhead	4796	6572
	St. Paul	2497	5497
Miss.	Vicksburg	....	468
Mo.	Columbia	1252	2939
	Hannibal	1635	3231
	Kansas City	1463	2980
	St. Louis	1186	2745
	Springfield	982	2423
Mont.	Havre	3736	5874
	Helena	2843	5071
	Kalispell	2874	5131
Neb.	Lincoln	3023	3850
	North Platte	2291	4152
	Omaha	2284	3982
	Valentine	2833	4801
Nev.	Winnemucca	1670	3468
N. H.	Concord	2646	4640
N. J.	Atlantic City	1123	2904
N. M.	Sante Fe	1634	3106
N. Y.	Albany	2018	4302
	Binghamton	2073	4296
	Buffalo	2359	4316
	Ithaca	2412	4923
	New York	1098	3089
	Oswego	2274	4363
	Rochester	2341	4231
N. C.	Charlotte	172	1388
	Hatteras	....	718
	Raleigh	229	1080
	Wilmington	....	729
N. D.	Bismarck	3831	6468
	Williston	4616	6399
Ohio	Cincinnati	1376	3003
	Cleveland	1525	3795
	Columbus	1600	3255
	Dayton	1487	3147
	Sandusky	1949	3425
	Toledo	1990	3757
Okla.	Oklahoma City	600	1835
Ore.	Baker	2321	4307
	Portland	373	1911
	Roseburg	272	1868
Pa.	Erie	2337	3837
	Harrisburg	1565	3236
	Philadelphia	1122	2695
	Scranton	1938	3755
	Pittsburgh	1377	3028
R. I.	Block Island	871	3388
S. C.	Charleston	....	336
	Columbia	....	759
	Greenville	297	1502
S.D.	Huron	3743	5678
	Pierre	3162	5234
	Rapid City	2590	4628
	Yankton	2898	6045
Tenn.	Chattanooga	242	1398
	Knoxville	431	1741
	Memphis	166	1284
	Nashville	419	1678
Tex.	Abilene	....	915
	Amarillo	786	2220
	El Paso	32	919
	Fort Worth	....	754
	Galveston	....	43
	Houston	....	110
	Palestine	....	484
	San Antonio	....	305
Utah	Modena	1978	3981
	Salt Lake City	1475	3202
Vt.	Burlington	3014	4984
	Northfield	3652	7121
Va.	Lynchburg	554	1928
	Norfolk	260	1496
	Richmond	549	1895
Wash.	North Head	184	2062
	Seattle	408	2185
	Spokane	1741	3672
	Tacoma	507	2365
	Walla Walla	993	2565
W. Va.	Elkins	1506	3327
	Parkersburg	1147	2784
Wis.	Green Bay	3318	5331
	La Crosse	3034	3992
	Madison	3067	4850
	Milwaukee	2657	4617
Wyo.	Cheyenne	2500	4700
	Lander	3208	5450

* From *Degree-Day Handbook*, by C. Strock and C. H. B. Hotchkiss (The Industrial Press, 1937, pp. 123-134). Values are not coordinated with Table 2 which is from a later source.

erage indoor temperature of approximately 70 F is assumed throughout the period. If an average indoor temperature other than approximately 70 F is to be used, the number of degree days should be obtained for the new base.

Example 7: An eight-story building in Pittsburgh is operated with a daytime temperature of 70 F. The calculated heat loss is 10,500 Btu per (hr) (*degree temperature difference*). What is the estimated average yearly steam consumption for building heating?

Solution: Since the average indoor temperature is approximately 70 F, the degree days from Table 2, based on 70 F may be used. Therefore, from Table 2, Pittsburgh has 5048 degree days per normal season. Inserting in Equation 5:

$$F = \frac{10{,}500 \times 24 \times 5048}{1000} = 1{,}272{,}000 \text{ lb of steam.}$$

Consideration has been given to the difference in steam utilization of different types of buildings, and Table 7 shows actual average units for these various types. These figures were obtained from operating results in 896 buildings located in all sections of the United States. Being averages, and for small groups in each type, the figures may need considerable modification to allow for local variations. It should be especially noted that the steam used for heating water for service is not included in the values given in Table 7.

Additional figures are to be found in a paper presented before the Society.[18]

Example 8: A store in Philadelphia with a heating system designed to maintain 70 F indoors in 0 F weather has 250,000 cu ft of heated space. What would be the estimated average yearly steam consumption of purchased steam for heating?

Solution: According to Table 7, a store would use 0.624 lb of steam per degree day per 1000 cu ft heated space. From Table 2, Philadelphia has 4523 degree days per normal year. Inserting in Equation 4:

$$F = 0.624 \times 250 \times 4523 = 706{,}000 \text{ lb of steam.}$$

Degree Day as an Operating Unit

The degree day is also widely used as a means of comparing the efficiency of the fuel consumption of one period with another *for the same building*. Since the fuel consumption is approximately proportional to the weather (degree days), and since the periods to be compared may not have the same weather conditions, the comparison can be made only after the fuel consumptions have been computed on a comparable weather basis, that is, upon the actual number of degree days occurring for a given month and year in the city under consideration. Since fuel consumption is proportional to the number of degree days, plant operators frequently compute each month the fuel burned per degree day by the heating plant. The resulting unit value, by eliminating the outdoor temperature variable, indicates whether the operating efficiency of the plant is above or below the previous month or year.

INDUSTRIAL DEGREE DAYS

When estimating the heating requirements of warehouses, factories, and other buildings where the indoor temperature to be maintained differs greatly from the usual indoor temperature range of 68 to 72 F, it is common to use degree days based on temperatures of 55 to 45 F. These are generally referred to as industrial degree days, and are listed in Table 8 for a number of cities using both 55 F and 45 F as the base. In general, the degree days (55 F base) are used when the temperature of the heated space is to be maintained at approximately 60 F. For an inside temperature of about 50 F, the degree days calculated on a 45 F base should be used.

The industrial degree days are useful for estimating fuel consumption by comparison between localities in which the difference in degree days is not too great, and where the known fuel consumption of one building may be used to determine unit consumption per degree day.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures ar shown for a number of buildings in Detroit in Table 9.[18]

Table 9 .... Building Load Factors and Demands of Some Detroit Buildings

Building Classification	Load Factor	Lb of Demand per (Hour) (Sq Ft EDR)
Clubs and Lodges	0.318	0.184
Hotels	0.316	0.207
Printing	0.287	0.217
Offices	0.263	0.209
Apartments	0.255	0.225
Retail Stores	0.238	0.182
Auto Sales and Service	0.223	0.248
Banks	0.203	0.158
Churches	0.158	0.152
Department Stores	0.138	0.145
Theaters	0.126	0.151

These maximum demands were measured by an attachment on the condensate meter, and therefore represent the amounts of condensate passed through the meter in the highest hours, rather than the true rate at which steam is supplied. There might be slight differences in these two quantities due to time lag and to storage of condensate in the system, but wherever this has been investigated it has been found to be negligible.

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization. Thus, in Table 9, the theaters, operating for short hours, have a load factor of 0.126 as compared with the figure of 0.318 for clubs and lodges.

REFERENCES

[1] W. S. Harris and R. J. Martin: Heat transmitted to the I=B=R research home from the chimney (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 97).

[2] S. Konzo: *The Stoker-fired Warm-air Furnace in the Research Residence* (University of Illinois, *Engineering Experiment Station Circular* 39, 1939, p. 98).

[3] *Performance of a Hot Water Heating System in the I=B=R Research Home at the University of Illinois* (University of Illinois, *Engineering Experiment Station Bulletin* No. 349, January 4, 1944).

[4] W. G. Colborne: Performance of intermittently-fired oil furnaces (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 427).

[8] *Comfort Heating* (American Gas Association, 1938).

[9] W. M. Myler, Jr. and H. W. Nelson: Are automatic air shutters justified on a gas-fired conversion burner? (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 111).

[10] J. W. Tieman and F. L. Bagby: Efficiency of bituminous-coal-burning space heater (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 89).

[11] R. A. Sherman and R. C. Cross: Heat losses and efficiencies of fuels in residential heating (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 185).

[12] *Fuels and Burners* (University of Illinois, *Small Homes Council Circular* G3.5, July 1949).

[13] A. P. Krats and S. Konso: *Investigation of Oil-fired Forced-air Furnace Systems in the Research Residence* (University of Illinois, *Engineering Experiment Station Bulletin* No. 318, November 7, 1939).

[14] Paul D. Close: Graphical method of calculating heat losses (ASHVE TRANSACTIONS, Vol. 49, 1943, p. 345).

[15] *House Heating* (American Gas Association, Industrial Gas Series, 3rd ed.).

[16] *Report* of commercial relations committee (*National District Heating Association Proceedings*, 1932).

[17] Rutcher Skagerberg and J. E. Phifer: Fuel consumption analysis for multi-family housing projects (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 113).

[18] C. W. Signor: Heat requirements of buildings (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 517).

[19] J. H. Walker and G. H. Tuttle: The heat requirements of buildings (ASHVE TRANSACTIONS, Vol. 41, 1935, p. 171).

CHAPTER 38

REFRIGERATION

Refrigeration Theory: Definitions and Basic Concepts, Refrigerants, Vapor Compression Refrigeration Cycles, Suction and Discharge Pressure Effect, Complex Refrigeration Cycles, Air Cycle, Steam Jet, Absorption System; Basic Refrigeration Equipment: Compression Machines and Controls, Condensers, Evaporators and Coolers, Refrigeration Control, Piping and Accessories, Equipment Characteristics and Selection

WITH the increasing use of all-year comfort air-conditioning installations, the importance of refrigeration to the air-conditioning engineer has been greatly magnified. The details of equipment operation, maintenance, and design remain problems for the refrigeration engineer, but the air-conditioning engineer does retain a responsibility to the customer which requires on his part some knowledge of the different refrigeration cycles and the relative merits of each. In order to assist in meeting this need, the present chapter has been divided into four parts, the first covering the fundamental technical relationships which govern the selection and analysis of an operating cycle, the next two presenting brief discussions of basic refrigerating equipment and auxiliaries, and the last, information on selection criteria.

REFRIGERATION THEORY

Definitions and Basic Concepts

The *ton of refrigeration* is a quantity unit which originated in the days when harvested ice was the principal source of summer cooling. By definition the *ton* is the cooling effect realized when one ton of 32 F ice melts to water at 32 F. Since the latent heat of fusion of ice is 144 Btu per pound, the *ton* represents a unit cooling effect of $144 \times 2{,}000 = 288{,}000$ Btu. In common practice the ton is usually considered a rate (rather than quantity) unit, and is taken as 288,000 Btu per day (24 hours), or 12,000 Btu per hour, or 200 Btu per minute. Thus for air-conditioning calculations, the size of the requisite refrigeration machine, expressed in tons, can be obtained by dividing the heat gain of the structure, expressed in Btu per hour, by 12,000. In equation form:

$$H_t = (\text{Btu per hour heat gain}) \div 12{,}000 \qquad (1)$$

where

H_t = load in tons.

The *working substance*, or *refrigerant*, is the fluid which carries heat through the refrigeration cycle from the evaporator, where heat enters the refrigerant, to the condenser where the heat is discharged to some cooling medium. The great majority of modern refrigeration systems use a liquefiable vapor as the working substance. By altering the pressure of the refrigerant its boiling temperature is changed, allowing the material to *boil* in the evaporator at a temperature sufficiently lower than that of the conditioned space, to insure maintenance of an effective heat transfer rate from the space (or in some cases from a secondary cooling fluid such as brine or cold water) to the refrigerant. The vapor formed in the evaporator is then raised in pressure (by a compressor, or by the absorber-generator combination of the absorption system) until its new boiling temperature exceeds the temperature of the available cooling medium. Under these conditions, heat transfer is established from the refrigerant vapor to the cooling medium with resultant condensation of the refrigerant. When condensed, the high-pressure liquid refrigerant is reduced in pressure and again allowed to *boil* in the evaporator.

In order to permit evaluation of the effectiveness with which any given cycle operates, some term is desirable that would be comparable to the term *efficiency* used for heat engines. In refrigeration the desired effect is heat extraction, and the cost of achieving this extraction is the amount of energy that must be supplied as shaft work. Thus the ratio of refrigerating effect to the heat equivalent of the compressor work is used as a measure of effectiveness, and is defined as the coefficient of performance.

If the desired effect is the rejection of heat through the condenser instead of heat extraction through the evaporator, the refrigeration system is then termed a heat pump. In this case the coefficient of performance is the ratio of the heat rejected from the condenser to the heat equivalent of the compressor work. The coefficient of performance for the heat pump is greater than that for a system operating as a refrigerating machine, because all mechanical shaft work required to operate the compressor is dissipated as useful heat through the condenser.

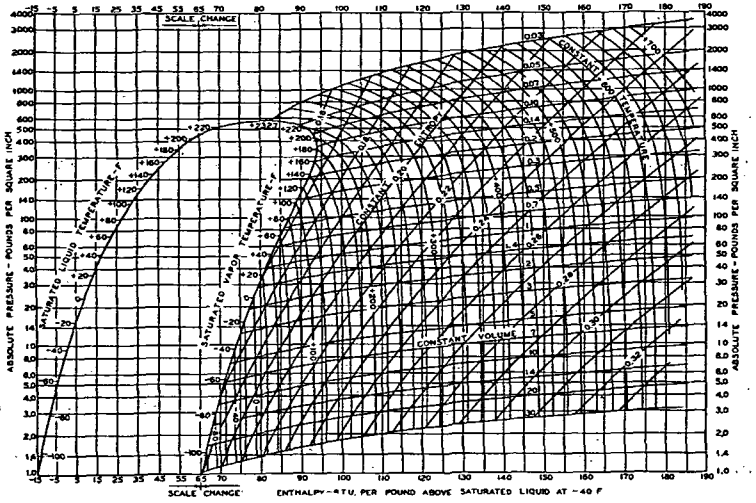
The Carnot cycle, an ideal, thermodynamically reversible cycle consisting of an adiabatic expansion and an isothermal expansion, followed by an adiabatic compression and an isothermal compression to form a closed cycle, may be shown to be a measure of the maximum possible conversion of heat energy into mechanical energy. In its reversed form it is a measure of the maximum performance possible for any refrigeration cycle operating either as a refrigerator or as a heat pump. Although it cannot be applied in an actual machine because of the impossibility of obtaining complete reversibility, it is, nevertheless, extremely valuable as a criterion of inherent limitations. The coefficient of performance (CP) of a reversed Carnot cycle system operating as a refrigeration system is

$$(\text{CP}) = \frac{T_e}{T_c - T_e} \qquad (2)$$

where

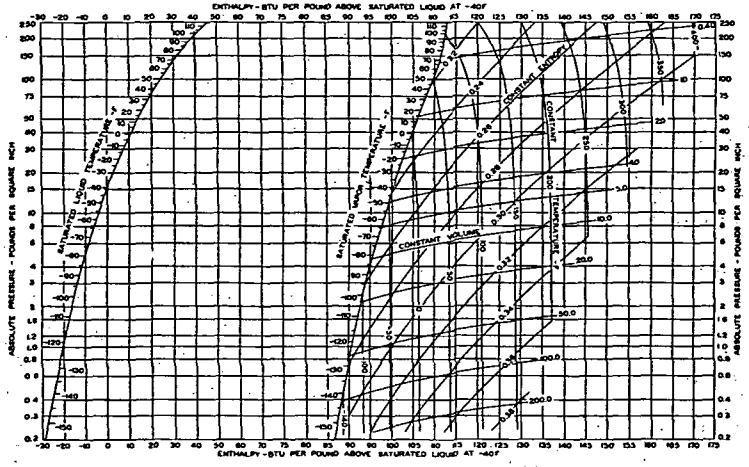
T_e = evaporator temperature, Fahrenheit, absolute.

T_c = condenser temperature, Fahrenheit, absolute.



Temperature—F; Volume—Cu Ft per Lb; Entropy—Btu per (Lb) (F deg).

Fig. 1 .... Pressure-Enthalpy Diagram for Dichlorodifluoromethane (CCl_2F_2) (Refrigerant 12)



Temperature—F; Volume—Cu Ft per Lb; Entropy—Btu per (Lb) (F deg).

Fig. 2 .... Pressure-Enthalpy Diagram for Monochlorodifluoromethane ($CHClF_2$) (Refrigerant 22)

With the ideal Carnot cycle operating as a heat pump, the coefficient of performance is

$$(CP) = \frac{T_c}{T_c - T_e} \tag{3}$$

The Carnot cycle coefficient of performance for both a refrigerating machine and a heat pump increases as the spread between the evaporator and the condenser temperatures decreases. In general, the same is true for an actual system operating as either a refrigerating machine or a heat pump.

Refrigerants

A desirable refrigerant should possess chemical, physical, and thermodynamic properties that permit its efficient application in refrigerating systems. In addition, when the volume of the charge is large, there should be little or no danger to health or to property in case of its escape.

Thermodynamically, a material for use as a refrigerant should have a large latent heat of vaporization since it is this heat quantity—subject to minor variations—that constitutes the working effectiveness of the refrigerant. Further, since the work required to compress a vapor increases rapidly with the pressure ratio, the thermodynamic characteristics of the fluid should be such that the required low-to-high temperature range can be achieved with only a moderate change in ratio. A further consideration, from the standpoint of practical operating effectiveness, is that the suction pressure should not be below atmospheric (to prevent leakage of air into the refrigerant lines) nor should the condenser pressure be excessively high (to prevent need for extra-heavy construction). The specific volume-specific enthalpy relationship is also important because some materials would have such low density, when in vapor form, that impractical compressor displacements would be needed to handle the suction vapor.

Properties of refrigerants are usually given either in tabular or graphic form. In contrast to the temperature-entropy plotting which is used almost exclusively in steam-power work, refrigeration problems are usually referred to a pressure-enthalpy chart. The advantage of pressure-enthalpy plotting in that linear distances on the chart correspond to energy gains or losses, and the two types of processes, constant-pressure and constant-enthalpy, that occur most frequently in refrigeration cycles, can both be represented by straight vertical or horizontal lines. Figs. 1 and 2 present pressure-enthalpy charts for dichlorodifluoromethane, CCl_2F_2, and monochlorodifluoromethane, $CHClF_2$, respectively. Although tabular arrangements of refrigerant properties require interpolation between values, they have the advantage of an accuracy greater than that obtainable from a chart. Tables 1, 2, 3, and 4 give the thermodynamic properties of four of the more common refrigerants used in air-conditioning installations: Refrigerant* 12, dichlorodifluoromethane; Refrigerant 22, monochlorodifluoromethane; Refrigerant 11, trichloromonofluoromethane; and Refrigerant 113, trichlorotrifluoroethane. The first two of these are commonly used in reciprocating compressors while the last two are commonly used in centrifugal machines.

Referring to Table 1, the first column gives the range of saturation temperatures likely to occur in practice. The second column gives the saturation pressure expressed in pounds per square inch absolute corresponding to a given temperature, while the next six columns give the three fundamental specific properties, volume, enthalpy, and entropy, of the saturated liquid and saturated vapor, respectively. The last four columns give values of enthalpy and entropy for gases with 25 deg and with 50 deg of superheat; note particularly that the column heading *50 F superheat* means, not that the gas is at a temperature of 50 F, but that its temperature exceeds by 50 deg the saturation temperature corresponding to its actual pressure. Thus, CCl_2F_2 vapor at 38.0 psig and 91 F possesses 50 deg of superheat, since its saturation temperature corresponding to 52.7 psia is 41 F.

The tabular arrangements of refrigerant properties are literally for saturated or superheated materials only. In many cases, however, the engineer must work with subcooled liquids. With an accuracy sufficient for all practical purposes, the specific volume and the enthalpy of any subcooled refrigerant can be taken as equal to the values read from the tables for a saturated liquid at the same temperature. Thus, if CCl_2F_2 at 121 psia and 40 F is passing through a pipe, its volume and enthalpy can be determined from Table 1 as 0.0116 cu ft per pound and 17.0 Btu per pound.

Frequently it is necessary to determine the properties of a *wet* vapor or of a mixture of liquid with some added vapor, such as is found at the discharge from an expansion valve. This can be done from the tables by noting that the specific enthalpy of the mixture must be equal to that of the saturated liquid, plus a fraction of the latent heat of vaporization equal to the fraction of refrigerant that is present in vapor form. Consider, for example, CCl_2F_2 with a quality (the percent in vapor form) of 30 percent; the enthalpy of this material would be equal to

$$h_m = h_f + 0.30\ (h_v - h_f) \tag{4}$$

where

h_m = specific enthalpy of the mixture.
h_f = specific enthalpy of the liquid.
h_v = specific enthalpy of the saturated vapor.

Values of h_f and h_v for use in Equation 4 can be obtained directly or by calculation from the tables of properties of refrigerants.

By a reversal of this same procedure the tabular data can be used to determine the state of a mixture leaving an expansion valve. Consider a valve to which saturated liquid at pressure p_s is admitted, and a mixture of saturated liquid and vapor at pressure p_d is discharged. The quality of the material at discharge is then determined by making use of the fact that the expansion process is completely irreversible, is a throttling process, and hence, occurs without change in enthalpy. Thus, the enthalpy of the mixture, h_m, is equal to the enthalpy of the saturated liquid at the entrance state, h_{fs}, and can therefore be read from the table. Thus,

$$h_{fs} = h_m = h_{vd} - (1 - x)\ (h_{vd} - h_{fd}) \tag{5}$$

or,

$$x = (h_m - h_{fd}) \div (h_{vd} - h_{fd}) \tag{6}$$

where

h_{fs} = enthalpy of saturated liquid at entrance to expansion valve.
h_m = enthalpy of mixture.
h_{vd} = enthalpy of saturated vapor at discharge.
h_{fd} = enthalpy of liquid at discharge.
x = proportion of liquid in the mixture, decimal.

Vapor Compression Refrigeration Cycle

Simple Cycle. The refrigerant *cycle* is the series of state changes (which occur in the conditioning processes) needed

* ASRE designations for these refrigerants.

Table 1 .... Properties of Dichlorodifluoromethane (CCl_2F_2)[a]

Sat. Temp. F	Abs. Press. lb per Sq in.	Volume		Enthalpy and Entropy Taken From −40 F							
				Enthalpy		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	23.87	0.0110	1.637	8.25	78.21	0.01869	0.17091	81.71	0.17829	85.26	0.18547
2	24.89	0.0110	1.574	8.67	78.44	0.01961	0.17075	81.94	0.17812	85.51	0.18529
4	25.96	0.0111	1.514	9.10	78.67	0.02052	0.17060	82.17	0.17795	85.76	0.18511
5	26.51	0.0111	1.485	9.32	78.79	0.02097	0.17052	82.29	0.17786	85.89	0.18502
6	27.05	0.0111	1.457	9.53	78.90	0.02143	0.17045	82.41	0.17778	86.01	0.18494
8	28.18	0.0111	1.403	9.96	79.13	0.02235	0.17030	82.66	0.17763	86.26	0.18477
10	29.35	0.0112	1.351	10.39	79.36	0.02328	0.17015	82.90	0.17747	86.51	0.18460
12	30.56	0.0112	1.301	10.82	79.59	0.02419	0.17001	83.14	0.17733	86.76	0.18444
14	31.80	0.0112	1.253	11.26	79.82	0.02510	0.16987	83.38	0.17720	87.01	0.18429
16	33.08	0.0112	1.207	11.70	80.05	0.02601	0.16974	83.61	0.17706	87.26	0.18413
18	34.40	0.0113	1.163	12.12	80.27	0.02692	0.16961	83.85	0.17693	87.51	0.18397
20	35.75	0.0113	1.121	12.55	80.49	0.02783	0.16949	84.09	0.17679	87.76	0.18382
22	37.15	0.0113	1.081	13.00	80.72	0.02873	0.16938	84.32	0.17666	88.00	0.18369
24	38.58	0.0113	1.043	13.44	80.95	0.02963	0.16926	84.55	0.17652	88.24	0.18355
26	40.07	0.0114	1.007	13.88	81.17	0.03053	0.16913	84.79	0.17639	88.49	0.18342
28	41.59	0.0114	0.973	14.32	81.39	0.03143	0.16900	85.02	0.17625	88.73	0.18328
30	43.16	0.0115	0.939	14.76	81.61	0.03233	0.16887	85.25	0.17612	88.97	0.18315
32	44.77	0.0115	0.908	15.21	81.83	0.03323	0.16876	85.48	0.17600	89.21	0.18303
34	46.42	0.0115	0.877	15.65	82.05	0.03413	0.16865	85.71	0.17589	89.45	0.18291
36	48.13	0.0116	0.848	16.10	82.27	0.03502	0.16854	85.95	0.17577	89.68	0.18280
38	49.88	0.0116	0.819	16.55	82.49	0.03591	0.16843	86.18	0.17566	89.92	0.18268
39	50.78	0.0116	0.806	16.77	82.60	0.03635	0.16838	86.29	0.17560	90.04	0.18262
40	51.68	0.0116	0.792	17.00	82.71	0.03680	0.16833	86.41	0.17554	90.16	0.18256
41	52.70	0.0116	0.779	17.23	82.82	0.03725	0.16828	86.52	0.17549	90.28	0.18251
42	53.51	0.0116	0.767	17.46	82.93	0.03770	0.16823	86.64	0.17544	90.40	0.18245
44	55.40	0.0117	0.742	17.91	83.15	0.03859	0.16813	86.86	0.17534	90.65	0.18235
46	57.35	0.0117	0.718	18.36	83.36	0.03948	0.16803	87.09	0.17525	90.89	0.18224
48	59.35	0.0117	0.695	18.82	83.57	0.04037	0.16794	87.31	0.17515	91.14	0.18214
50	61.39	0.0118	0.673	19.27	83.78	0.04126	0.16785	87.54	0.17505	91.38	0.18203
52	63.49	0.0118	0.652	19.72	83.99	0.04215	0.16776	87.76	0.17496	91.61	0.18193
54	65.63	0.0118	0.632	20.18	84.20	0.04304	0.16767	87.98	0.17486	91.83	0.18184
56	67.84	0.0119	0.612	20.64	84.41	0.04392	0.16758	88.20	0.17477	92.06	0.18174
58	70.10	0.0119	0.593	21.11	84.62	0.04480	0.16749	88.42	0.17467	92.28	0.18165
60	72.41	0.0119	0.575	21.57	84.82	0.04568	0.16741	88.64	0.17458	92.51	0.18155
62	74.77	0.0120	0.557	22.03	85.02	0.04657	0.16733	88.86	0.17450	92.74	0.18147
64	77.20	0.0120	0.540	22.49	85.22	0.04745	0.16725	89.07	0.17442	92.97	0.18139
66	79.67	0.0120	0.524	22.95	85.42	0.04833	0.16717	89.29	0.17433	93.20	0.18130
68	82.24	0.0121	0.508	23.42	85.62	0.04921	0.16709	89.50	0.17425	93.43	0.18122
70	84.82	0.0121	0.493	23.90	85.82	0.05009	0.16701	89.72	0.17417	93.66	0.18114
72	87.50	0.0121	0.479	24.37	86.02	0.05097	0.16693	89.93	0.17409	93.99	0.18106
74	90.20	0.0122	0.464	24.84	86.22	0.05185	0.16685	90.14	0.17402	94.12	0.18098
76	93.00	0.0122	0.451	25.32	86.42	0.05272	0.16677	90.36	0.17394	94.34	0.18091
78	95.85	0.0123	0.438	25.80	86.61	0.05359	0.16669	90.57	0.17387	94.57	0.18083
80	98.76	0.0123	0.425	26.28	86.80	0.05446	0.16662	90.78	0.17379	94.80	0.18075
82	101.70	0.0123	0.413	26.76	86.99	0.05534	0.16655	90.98	0.17372	95.01	0.18068
84	104.8	0.0124	0.401	27.24	87.18	0.05621	0.16648	91.18	0.17365	95.22	0.18061
86	107.9	0.0124	0.389	27.72	87.37	0.05708	0.16640	91.37	0.17358	95.44	0.18054
88	111.1	0.0124	0.378	28.21	87.56	0.05795	0.16632	91.57	0.17351	95.65	0.18047
90	114.3	0.0125	0.368	28.70	87.74	0.05882	0.16624	91.77	0.17344	95.86	0.18040
92	117.7	0.0125	0.357	29.19	87.92	0.05969	0.16616	91.97	0.17337	96.07	0.18033

Table 1 .... Properties of Dichlorodifluoromethane (CCl_2F_2)[a] (Concluded)

Sat. Temp. F	Abs. Press. lb per Sq in.	Volume		Enthalpy and Entropy Taken From −40 F							
				Enthalpy		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
94	121.0	0.0126	0.347	29.68	88.10	0.06056	0.16608	92.16	0.17330	96.28	0.18026
96	124.5	0.0126	0.338	30.18	88.28	0.06143	0.16600	92.36	0.17322	96.50	0.18018
98	128.0	0.0126	0.328	30.67	88.45	0.06230	0.16592	92.55	0.17315	96.71	0.18011
100	131.6	0.0127	0.319	31.16	88.62	0.06316	0.16584	92.75	0.17308	96.92	0.18004
102	135.3	0.0127	0.310	31.65	88.79	0.06403	0.16576	92.93	0.17301	97.12	0.17998
104	139.0	0.0128	0.302	32.15	88.95	0.06490	0.16568	93.11	0.17294	97.32	0.17993
106	142.8	0.0128	0.293	32.65	89.11	0.06577	0.16560	93.30	0.17288	97.53	0.17987
108	146.8	0.0129	0.285	33.15	89.27	0.06663	0.16551	93.48	0.17281	97.73	0.17982
110	150.7	0.0129	0.277	33.65	89.43	0.06749	0.16542	93.66	0.17274	97.93	0.17976
112	154.8	0.0130	0.269	34.15	89.58	0.06836	0.16533	93.82	0.17266	98.11	0.17969
114	158.9	0.0130	0.262	34.65	89.73	0.06922	0.16524	93.98	0.17258	98.29	0.17961
116	163.1	0.0131	0.254	35.15	89.87	0.07008	0.16515	94.15	0.17249	98.48	0.17954
118	167.4	0.0131	0.247	35.65	90.01	0.07094	0.16505	94.31	0.17241	98.66	0.17946
120	171.8	0.0132	0.240	36.16	90.15	0.07180	0.16495	94.47	0.17233	98.84	0.17939
122	176.2	0.0132	0.233	36.66	90.28	0.07266	0.16484	94.63	0.17224	99.01	0.17931
124	180.8	0.0133	0.227	37.16	90.40	0.07352	0.16473	94.78	0.17215	99.18	0.17922
126	185.4	0.0133	0.220	37.67	90.52	0.07437	0.16462	94.94	0.17206	99.35	0.17914
128	190.1	0.0134	0.214	38.18	90.64	0.07522	0.16450	95.09	0.17196	99.53	0.17906
130	194.9	0.0134	0.208	38.69	90.76	0.07607	0.16438	95.25	0.17186	99.70	0.17897
132	199.8	0.0135	0.202	39.19	90.86	0.07691	0.16425	95.41	0.17176	99.87	0.17889
134	204.8	0.0135	0.196	39.70	90.96	0.07775	0.16411	95.56	0.17166	100.04	0.17881
136	209.9	0.0136	0.191	40.21	91.06	0.07858	0.16396	95.72	0.17156	100.22	0.17873
138	215.0	0.0137	0.185	40.72	91.15	0.07941	0.16380	95.87	0.17145	100.39	0.17864
140	220.2	0.0138	0.180	41.24	91.24	0.08024	0.16363	96.03	0.17134	100.56	0.17856

[a] ASRE designation—Refrigerant 12.

to restore the refrigerant to a condition in which it will possess the ability to extract heat from the space to be cooled. For all compression-type systems the cycle consists of four processes: heat gain in the evaporator; pressure rise in the compressor; heat loss in the condenser; pressure loss in the expansion valve. The compression process is accomplished at the expense of energy added to the compressor in the form of shaft work, and the expansion process could be carried out, if the economics of the system would permit, in an expanding engine with consequent release of energy as shaft work. In ordinary systems, however, the additional first cost and maintenance cost of an expanding engine so greatly exceed the advantage resulting from the work realized, that such engines are not used, and instead, the pressure reduction is allowed to occur irreversibly in an expansion valve. Basically, then, a refrigeration cycle consists of two heat-transfer processes and two pressure-change processes, no work entering into the heat-transfer processes and—in the simple cycle—no heat transfer occurring during the pressure-change processes.

The most common and least complicated type of refrigeration cycle is called the simple saturation cycle, and is shown diagrammatically in Fig. 3 and plotted upon pressure-enthalpy coordinates in Fig. 4. For this system, saturated vapor flows without gain or loss of heat from the evaporator to the suction of the compressor. During passage through the compressor the energy added as shaft work goes entirely to increase the enthalpy of the refrigerant, and the compression process, which is assumed to occur reversibly and without external heat transfer, is characterized by constant entropy. Thus, the state of the superheated vapor leaving the compressor can be determined from the tables of thermodynamic properties by noting the discharge pressure and fixing, also, the entropy of the saturated vapor at entrance to the compressor.

Superheated vapor from the compressor flows to the condenser where de-superheating and condensation take place. From the condenser the refrigerant flows to the expansion valve, undergoes a constant-enthalpy pressure reduction, and returns to the evaporator where it again removes a quantity of undesired heat. When the evaporator is arranged to permit direct cooling of room air by the refrigerant, the system is said to be of the *direct-expansion* type, while a system in which the evaporating refrigerant cools water or brine, which in turn cools the air, is said to be *indirect*. Although many differences exist between most actual systems and that of the simple saturation cycle, this latter is, nonetheless, of great value in that it provides an extremely simple method of rapidly achieving an approximate analysis of probable power requirements, compressor size, etc. Further, the equations used in analysis of a simple saturation cycle form the basis of the more complex treatments required for compound refrigeration cycles. For these reasons a typical simple saturation problem will be worked in detail.

Example 1: A simple saturation cycle carries a 7-ton load when operating between suction and discharge pressure of

Table 2 .... Properties of Monochlorodifluoromethane ($CHClF_2$)[a]

Sat. Temp. F	Abs. Press. lb per Sq in.	Volume		Enthalpy and Entropy Taken from −40 F							
				Enthalpy		Entropy		50 Deg Superheat		100 Deg Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	38.79	0.01192	1.373	10.63	105.02	0.0240	0.2293	112.35	0.2446	120.00	0.2590
2	40.43	0.01195	1.320	11.17	105.24	0.0251	0.2289	112.59	0.2442	120.26	0.2586
4	42.14	0.01198	1.270	11.70	105.45	0.0262	0.2285	112.83	0.2438	120.52	0.2581
5	43.02	0.01200	1.246	11.97	105.56	0.0268	0.2283	112.95	0.2436	120.65	0.2579
6	43.91	0.01201	1.221	12.23	105.66	0.0274	0.2280	113.07	0.2434	120.78	0.2577
8	45.74	0.01205	1.175	12.76	105.87	0.0285	0.2276	113.31	0.2430	121.04	0.2572
10	47.63	0.01208	1.130	13.29	106.08	0.0296	0.2272	113.55	0.2426	121.30	0.2568
12	49.58	0.01211	1.088	13.82	106.29	0.0307	0.2268	113.79	0.2422	121.56	0.2564
14	51.59	0.01215	1.048	14.36	106.50	0.0319	0.2264	114.02	0.2418	121.82	0.2560
16	53.66	0.01218	1.009	14.90	106.71	0.0330	0.2260	114.25	0.2414	122.08	0.2556
18	55.79	0.01222	0.9721	15.44	106.92	0.0341	0.2257	114.48	0.2410	122.33	0.2552
20	57.98	0.01225	0.9369	15.98	107.13	0.0352	0.2253	114.71	0.2406	122.59	0.2548
22	60.23	0.01229	0.9032	16.52	107.33	0.0364	0.2249	114.94	0.2402	122.84	0.2544
24	62.55	0.01232	0.8707	17.06	107.53	0.0375	0.2246	115.17	0.2398	123.10	0.2540
26	64.94	0.01236	0.8398	17.61	107.73	0.0379	0.2242	115.40	0.2395	123.35	0.2537
28	67.40	0.01239	0.8100	18.17	107.93	0.0398	0.2239	115.62	0.2391	123.60	0.2533
30	69.93	0.01243	0.7816	18.74	108.13	0.0409	0.2235	115.84	0.2387	123.85	0.2529
32	72.53	0.01247	0.7543	19.32	108.33	0.0421	0.2232	116.07	0.2383	124.10	0.2525
34	75.21	0.01250	0.7283	19.90	108.52	0.0433	0.2228	116.29	0.2380	124.35	0.2522
36	77.97	0.01254	0.7032	20.49	108.71	0.0445	0.2225	116.52	0.2376	124.59	0.2518
38	80.81	0.01258	0.6791	21.09	108.90	0.0457	0.2222	116.74	0.2373	124.84	0.2515
40	83.72	0.01262	0.6559	21.70	109.09	0.0469	0.2218	116.96	0.2369	125.08	0.2511
42	86.69	0.01266	0.6339	22.29	109.27	0.0481	0.2215	117.18	0.2366	125.32	0.2508
44	89.74	0.01270	0.6126	22.90	109.45	0.0493	0.2211	117.40	0.2363	125.56	0.2504
46	92.88	0.01274	0.5922	23.50	109.63	0.0505	0.2208	117.61	0.2359	125.80	0.2501
48	96.10	0.01278	0.5726	24.11	109.80	0.0516	0.2205	117.82	0.2356	126.04	0.2497
50	99.40	0.01282	0.5537	24.73	109.98	0.0528	0.2201	118.02	0.2353	126.27	0.2494
52	102.8	0.01286	0.5355	25.34	110.14	0.0540	0.2198	118.22	0.2350	126.50	0.2491
54	106.2	0.01290	0.5184	25.95	110.30	0.0552	0.2194	118.42	0.2347	126.73	0.2488
56	109.8	0.01294	0.5014	26.58	110.47	0.0564	0.2191	118.62	0.2343	126.96	0.2484
58	113.5	0.01299	0.4849	27.22	110.63	0.0576	0.2188	118.82	0.2340	127.19	0.2481
60	117.2	0.01303	0.4695	27.83	110.78	0.0588	0.2185	119.01	0.2337	127.42	0.2478
62	121.0	0.01307	0.4546	28.46	110.93	0.0600	0.2181	119.21	0.2334	127.65	0.2475
64	124.9	0.01312	0.4403	29.09	111.08	0.0612	0.2178	119.40	0.2331	127.87	0.2472
66	128.9	0.01316	0.4264	29.72	111.22	0.0624	0.2175	119.59	0.2327	128.10	0.2469
68	133.0	0.01320	0.4129	30.35	111.35	0.0636	0.2172	119.77	0.2324	128.32	0.2466
70	137.2	0.01325	0.4000	30.99	111.49	0.0648	0.2168	119.96	0.2321	128.54	0.2463
72	141.5	0.01330	0.3875	31.65	111.63	0.0661	0.2165	120.15	0.2318	128.76	0.2460
74	145.9	0.01334	0.3754	32.29	111.75	0.0673	0.2162	120.32	0.2315	128.97	0.2457
76	150.4	0.01339	0.3638	32.94	111.88	0.0684	0.2158	120.50	0.2312	129.19	0.2455
78	155.0	0.01344	0.3526	33.61	112.01	0.0696	0.2155	120.67	0.2309	129.40	0.2452
80	159.7	0.01349	0.3417	34.27	112.13	0.0708	0.2151	120.85	0.2306	129.61	0.2449
82	164.5	0.01353	0.3313	34.92	112.24	0.0720	0.2148	121.02	0.2303	129.82	0.2446
84	169.4	0.01358	0.3212	35.60	112.36	0.0732	0.2144	121.18	0.2300	130.02	0.2443
86	174.5	0.01363	0.3113	36.28	112.47	0.0744	0.2140	121.34	0.2297	130.23	0.2441
88	179.6	0.01368	0.3019	36.94	112.57	0.0756	0.2137	121.50	0.2294	130.43	0.2438

Table 2 .... Properties of Monochlorodifluoromethane ($CHClF_2$)[a] (*Concluded*)

Sat. Temp. F	Abs. Press. lb per Sq in.	Volume		Enthalpy and Entropy Taken from −40 F							
				Enthalpy		Entropy		50 Deg Superheat		100 Deg Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
90	184.8	0.01374	0.2928	37.61	112.67	0.0768	0.2133	121.66	0.2291	130.63	0.2435
92	190.1	0.01379	0.2841	38.28	112.76	0.0780	0.2130	121.82	0.2288	130.83	0.2432
94	195.6	0.01384	0.2755	38.97	112.85	0.0792	0.2126	121.97	0.2285	131.03	0.2429
96	201.2	0.01390	0.2672	39.65	112.93	0.0803	0.2122	122.12	0.2282	131.23	0.2427
98	206.8	0.01396	0.2594	40.32	113.00	0.0815	0.2119	122.26	0.2279	131.42	0.2424
100	212.6	0.01402	0.2517	40.98	113.06	0.0827	0.2115	122.40	0.2276	131.61	0.2421
102	218.5	0.01408	0.2443	41.65	113.12	0.0839	0.2111	122.53	0.2273	131.80	0.2418
104	224.6	0.01414	0.2370	42.32	113.16	0.0851	0.2107	122.66	0.2270	131.99	0.2416
106	230.7	0.01420	0.2301	42.98	113.20	0.0862	0.2104	122.79	0.2267	132.17	0.2413
108	237.0	0.01426	0.2233	43.66	113.24	0.0874	0.2100	122.92	0.2264	132.35	0.2411
110	243.4	0.01433	0.2167	44.35	113.29	0.0886	0.2096	123.04	0.2261	132.53	0.2408
112	249.9	0.01440	0.2104	45.04	113.34	0.0898	0.2093	123.16	0.2258	132.71	0.2405
114	256.6	0.01447	0.2043	45.74	113.38	0.0909	0.2089	123.28	0.2255	132.88	0.2403
116	263.4	0.01454	0.1983	46.44	113.42	0.0921	0.2085	123.40	0.2253	133.05	0.2400
118	270.3	0.01461	0.1926	47.14	113.46	0.0933	0.2081	123.51	0.2250	133.22	0.2398
120	277.3	0.01469	0.1871	47.85	113.52	0.0945	0.2078	123.62	0.2247	133.39	0.2395

Data from Kinetic Chemicals, Inc., 1945. [a] ASRE designation—Refrigerant 22.

52.7 psia and 121 psia with dichlorodifluoromethane, CCl_2F_2, as the refrigerant. Determine: (*a*) the cooling effect provided by each pound of refrigerant; (*b*) the refrigerant circulating rate; (*c*) the horsepower required; (*d*) the quantity of heat to be dissipated from the condenser; (*e*) the required condenser cooling water, in gallons per minute, if the temperature rise of water passing through the condenser is 8 deg; (*f*) the bore and stroke of a double acting cylinder (neglecting the effect of the piston rod) if speed of compressor is 500 revolutions per minute; (*g*) coefficient of performance.

Solution: (*a*) Saturated liquid CCl_2F_2 at 121 psia leaves the condenser and enters the expansion valve. The enthalpy of this material (from Table 1) is 29.68 Btu per pound, and this must also be its enthalpy at entrance to the evaporator. Leaving the evaporator as a saturated vapor at 52.7 psia, its enthalpy is 82.82, so the refrigerating effect must be 82.82 − 29.68 = 53.14 Btu per pound.

(*b*) The refrigerant circulating rate is equal to the total heat to be picked up in unit time, divided by the pickup per pound of refrigerant or,

$$W_r = (7 \text{ ton} \times 200) + 53.14 = 26.3 \text{ lb per minute.}$$

(*c*) The horsepower required is equal to the increase in energy of the refrigerant passing through the compressor (expressed in Btu per minute) divided by the conversion factor 42.42, which is the number of Btu per minute corresponding to 1 hp,

$$(\text{hp}) = W_r(h_d - h_{ss}) \div 42.42 \qquad (7)$$

where

(hp) = horsepower.
W_r = refrigerant circulating rate in pounds per minute.
h_d = enthalpy of vapor at condition of discharge from compressor.
h_{ss} = enthalpy of saturated vapor entering compressor.

W_r is known from (*b*) and h_{ss} is the enthalpy of refrigerant as it enters the compressor in a saturated vapor state at 52.7 psia; thus h_{ss} = 82.82.

In order to determine h_d, the state of the refrigerant must first be determined at the compressor discharge. At the known suction state the entropy (from Table 1 for saturated vapor at 52.7 psia) is 0.16828 and, since the compression is assumed to occur isentropically, it therefore follows that the discharge stage must have the same entropy at 121 psia. From the table the entropy of vapor superheated 25 deg is 0.17330, so the superheat, t_{sd}, possessed by the actual gas discharged from this compressor can be obtained by interpolation as,

$$\frac{t_{sd}}{25} = \frac{0.16828 - 0.16608}{0.17330 - 0.16608}$$

from which t_{sd} = 7.6 deg.

As the saturation temperature at 121 psia is 94 F the actual temperature, t_d, of the vapor leaving the compressor is, t_d = 94 + t_{sd} = 94 + 7.6 = 101.6 F. By the same kind of interpolation the enthalpy of the discharged vapor can be determined from the enthalpies given for vapor superheated 25 F and for saturated vapor,

$$\frac{(h_d - 88.10)}{(92.16 - 88.10)} = \frac{(0.16828 - 0.16608)}{(0.17330 - 0.16608)}$$

from which, h_d = 89.34 Btu per pound.

Then substituting in Equation 7,

$$(\text{hp}) = 26.3\ (89.34 - 82.82) \div 42.42 = 4.03.$$

(*d*) The rate of heat loss from the condenser, Q_c, must be equal to the sum of the energies picked up by the refrigerant in the evaporator and the compressor, Q_c = 53.14 + (89.34 − 82.82) = 53.14 + 6.52 = 59.66 Btu per pound or 26.3 × 59.66 = 1569 Btu per minute. This same figure can, of course, be determined more directly by subtraction of the enthalpy

Table 3 . . . . Properties of Trichloromonofluoromethane (CCl_3F)[a]

Sat. Temp. F	Abs. Press. lb per Sq In.	Volume		Enthalpy and Entropy Taken from −40 F							
				Enthalpy		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	2.59	0.01020	13.700	7.81	90.4	0.0178	0.1975	93.9	0.2049	97.4	0.2120
5	2.96	0.01024	12.100	8.81	91.2	0.0200	0.1974	94.7	0.2047	97.2	0.2117
10	3.38	0.01028	10.700	9.82	92.0	0.0222	0.1973	95.5	0.2045	99.0	0.2114
15	3.85	0.01032	9.530	10.80	92.8	0.0243	0.1971	96.3	0.2043	99.8	0.2111
20	4.36	0.01036	8.490	11.90	93.7	0.0264	0.1970	97.2	0.2041	100.7	0.2109
25	4.94	0.01040	7.580	12.90	94.5	0.0286	0.1969	98.0	0.2039	101.5	0.2107
30	5.57	0.01045	6.770	13.90	95.3	0.0307	0.1969	98.8	0.2038	102.3	0.2105
35	6.27	0.01049	6.080	14.90	96.1	0.0328	0.1968	99.6	0.2037	103.1	0.2103
40	7.03	0.01053	5.460	16.00	96.8	0.0349	0.1968	100.3	0.2036	103.8	0.2101
45	7.88	0.01057	4.920	17.00	97.6	0.0370	0.1967	101.1	0.2035	104.6	0.2099
50	8.79	0.01062	4.440	18.10	98.4	0.0391	0.1967	101.9	0.2034	105.4	0.2098
55	9.80	0.01066	4.020	19.10	99.2	0.0412	0.1967	102.7	0.2033	106.2	0.2097
60	10.90	0.01071	3.640	20.20	100.0	0.0432	0.1967	103.5	0.2033	107.0	0.2096
65	12.10	0.01076	3.300	21.30	100.8	0.0453	0.1967	104.3	0.2032	107.8	0.2094
70	13.40	0.01081	3.000	22.40	101.5	0.0473	0.1967	105.0	0.2032	108.5	0.2093
75	14.80	0.01086	2.740	23.50	102.2	0.0493	0.1967	105.7	0.2031	109.2	0.2092
80	16.30	0.01091	2.500	24.50	102.9	0.0513	0.1966	106.4	0.2030	109.9	0.2090
85	17.90	0.01096	2.280	25.60	103.6	0.0533	0.1966	107.1	0.2029	110.6	0.2089
90	19.70	0.01101	2.090	26.70	104.4	0.0553	0.1966	107.9	0.2028	111.4	0.2088
95	21.60	0.01106	1.918	27.80	105.1	0.0573	0.1966	108.6	0.2028	112.1	0.2087
100	23.60	0.01111	1.761	28.90	105.7	0.0593	0.1965	109.2	0.2027	112.7	0.2085
105	25.90	0.01116	1.620	30.10	106.4	0.0613	0.1965	109.9	0.2026	113.4	0.2084

[a] ASRE designation—Refrigerant 11.

of liquid leaving the condenser from the enthalpy of superheated vapor going into it, thus,

$$Q_c = 26.3\ (89.34 - 29.68) = 1569 \text{ Btu per minute.}$$

(e) The cooling water rate (based on a gallon as 8.34 lb) is $1569 \div (8 \times 8.34) = 23.5$ gpm.

(f) The compressor size is fixed by the volume of gas which must be drawn into the machine per unit time. Saturated vapor at 52.7 psia has a specific volume, from Table 1, of 0.779 cu ft per pound, hence $26.3 \times 0.779 = 20.49$ cfm of gas must be handled. Assuming a volumetric efficiency of 90 percent, the compressor must then displace $20.49 \div 0.9 = 22.8$ cfm. The speed is given as 500 rpm and, as the unit is known to be double-acting, the displacement is therefore $(22.8 \times 1728) \div (2 \times 500) = 39.4$ cu in. If the unit were designed so that bore d and stroke were the same,

$$(\pi d^3) \div 4 = 39.4 \qquad d = 3.69 \text{ in.}$$

(g) $$\text{(CP)} = (h_{ss} - h_{fc}) + (h_e - h_{se})$$
$$= (82.82 - 29.68) + (89.34 - 82.82) = 8.17$$

where h_{fc} is the specific enthalpy of liquid at discharge from the condenser.

The coefficient of performance of *Example 1* may be compared with that of an ideal system operating on the Carnot cycle between the same temperature limits. Then $T_s = 501$ F (which is 41 F + 460) and $T_c = 554$ F (which is 94 F + 460) and,

$$\text{(CP)} = \frac{501}{554 - 501} = 9.6$$

The actual cycle is therefore $8.17 \div 9.6$ or 85 percent as effective as a Carnot cycle between the same temperature limits.

Influence of Suction Pressure

Brief consideration of the analytical procedure used in discussion of the simple saturation cycle will bring out the need for maintaining the suction pressure on any refrigeration system as high as the load will permit. As the suction pressure increases, for fixed discharge pressure, the enthalpy of refrigerant entering the evaporator remains unchanged, but the leaving enthalpy increases and, hence, the refrigerating effect increases. Further, compressor energy input is reduced not merely because of the greater enthalpy of the gas at suction, but also because of a reduction in the enthalpy of the superheated gas at discharge. Since the refrigerating effect is greater and the work less, it is obvious that there will be a substantial gain in the coefficient of the performance. See Chapter 54, Fig. 1.

The actual value of suction pressure on any system is obviously determined by the required temperature which must be maintained in the conditioned space. For a direct-expansion system the evaporator can be held at a temperature not much less than that of the conditioned enclosure, except in cases where lower temperatures may be needed in order to establish a desired ratio of dehumidifying to cooling load. When dehumidification requirements dictate the use of unusually low evaporator temperatures, the increased operating cost should properly be charged against the dehumidification rather than the sensible cooling.

Table 4 . . . . Properties of Trichlorotrifluoroethane ($C_2Cl_3F_3$)[a]

Temp	Pressure		Volume		Density		Enthalpy from −40°F			Entropy from −40°F	
F	psia	psig	Liquid cu ft/lb	Vapor cu ft/lb	Liquid lb/cu ft	Vapor lb/cu ft	Liquid Btu/lb	Latent Btu/lb	Vapor Btu/lb	Liquid Btu/lb/ F deg	Vapor Btu/lb/ F deg
0	0.8377	28.21*	0.00966	31.31	103.56	0.03194	7.98	70.92	78.89	0.0182	0.1725
4	.9503	27.99*	.00968	27.84	103.27	.03592	8.78	70.68	79.46	.0199	.1724
8	1.075	27.73*	.00971	24.81	102.98	.04031	9.59	70.44	80.03	.0216	.1723
12	1.213	27.45*	.00974	22.17	102.69	.04511	10.41	70.20	80.61	.0234	.1722
16	1.366	27.14*	.00977	19.84	102.40	.05040	11.22	69.96	81.18	.0251	.1722
20	1.534	26.80*	0.00979	17.81	102.10	0.05616	12.03	69.72	81.75	0.0268	0.1722
24	1.719	26.42*	.00982	16.02	101.81	.06243	12.85	69.48	82.33	.0285	.1721
28	1.922	26.01*	.00985	14.43	101.51	.06929	13.67	69.24	82.91	.0302	.1722
32	2.145	25.55*	.00988	13.03	101.21	.07675	14.49	69.00	83.49	.0318	.1722
36	2.388	25.06*	.00991	11.79	100.91	.08483	15.32	68.75	84.07	.0335	.1722
40	2.655	24.52*	0.00994	10.68	100.60	0.09361	16.16	68.50	84.65	0.0352	0.1723
44	2.944	23.93*	.00997	9.703	100.30	.1031	16.99	68.25	85.24	.0368	.1723
48	3.258	23.29*	.01000	8.830	99.99	.1133	17.82	68.00	85.82	.0385	.1724
52	3.602	22.59*	.01003	8.044	99.68	.1243	18.66	67.74	86.40	.0401	.1726
56	3.973	21.83*	.01006	7.342	99.37	.1362	19.50	67.48	86.98	.0418	.1727
60	4.374	21.02*	0.01010	6.713	99.05	0.1490	20.35	67.22	87.57	0.0434	0.1728
64	4.807	20.14*	.01013	6.149	98.73	.1626	21.19	66.96	88.15	.0450	.1729
68	5.275	19.18*	.01016	5.640	98.42	.1773	22.05	66.69	88.74	.0467	.1731
72	5.780	18.16*	.01019	5.180	98.10	.1931	22.90	66.43	89.33	.0483	.1732
76	6.320	17.06*	.01023	4.769	97.77	.2097	23.76	66.16	89.92	.0499	.1734
80	6.902	15.87*	0.01026	4.392	97.45	0.2277	24.63	65.88	90.51	0.0515	0.1736
84	7.527	14.60*	.01030	4.051	97.12	.2468	25.49	65.60	91.09	.0531	.1738
88	8.194	13.24*	.01033	3.742	96.79	.2672	26.36	65.32	91.68	.0547	.1740
92	8.908	11.79*	.01037	3.463	96.46	.2888	27.24	65.04	92.28	.0563	.1742
96	9.668	10.24*	.01040	3.208	96.13	.3117	28.11	64.75	92.86	.0578	.1744
100	10.48	8.59*	0.01044	2.976	95.79	0.3360	28.99	64.46	93.45	0.0594	0.1746
104	11.35	6.82*	.01048	2.762	95.46	.3620	29.89	64.16	94.05	.0610	.1748
108	12.28	4.93*	.01051	2.567	95.12	.3896	30.78	63.86	94.64	.0626	.1751
110	12.76	3.95*	0.01053	2.477	94.95	0.4038	31.22	63.71	94.93	0.0634	0.1752

* Inches of mercury below one atmosphere.
† Standard cycle temperatures.
[a] ASRE designation—Refrigerant 113.

Influence of Discharge Pressure

In contrast to suction pressure, the compressor discharge pressure should be kept as low as operating conditions will allow. This pressure must be high enough to provide a saturation temperature of refrigerant within the condenser that is greater than the exit temperature of the cooling water. The discharge pressure therefore is a direct function of the temperature of the cooling fluid, and will automatically rise whenever the temperature of cooling water (or air) rises; it will also rise when the flow rate of the cooling medium is decreased.

Increase in discharge pressure (for fixed suction pressure) raises the enthalpy of the gas leaving the compressor; hence, increases the work of compression. Further, as the enthalpy of saturated liquid leaving the condenser increases with pressure, the refrigerating effect must decrease. Thus the effect of such a pressure rise is to require more work per pound of refrigerant handled, and at the same time to necessitate an increase in the refrigerant flow rate. (See Chapter 54, Fig. 1.)

Influence of Water Jacket

The preceding discussion has, in every case, assumed isentropic compression. Where exact performance data are not available, this assumption is a desirable one since it leads to a conservatively large determination of the power required. In most actual systems, the compression process departs from isentropic due to irreversible heat transfers which occur between the vapor in the cylinder and the cylinder wall, and also because of intentional heat dissipation from the outside of the cylinder walls to the surroundings, or to a cooling fluid passing through a water jacket around the

cylinder. Compressor cooling is highly desirable as a method of reducing power consumption.

Influence of Superheating and Subcooling

The most common departure from conditions of the simple saturation cycle is that resulting from admission of superheated vapor to the compressor. Thermodynamically, superheat is undesirable because the enthalpy increase required to compress a vapor through a given pressure range increases with superheat. Further, superheated vapor leaving an evaporator is usually an indication that the suction pressure is lower than necessary. Under practical operating conditions, however, superheat is almost universally used as a

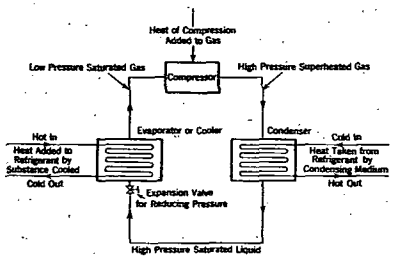


Fig. 3 . . . . Mechanical Refrigeration System

means of assuring complete vaporization of the refrigerant going to the compressor. With modern compressors operating at high speed, and with relatively small clearance space, it is particularly necessary to avoid admission through the suction valves of liquid refrigerant.

Another common departure of actual systems from the simple saturation cycle occurs because of subcooling of refrigerant in the condenser. Thermodynamically, such subcooling is advantageous since it increases the refrigerating effect without affecting the unit energy requirements of the compressor. Further, it can be shown that for a fixed ratio of condenser cooling water to refrigerant circulating rate, the total compressor power requirements will be greater when operating at simple saturation than when operating with maximum subcooling. What is even more surprising is that condenser pressure may be lower for the subcooling cycle than for the saturation cycle. This condition results from the fact that, for the same capacity on a heavily loaded condenser, the refrigerant flow rate is less when there is subcooling.

Because of the advantages attendant upon the use of subcooling, many methods are in use for obtaining some subcooling effect outside of the condenser. One common procedure is to use the cold vapor leaving the evaporator to cool the liquid flowing from condenser to expansion valve.

Another somewhat unusual subcooling cycle allows cold refrigerant from the downstream side of the expansion valve to cool liquid refrigerant from the condenser down to the evaporator temperature. Fig. 5 shows the pressure-enthalpy diagram for a typical refrigeration cycle operating with both subcooling of the refrigerant from the condenser and superheating of the refrigerant leaving the evaporator.

Complex Refrigeration Cycles

The preceding sections have dealt only with refrigeration systems in which there is but one evaporator; compression is accomplished through but a single stage, and expansion proceeds through a single expansion valve. In large systems or in low-temperature systems in which the compression ratio is high, the compression process can be carried out in stages, with the refrigerant passing through several cylinders arranged for operation in series. The thermodynamic advantage of such compound compression arises from the fact that intercoolers can be placed between the stages of compression to extract heat from the vapor, and thereby cause the overall compression process to approach more closely the ideal condition of isothermal compression. Essentially, such intercoolers serve the same purpose as a cooling jacket, but with greater effectiveness because of the more satisfactory heat-transfer conditions.

In the simple saturation cycle the saturated liquid entering the expansion valve commences to vaporize as soon as its pressure starts to drop. The vapor produced during the expansion process has no further use, in terms of refrigerating effect, since it has already picked up its latent heat of vaporization as a result of heat which it has extracted from the unvaporized residue. Thus the instant such vapor forms, its usefulness is at an end, and to allow such material to undergo a further drop in pressure is uneconomical. With compound compression, there is at least one intermediate pressure at which flash vapor can be extracted. In such cases several expansion valves can be utilized with all of the refrigerant from the condenser passed through a first expansion valve to the higher suction pressure, and the flash vapor then extracted and returned to the condenser through the high compression stage. The remaining refrigerant can then pass through a second expansion valve where the pressure is dropped to that corresponding to the low-pressure evaporator. The number of expansion valves is limited by the number of stages of compression.

Further cycle complications may arise if more than one evaporator is to be operated with a single compressor, and particularly, if the pressures in these evaporators are to differ. The most common solution is to operate the compressor at the suction pressure of the lowest pressure evaporator, and to equip all other evaporators with back-pressure regulating valves or throttling devices between the evaporator and the compressor suction.

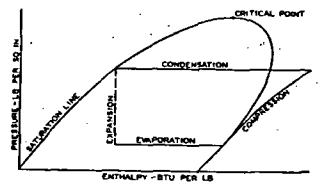


Fig. 4 . . . . Pressure-Enthalpy Diagram for Simple Saturation Cycle

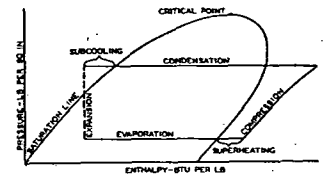


Fig. 5 . . . . Pressure-Enthalpy Diagram for Refrigeration Cycle with Subcooling and Superheating

The Air-Cycle System

Fundamentally, the air cycle[1] is essentially the same as the vapor cycle. Compression is accomplished by a reciprocating or centrifugal compressor, and, since there is no change of phase of the refrigerant upon expansion, an *air cooler* replaces the condenser, and a *refrigerator* replaces the evaporator. Although some cooling would result from the expansion of the gas through an ordinary expansion valve, a much greater drop in air temperature is accomplished if the expansion is controlled to approach the isentropic by replacing the valve with an expansion engine or turbine. Furthermore, the work recovered by such an expansion engine can be utilized to supply part of the work of compression or to drive other devices.

The Steam-Jet System

The *steam-jet system*, under certain circumstances, is desirable for use in air conditioning.[2] Steam supplies directly the power used for compressing the refrigerant, thus eliminating the losses connected with other methods of supplying energy. As the compression ratio between the evaporator and condenser under normal circumstances is large, the mechanical efficiency of the equipment is somewhat lower than that of the positive mechanical type compressor. The condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam-jet system functions on the principle that water under high vacuum will vaporize at low temperatures. Steam-jet boosters or compressors of the type commonly used in power plants for various processes, will produce the necessary low absolute pressure to cause evaporation of the water.

A diagrammatic representation of a typical steam-ejector water-cooling system is shown in Fig. 6. The figures correspond to an average representative system. The water to be cooled enters the evaporator and is cooled to a temperature corresponding to the vacuum maintained. Because of the high vacuum, a small amount of the water introduced in the evaporator is flashed into steam. As this requires heat, and the only source of heat is the rest of the water in the evaporator tank, this other water is almost instantly cooled to a temperature corresponding to the boiling point determined by the vacuum maintained. The amount of water flashed into steam is a small percentage of the total water circulated through the evaporator, amounting to approximately 11 lb per (hr) (ton of refrigeration developed). The remainder of the water at the desired low temperature is pumped out of the evaporator and used at the point where it is required.

The ejector compresses the vapor which has been flashed in the evaporator, plus any entrained air taken from the circulated water, to a somewhat higher absolute pressure. The vapor and air mix with the impelling steam on the discharge side of the jet, and the total mixture then passes from the ejector into the condenser.

The slight amount of air which may be entrained in the cooled water is removed by a small secondary ejector that raises the pressure sufficiently so that the air can be discharged to the atmosphere. A secondary condenser is then necessary to condense the steam in the secondary jet.

While a single booster of smaller than 15 tons capacity is difficult to build, steam-jet vacuum-cooling units have been built for as small as 5 to 6 tons capacity. They can readily be built for steam pressures of from 5 to 200 psig, and condenser water temperatures as high as 90 F. The

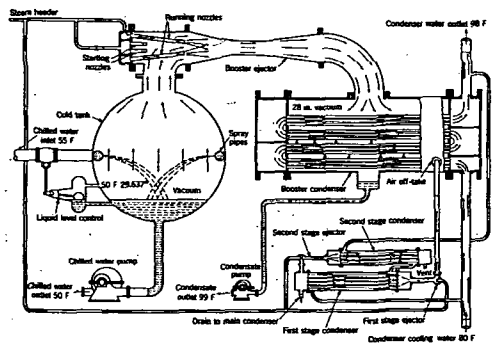


Fig. 6 . . . . Diagrammatic Arrangement of Steam-Jet Vacuum-Cooling Unit

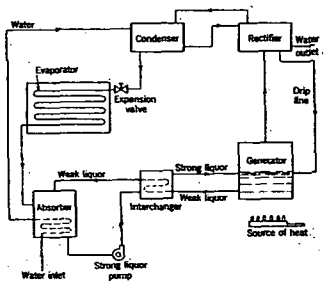


Fig. 7 . . . . Closed Absorption System

steam consumption in pounds per hour per ton of refrigeration increases rapidly as the booster steam pressure is lowered. For example, the lowering of the booster steam pressure from 200 to 90 psig results in an increase in steam consumption of approximately 5 percent, whereas a further decrease in booster steam pressure to 10 psig increases steam consumption by approximately 72 percent over that required at 200 psig.

The capacity of a steam-jet system is usually controlled by controlling the number of boosters in use since the unit usually has several boosters operating on the same evaporator. Usually one booster is automatically controlled, whereas the others are manually operated. The capacity is dependent, as for all compressors, upon the evaporator *temperature*, or in other words, the suction pressure. For example, the capacity is lowered approximately 17 percent if the evaporator or chilled water temperature is lowered from 50 to 45 F. The capacity therefore can be controlled to some extent by regulating the evaporator temperature.

The Absorption System

The absorption and mechanical compression cycles have in common the vaporizing and condensing of a liquid, but differ in that one has a compressor unit and the other an absorber-generator unit for producing the necessary pressure difference between vaporizing and condensing levels. Both cycles require energy for operation: mechanical energy for the compression cycle; heat for generation in the absorption cycle.

Thermodynamic analysis of absorption cycles is relatively complex and requires the use either of tables or graphs showing the equilibrium relationships and thermodynamic properties of the refrigerant-absorbent combination. Data of this kind are given in the first book in the bibliography. A discussion of various absorbents is given in the sixth book. Thermodynamically, the effectiveness of a refrigerant-absorbent combination increases directly with its negative deviation from Raoult's Law.

Fig. 7 shows a typical absorption cycle flow diagram. Cooling water first goes through the absorber (where it extracts the heat of absorption which is liberated by the refrigerant vapor as it goes into solution), then through the condenser, and finally through the rectifier. Refrigerant from the evaporator enters the absorber where it goes into solution in the absorbent; the high concentration solution is then pumped to the generator where heat is supplied; the refrigerant (with some absorbent vapor) leaves for the rectifier and the warm low concentration solution is returned to the absorber. In the rectifier selective condensation occurs, the concentration of the absorbent in the condensate being much greater than its concentration in the entering vapor mixture; rectifier condensate is dripped back to the generator.

Operating Cycle. The components of one form of lithium bromide water absorption system are illustrated in Fig. 8.

Diagram A shows two closed vessels, with salt (lithium bromide) solution in one and water in the other. The salt solution in the absorber has affinity for water vapor and evaporates some of the water. The water that remains in the evaporator is thus cooled by evaporation, thereby producing a refrigeration effect.

Since these two vessels operate under a high vacuum, chilled water temperatures down to 38 F may be obtained. To utilize the refrigeration effect, a coil is placed inside the evaporator as shown in diagram B. A pump circulates water from the evaporator tank to a spray header to wet the surface of the coil. The evaporation and cooling effect of the spray on the coil surface chills the water as it is circulated through the coil in a closed circuit to a refrigeration load.

In an actual operating cycle the salt solution is continu-

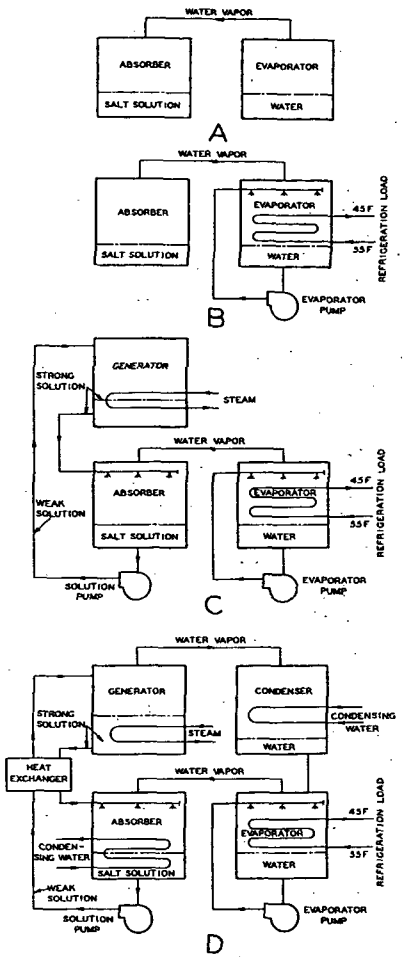


Fig. 8 .... Components of Lithium Bromide Absorption System

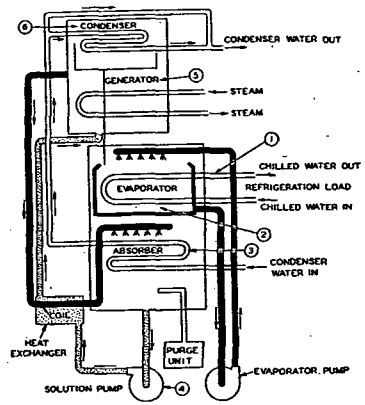


Fig. 9 .... Diagram of Lithium Bromide Water Absorption System

*ally absorbing water vapor, thus diluting the solution and reducing its ability to absorb additional water vapor. To keep the salt solution in the absorber at the proper strength a generator is added to the system as shown in diagram C. The weak solution is pumped from the absorber to the generator where heat is used to boil off the excess water vapor. The reconcentrated salt solution is then returned to the absorber to continue the cycle.

The water vapor boiled off in the generator is condensed and returned to the evaporator. A heat exchanger is added to the system (diagram D) to conserve heat in the cycle by using the hot concentrated salt solution from the generator to preheat the cooler weak solution from the absorber. Condensing water from a cooling tower or other available source is circulated through coils in the absorber and condenser to remove waste heat from the refrigeration cycle.

For the sake of simplification, the cycle is shown as operating with four separate chambers. In actual practice, the condenser and generator are combined into a *high pressure* shell, with the absorber and evaporator located in a second shell. Fig. 9 schematically shows the arrangement. A purge unit is included to remove any non-condensables from the machine, thus assuring maximum efficiency and capacity at all times.

Coefficient of Performance. The coefficient of performance (CP) of an absorption system using lithium-bromide and water is approximately 0.6. The heat required can be obtained from steam, hot water or other heated liquids.

Absorption type equipment is fully automatic from full load to no load. The performance does not vary greatly between 20 percent of capacity and full load. The operating costs of absorption equipment are considered competitive with those of other forms of refrigeration. For a typical lithium-bromide-water system, Figs. 10, 11, and 12 illus-

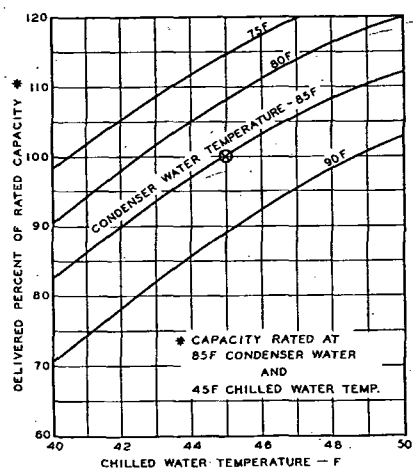


Fig. 10 .... Performance Characteristics of a Typical Lithium Bromide Absorption System

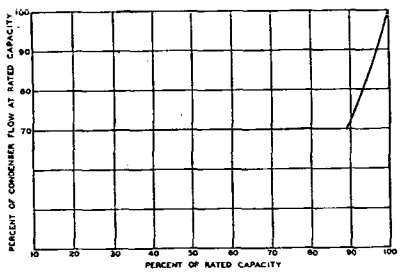


Fig. 11 .... Condenser Water Requirements for a Typical Lithium Bromide Absorption System

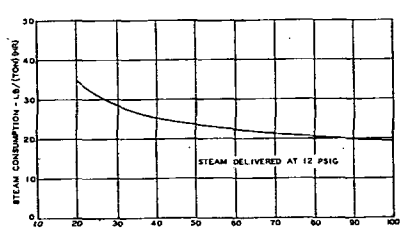


Fig. 12 .... Steam Requirements for a Typical Lithium Bromide Absorption System

trate respectively the performance characteristics, the condenser water requirements, and the steam requirements.

BASIC REFRIGERATION EQUIPMENT

Compression Refrigeration Machines

Compression of the refrigerant gas drawn from the evaporator may be accomplished by one of several means. Positive displacement may be used as in the reciprocating; centrifugal force may be applied as in the centrifugal compressor; an ejector may be used as in the steam jet refrigeration cycle; or absorption of a low pressure refrigerant gas in a secondary fluid, followed by the absorbent's release upon application of heat, may be utilized. A detailed discussion of the equipment required for each of these types of systems is beyond the scope of this chapter. For a more comprehensive treatment, reference may be made to the bibliography. The present discussion is limited to positive displacement reciprocating compressors, and centrifugal compressors.

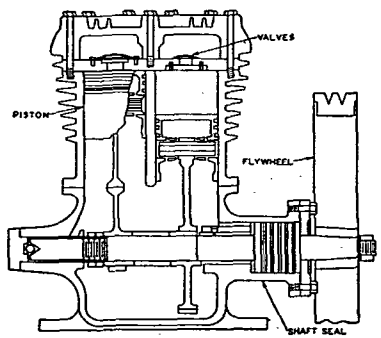


Fig. 13 .... Diagrammatic View of a Reciprocating Compressor

Reciprocating Compressors

Reciprocating compressors (see Fig. 10) may be classified according to (*a*) cylinder design, (*b*) compressor drive, (*c*) valves, and (*d*) lubrication and cooling.

Cylinder Design. Cylinder design may vary as to number, arrangement, and action (i.e., single-acting or double-acting). Single-acting compressors usually have their cylinders arranged vertically, radially, or in a V or W shape arrangement. Double-acting compressors, with refrigerant gas drawn in, and compressed on both the head and crank ends of the cylinder, are usually arranged horizontally. Reciprocating units are available with from one to sixteen cylinders with the V, W, or radial arrangements best adapted to the greatest numbers. The present trend is toward higher operating speeds with a low displacement per cylinder, together with an increase in the number of cylinders. Whereas the original reciprocating compressors were slow-speed (50 to 55 rpm) steam-driven devices, modern electric motor-driven compressors range up to 3500 rpm. Cylinder heads are usually bolted tight to the cylinders, but in some large compressors where there is danger of wet compression or of foreign materials entering the compressor space, a secondary head known as a safety head, may be seated at the end of the cylinder and held in position with heavy springs. Normally, this head remains stationary, but excessive pressures in the clearance space are relieved by movement of the safety head, and thus prevent damage to the cylinder.

Compressor Drives. Reciprocating compressors may be subdivided on the basis of source of motive power, and whether they are open or hermetic. Practically all modern compressors are electric motor-driven, although a few large, steam-driven compressors are still being installed where steam forms the most economical source of energy.

For summer air conditioning, the compressors may be driven by medium-speed heavy-duty internal combustion engines using natural gas as the fuel. Field tests indicated that fuel consumption of natural gas engine compressor drives is 13,000 Btu per ton-hour.

The natural gas engine compressor drives are designed as packaged units which combine engine speed variation and compressor cylinder unloading to meet fluctuating cooling requirements. Fuel consumption is in direct proportion to the actual cooling load throughout over 80 percent of the capacity range.

The division of compressors into open or closed types is dependent upon whether the motive power is received from an external source, or whether the motor is direct drive and sealed within the housing. In the open type, power is applied from an external source to one end of the compressor crankshaft extending through the crankcase, usually by means of a V-belt drive. The point of emergence of the shaft from the crankcase forms a weak point of refrigerant leakage, and is most frequently sealed with a bellows type crankshaft seal. Horizontal double-acting compressors operate with a sliding piston rod, moving back and forth through a stuffing box. If the motor is direct-drive and enclosed within the compressor housing, the compressor is classified as closed or hermetic. This eliminates the necessity of any shaft seal, and not only prevents refrigerant leakage at this point, but reduces operating noise. One disadvantage is the inaccessibility of moving parts for repairs, but lubrication is greatly simplified since both the motor and compressor operate in a sealed space containing the lubricating oil.

Compressor Valves. All refrigeration compressor valves are dependent for their operation upon a difference in pressure between the inside of the cylinder and the suction or discharge line. Although mechanically-operated valves might have some advantage, they have proved unsatisfactory because each change in the evaporator or condenser-operating pressures requires a change of valve setting. The pressure differentials required for operation of the valves depend upon the valve design and the compressor speed. The suction and discharge valves may be arranged with both located in the compressor head, or with the suction valve on the top of the piston and the discharge valve in the compressor head (uniflow arrangement). The valves themselves are usually classified as either poppet, ring-plate, or flexing.

Lubrication and Cooling. Lubrication of modern compressors is accomplished by either splash lubrication or forced lubrication. The latter is used on large compressors, while simple splash lubrication is used in the smaller units.

Large compressors are usually water-cooled with the water jacket either cooling the cylinder walls, or both the cylinder walls and the compressor head. Small compressors are either water-cooled or air-cooled with extended finned surfaces cast on the exterior of the cylinder. In a few cases small compressors may be found in which there is no attempt to add any purposive cooling other than through non-finned surfaces to the lower temperature air.

Water cooling is more effective than air cooling, but even under the best conditions cylinder cooling removes only a portion of the superheat in the refrigerant gas. This removal of heat from the cylinder results in some decrease in the work of compression, as well as reduction in condenser load.

Control of Reciprocating Compressors

Capacity Control of reciprocating compressors to match the imposed load is accomplished by a controller responding to load variations, and operating one of the several means for changing the compressor capacity. This controller may respond to the condition being controlled, such as space temperature in the case of a direct-expansion system, or water temperature in the case of a water chiller; or it may respond to suction pressure, which varies with the imposed load. The common methods of varying the compressor capacity are:

1. *Starting and stopping the compressor.* This is more common on small units where frequent cycling of the compressor offers no serious complications. On large installations two or more smaller compressors may be employed and these may be started and stopped in sequence, thus providing more capacity steps: Sequence control of multiple compressors is accomplished by the use of a program or step controller, and in most cases this is arranged to return to the off position when the system is shut down or in the event of power failure, and to provide a suitable time delay between steps on start-up to prevent the full electrical load from being thrown across the line at one time. Means for manually or automatically varying the sequence to equalize wear on the compressors are sometimes employed.

2. *Cylinder Unloaders.* These usually are available on all but the smaller compressors and are used on both single and multiple compressor installations. They consist of a device built into the cylinder head to hold the suction valve open for unloading the cylinder. Oil pressure or head pressure is applied to these devices to load the cylinders, hence the compressor always starts at minimum capacity. Depending upon the make of compressor, the application of pressure to the unloaders is accomplished by:

a. Solenoid valves usually operated by the program controller in sequence with starting and stopping of the compressor or compressors. The program controller starts the compressor at minimum capacity, then loads the cylinders in sequence as the demand increases. If more than one compressor is used, it starts and loads the second compressor in sequence after the first is fully loaded, etc.

b. A self-contained stepping valve built into the compressor and actuated by a bellows measuring suction pressure. This valve is supplied with oil pressure from the compressor oil pump and has an oil outlet connection to each cylinder unloader. This valve supplies oil pressure to, or releases it from, the cylinder unloaders progressively as the suction pressure increases or decreases. The suction-pressure bellows is opposed by a spring which has an external manual adjustment for selection of the desired operating range. The other side of the bellows may be open to atmosphere so that the suction pressure is controlled with reference to atmospheric pressure, or it may be sealed and provided with an air connection which permits variation of the operating range as a function of air pressure from a pneumatic controller. For example, if the spring is so adjusted that the basic operating range is 35 to 40 psi suction pressure when the pressure from the pneumatic controller is zero, it will be increased to 45 to 50 psi when the control pressure is 10 psi. Thus these self-contained cylinder unloaders not only can control the suction pressure within a fixed range, but this range can be moved up or down automatically to meet the varying demands on the system. Pneumatic controllers used for this purpose often are supplied with air pressure up to 25 psi to permit a wide range of suction pressure variation. Most of these self-contained cylinder unloaders also can be arranged for mechanically varying the suction pressure by means of an electric motor operator acting against the manual adjustment spring to permit the use of proportional electric or electronic controls.

3. *Speed Control.* Multi-speed motors are sometimes used, actuated by a multi-stage thermostat or a proportional thermostat and a program or step controller.

Safety Controls are provided on reciprocating compressors to stop the motor whenever certain conditions are outside of safe limits. These conditions include:

1. *Low Suction Pressure.* This is a pressure controller responsive to the suction pressure. It stops the compressor when the suction pressure drops below a safe limit. It is sometimes also used as a capacity control, as described in preceding section, paragraph numbered 1, when it is desired to start and stop the compressor in accordance with the load demand as determined by changes in suction pressure. When other means are provided for starting and stopping the compressor in accordance with load, the low-pressure control functions solely as a safety device.

2. *High Head Pressure.* This is a pressure controller responsive to the high-side pressure. It stops the compressor when the high-side pressure exceeds a safe limit. This is strictly a safety device and frequently is of the manual-reset type so that the compressor cannot again start until the reset button is manually operated. In many cases the high- and low-pressure switches are combined into a single unit.

3. *Low Oil Pressure.* This is a pressure controller which stops the compressor in the event of low oil pressure. It also is frequently of the manual-reset type.

4. *Low Line Voltage.* This is a low-voltage cutoff which stops the compressor when the line voltage drops below a minimum value.

5. *High Motor Current.* This may be built into the motor or into the motor starter. It stops the motor in the event excessive current persists for more than the brief interval required in starting.

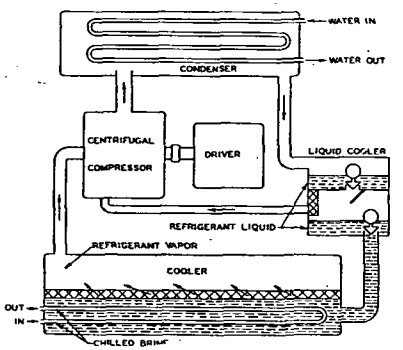


Fig. 14 .... Enclosed-Type Centrifugal Condensing Unit

6. *High Pressure Relief.* This is a device for preventing damage if excessive head pressures are encountered, and provides protection from damage beyond that provided by the high-pressure cutout. It may be a relief plug which blows out in the event of excessive pressure. This has the disadvantage of losing the entire refrigerant charge. A relief valve, which functions like a pop-safety valve, is preferred since it wastes only enough refrigerant to relieve the danger.

7. *Low Water Temperature.* If the compressor is used for chilling water, a low-limit thermostat usually is installed in the chiller to stop the compressor in the event a freezing temperature is approached. This is in addition to a controlling thermostat which also may respond to water temperature.

Centrifugal Compressors

Centrifugal compressors are used with very low pressure refrigerants; sometimes both evaporator and condenser work below atmospheric pressure. Trichlorotrifluoroethane ($C_2Cl_3F_3$) and monofluorotrichloromethane (CCl_3F) are the refrigerants commonly used in centrifugal machines.

Centrifugal compressors, like reciprocating compressors, can be divided into two general types, open and enclosed. In general, the open-type compressor is geared to the driving mechanism, and operates at higher speed than the driving

motor or turbine. A modern, completely enclosed, direct-driven centrifugal compressor is illustrated in Fig. 14.

Compression of the refrigerant is accomplished by means of centrifugal force; therefore, this type of compressor is inherently suitable for large volumes of refrigerant at low pressure differentials. Two or more stages are usually required and high speeds are necessary to obtain good efficiency.

The evaporator is usually constructed as an integral part of the centrifugal-type condensing unit, to chill water which is then circulated to the air-conditioning system. This is done because it would not be economical to pipe these large volumes of refrigerant any distance.

Centrifugal refrigeration compressors are particularly well suited to direct steam-turbine drive because of their high operating speed. Water cooling equipment of one design is operated between 3500 and 4000 rpm for units developing 1000 to 2000 tons capacity, and from 7000 to 8000 rpm for units developing 100 to 200 tons capacity. However, a great many applications, particularly in the smaller sizes, are electric motor-driven and equipped with standard gear-type speed increasers. Centrifugal systems are particularly well adapted to large capacities (up to 3000 tons) although it is also possible to secure units as low as 50 tons in rating. Because centrifugal units operate best with refrigerants possessing a high specific volume, and because of the simplification of lubrication difficulties, they are frequently used for extremely low temperature applications. They are adaptable to a wide range of temperatures from —130 F to 50 F. One important advantage is their flexibility under varying loads, since units may be designed to operate with reasonable efficiency at capacities as low as 20 percent of normal load.

Control of Centrifugal Compressors

Capacity Control usually is accomplished by a controller actuated by the chilled water temperature or evaporator pressure and either (1) varying the flow of vapor from the chiller to the compressor or (2) varying the compressor speed.

1. *Vapor flow control* for compressors driven by constant-speed motors is accomplished by the use of suction dampers or inlet vanes which are positioned by automatic operators to maintain constant water temperature leaving the chiller or constant suction pressure in the evaporator.

2. *Speed Control* for compressors driven by constant-speed motors may be accomplished through the use of an eddy current clutch or a hydraulic coupling under control of an automatic operator. Variable-speed motors for speed control are more common on the larger machines. Steam turbines for driving compressors can operate at various speeds under control of an automatic valve in the governor oil circuit.

Safety Controls provide protection for the compressor by stopping the motor or reducing the capacity through the capacity-control system. Those which usually stop the motor are similar to those described for reciprocating compressors and respond to (1) low oil pressure, (2) low refrigerant temperature, (3) low chilled water temperature, (4) high condensing temperature, (5) high motor temperature, (6) high motor current, and (7) low line voltage. Frequently controls are used that are actuated by some of these conditions to reduce capacity before the limit is reached in order to prevent system shutdown.

In addition to the above safety-controls most centrifugal machines have thermostatically controlled heating and cooling of the oil, and some have water-cooled motors with the water sometimes thermostatically controlled. Usually there is a system of interlocks which prevents the compressor from starting unless the condenser water pump and the chilled water pump are running.

Condensers

Condensers used for liquefying the refrigerant are of three general types: (1) air cooled, (2) water cooled, and (3) evaporative.

1. *Air-Cooled Condensers* are readily available in unitary equipment in capacities up to 20 tons of refrigeration. They are also available as separate assemblies for field-assembled systems in capacities up to over 100 tons per condenser. Air-cooled condensers first became popular on self-contained room air conditioners and they are now used almost exclusively on this type of equipment. The use of air-cooled condensers for larger equipment was introduced only recently but is rapidly gaining in usage where an adequate water supply is costly to obtain or where other water problems are encountered.

The conventional air-cooled condenser consists of an extended-surface coil across which air is blown by a fan. The hot discharge gas enters the coil at the top and, as it is condensed, flows to a receiver located below the condenser. Air-cooled condensers should always be located in a well ventilated space so that the heated air may escape and be replaced by ambient temperature air.

When adequate supply of city or acceptable well water is readily available at a moderate cost, it is still the preferred condensing medium. However, where some form of water saving is required, the air-cooled condensers are rapidly gaining favor over the cooling tower and evaporative condenser methods in the smaller and medium tonnage classes. Where some form of water saving is required, the capacity of the equipment, under the design conditions of higher wet-bulb weather in the case of cooling towers and evaporative condensers and higher dry-bulb in the case of air-cooled condensers, must be taken into account. The air-cooled condensing equipment does have a higher peak current demand per ton over the cooling tower and evaporative condenser methods, but there is some evidence that in actual installations, it does not have any materially higher seasonal power consumption per ton than the other methods of water saving in the warm-humid and hot-humid areas. The principal advantages of the air-cooled condensers are low installation costs, lower maintenance costs and simplicity, and for these reasons they are used almost exclusively in small self-contained units and frequently in the larger sizes of compression refrigerating equipment.

2. *Water-Cooled Condensers* are commonly used with compressors of one horsepower or larger in size, and they are found almost exclusively on large installations. Although water-cooled condensers may be of many designs, the shell-and-coil and the shell-and-tube are most commonly found in present day practice.

The amount and temperature of the condensing water determine the condensing temperature and pressure, and indirectly the power required for compression. It is therefore necessary to determine a balance so that the quantity of water insures economical compressor operation.

City water is often used as the condensing medium in the smaller capacities providing an adequate water supply and means for its disposal are available, its cost is not prohibitive, and there are no restrictions against its use.

Because there is a decided trend toward restricting the use of city water for air conditioning and refrigeration purposes, cooling towers are almost universally used with water-cooled condensers in the larger sizes. Since cooling towers produce the warmest condensing water during periods of greatest load, the refrigeration equipment must be designed to meet the maximum load at abnormal condensing water temperature. This makes little difference in efficiency of operation except at those times when the condensing water temperature is highest. In many localities this occurs for only about 5 per cent of the entire cooling period and can be disregarded as a factor in establishing yearly operating costs. Further information on cooling towers is contained in Chapter 40.

3. *Evaporative Condensers* provide another means of conserving water used for condensing purposes. In effect they combine the condenser and cooling tower in a single unit.

The fan draws air over the condenser coil which is kept wet by a water spray.

Evaporative condensers are made in sizes up to 300 tons or more. They must be located where air is available and usually at not too great a distance from the compressor to avoid long refrigerant lines. Like cooling towers they result in least efficiency at periods of maximum load. Chapter 40 provides additional information on evaporative condensers.

Evaporators and Coolers

Refrigeration evaporators must be designed for efficient removal of heat from the medium being cooled, as well as effective boiling of the refrigerant and a minimum drop of pressure through the coil. There are two general types of evaporators, dry and flooded. In the dry evaporator the refrigerant enters in the liquid state, and the design provides for complete evaporation with the vapors leaving slightly superheated. In flooded evaporators not all of the refrigerant is evaporated, the liquid-vapor mixture leaving the evaporator flows into a surge drum from which the vapors are drawn into the compressor suction line, and the liquid is recirculated through the evaporator.

The types of coolers used in connection with air-conditioning work fall into three general groups: (1) direct water coolers, (2) direct air coolers, and (3) brine coolers for circulation of the brine in a closed system, and thus cooling indirectly either water or air.

1. *Water coolers*. One method of the direct cooling of water is to install direct expansion coils in the spray chamber so that the water sprayed into the air comes in direct contact with the cooling coils. Another common and efficient method of cooling spray water is to use a Baudelot type of heat absorber where the water flows over direct expansion coils at a rate sufficiently high to give efficient heat transfer from water to refrigerant.

Another type of spray-water cooler is the shell-and-tube heat exchanger in which the refrigerant is expanded into a shell enclosing the tubes through which the water flows. The velocity of the water in the tubes affects the rate of heat transfer, and as the refrigerant is in the shell completely surrounding the tubes at all times, good contact and a high rate of heat transfer are insured. The disadvantage of such a system is that with the falling off of load on the compressor, the suction temperature or the temperature in the evaporator drops, and there is a possibility of freezing the water in the tubes, which, of course, might split the tubes and allow the refrigerant to escape into the water passage. This danger can be eliminated by automatic safety devices.

Another system of cooling spray water is to submerge coils in the spray-collecting tank, or in a separate tank used for storage. The heat transmission through the walls of the coils, however, is low and a great deal more surface is required than for any other type of cooler. However, with large storage tanks this type of cooling can be utilized to advantage.

2. *Air coolers*. When direct cooling of air is employed, the refrigerant is inside the coil and the air passes over it. Cooling depends upon convection and conduction for removing the heat from the air. The type of coil used can be either smooth or finned, the finned coil being more economical in space requirement than the smooth coil. The fins, however, must be far enough apart so as not to retain the moisture which condenses out of the air.

When refrigeration evaporators are used for cooling air or other gases by forced convection, they are usually termed blast coils or unit coolers. A blast coil may be placed in a duct or in an assembled unit, and the air forced across the coil and discharged through distributing ducts or directly into the space to be conditioned. Unit coolers, designed much like unit heaters, consist of a finned coil, propeller fan, and controls suspended directly in the space to be cooled.

3. *Indirect brine coolers*. The indirect cooler, where brine is cooled by the refrigerant and the resulting cold brine is used to cool either air or water, introduces several other considerations. It is not the most economical from a power consumption standpoint, as it is necessary to cool the brine to a temperature sufficiently low so that there is an appreciable difference between the average brine temperature and that of the substance being cooled. This requires that the temperature of the refrigerant must be still lower, and consequently the amount of power required to produce a given amount of refrigeration increases due to the higher compression ratio. There are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used because of fire or other risks, especially in densely populated areas, the brine can be cooled in an isolated room or building and can then be circulated through the air-conditioning equipment. This arrangement eliminates any possibility of direct contact between the air and refrigerant.

REFRIGERATION CONTROL

Expansion Devices

In addition to means of controlling the compressor capacity and the necessary limit and safety controls as outlined under the section describing the compressors, reciprocating compressor refrigeration systems require a device for controlling the expansion of the refrigerant from the high liquid pressure to the low evaporator pressure.

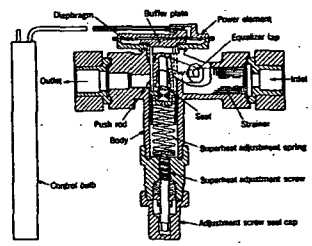


Fig. 15 .... Typical Thermostatic Expansion Valve

Automatic Expansion Valves. An automatic or pressure-controlled expansion valve operates to maintain a constant pressure in the evaporator. The liquid refrigerant passes through an orifice, the opening size of which is controlled by means of a needle valve connected to a flexible bellows. This bellows expands or contracts with variations in the evaporator pressure transmitted to the expansion chamber through the refrigerant outlet from the evaporator. The position of this needle valve is controlled by the degree of compression in an adjusted spring, balanced against the bellows, and these two forces operate to maintain a constant pressure in the evaporator by increasing or decreasing the flow of liquid refrigerant. Such an expansion valve is usually applied to evaporators of the direct-expansion type, but is not satisfactory for fluctuating loads such as are encountered in air-conditioning installations.

Thermostatic Expansion Valves. A thermostatic expansion valve controls the flow of liquid refrigerant to the evaporator so as to maintain the entire coil filled with evaporating refrigerant, and to keep a constant superheat in the refrigerant gas leaving the coil. The construction of such a valve is shown in Fig. 15 and is similar to that for an automatic expansion valve but incorporates, in addition, a power element responsive to changes in the degree of superheat of the refrigerant gas leaving the coil. This power element consists of a bellows connected by means of a capillary tube to a feeler bulb fastened to the suction line from the evaporator. The bulb, bellows, and tube are usually charged with the same liquid refrigerant used in the evaporator itself. A starved condition in the evaporator results in a greater superheat in the gas leaving the

evaporator, and this in turn operates through the power element to increase the flow of liquid refrigerant. A flooded evaporator reduces the discharge superheat, and thus tends to reduce the flow of liquid refrigerant. Such an expansion valve is satisfactory for operation with fluctuating loads since this type of control tends to keep the evaporator filled with refrigerant at all times.

Low-Side Float Valves. A liquid refrigerant control of the low-side float valve type consists of a ball float located in either a receiver or the evaporator itself on the low-pressure side of the system. A needle valve, operated through a simple lever mechanism attached to the float, permits the passage of more or less refrigerant, as the level in the receiver or the evaporator fluctuates. Such a control must be used in conjunction with a flooded evaporator, and has been applied extensively to household refrigerators and, to some extent, in commercial and industrial installations.

High-Side Float Valves. A high-side float valve differs from a low-side float valve in that the float is located in a receiver or container on the high-pressure side of the system. Proper operation again depends upon metering of the refrigerant through a controlled opening, depending upon the level of the liquid refrigerant in the container. Such a control has the disadvantage that the evaporator must be placed directly adjacent to the float container, or some intermediate pressure device must be applied to prevent flashing of the refrigerant upon pressure drop.

Capillary Tubes. A capillary tube may be used as a liquid refrigerant expanding device. Such a device consists of an extremely small bore tube (in the order of 0.04 inch in diameter) of five to twenty feet in length. Although such a restricting device operates as a very simple means of expanding the liquid refrigerant, it has the disadvantage that no modifications are possible to adjust the rate of expansion under various operating conditions. The bore and length of the tube, as well as the proportions of the rest of the system, are critical. It is for these reasons that its application has been limited to factory-assembled domestic and commercial units.

Refrigerant Flow Control

The capacity of the refrigeration system must be controlled in accordance with the load imposed on the system. Except for those cases where the compressor is started and stopped by a thermostat responding to load conditions, some form of control of the refrigerant flow is usually required. Controlling the flow of refrigerant in accordance with the load may be accomplished in a number of ways, some of which are:

1. *Solenoid valves* are frequently used for control of gas or liquid flow. A solenoid valve is placed in the liquid line ahead of the expansion valve. It is closed whenever the compressor is not in operation and thus leakage into the evaporator is prevented. In most cases it is controlled by a thermostat responding to the load. Solenoid liquid valves are widely used for control of refrigerant flow to individual evaporators in a multiple evaporator system operated by one compressor. The compressor capacity is controlled separately by a controller responding to suction pressure. Solenoid liquid and suction valves are sometimes used to isolate an evaporator completely for defrosting purposes.

When solenoid liquid valves are used for controlling refrigerant flow, a *pump-down cycle* of control is often employed. The thermostat closes the solenoid liquid valve and the compressor then pumps down until stopped by the low-suction pressure switch. The compressor cannot restart until the thermostat again opens the solenoid liquid valve.

2. *Expansion Valve Modulation.* This form of flow control utilizes the throttling characteristics of the expansion valve to starve the coil under reduced-load conditions. This may be accomplished by a device attached to the expansion valve or a pilot control which varies the setting of the expansion valve by changing the pressure in the equalizer line. Either of these are actuated by a controller sensitive to load conditions and the compressor capacity control is from suction pressure. Since short-cycling of the compressor is apt to occur if the flow of refrigerant is reduced too much, some limit to prevent this is required.

3. *Evaporator Pressure Control* is accomplished by back-pressure valves. These are placed in the suction line between the evaporator and the compressor, and maintain the evaporator pressure constant by regulating the amount of vapor drawn from the evaporator. The compressor capacity is controlled from suction pressure at the compressor. The use of back pressure valves permits a lower pressure at the compressor than would be permissible at the evaporator, thus minimizing short-cycling. On a multiple-evaporator system served by one compressor, the evaporators may be maintained at different temperatures. Back pressure valves can be equipped with an air connection for automatically varying the evaporator pressure in accordance with the demands of a pneumatic controller. Means for mechanically readjusting the valves by use of a proportioning electric operator also are available.

Condenser Control

The majority of refrigeration systems, other than fractional horsepower, use water-cooled rather than air-cooled condensers. Condenser control is used for the purpose of conserving water or limiting the condenser pressure, or both. Solenoid water valves on small compressors, or pneumatic or electric valves on larger compressors, are usually controlled simultaneously with starting or stopping of the compressor. A pressure-operated valve will provide proportional control of water flow in response to condenser pressure. Similar water valves controlled thermostatically by the temperature of water discharged from the condenser are sometimes used. Evaporative condensers may be controlled by an automatic damper responding to a condenser-pressure controller. Cooling tower fans sometimes are started and stopped by a thermostat in the condensing water or a pressure controller in the condenser. Some form of freeze protection for evaporative condensers and cooling towers is necessary if they are required to operate in cold weather.

Refrigeration Control for Air Conditioning

When refrigerating equipment is used for space cooling, two major control problems exist: one is control of the temperature and the other, control of the humidity. In some applications the amount of latent heat to be removed is small compared with the sensible heat. In such cases, sufficient dehumidification will usually occur without any special provisions. In other cases, such as theaters, where the latent load is relatively high, the air must be cooled below its dew-point temperature, and sometimes rewarmed to return it to the comfort range. Refer to Chapter 43 for general information on the subject of controls and for applications relating to air-conditioning systems.

REFRIGERATION PIPING

The pressure drop which occurs during passage of the refrigerant through connecting piping is similar in effect to that which occurs through suction and discharge valves of the compressor. Thus, the effect of the pressure drop in the suction line between evaporator and compressor requires that a lower pressure be maintained inside the compressor during suction than is maintained in the evaporator. The pressure drop through the connecting piping between the compressor and condenser requires that a higher pressure be maintained inside the compressor during discharge than in the condenser. These losses result in a greater compression ratio, and therefore greater power requirements, as well as a lower volumetric efficiency and higher displacement requirements. Pressure losses in the liquid line between condenser or receiver and the expansion valve may result in some flashing of the liquid refrigerant, unless the liquid is subcooled. In all cases, friction losses should be kept to a minimum, and piping should be selected which will give the smallest loss consistent with overall economy in the system.

Table 5 .... Dichlorodifluoromethane* (CCl_2F_2) Liquid Lines, Tons Capacity per 100 Ft Equivalent Length

Line Size, Inches	Pressure Drop per 100 Ft Equivalent Length, Psi			
	3	5	10	20
⅜ OD	0.88	1.14	1.80	2.58
½ OD	2.89	3.64	5.56	8.50
½ IPS	4.86	6.81	10.2	15.8
⅝ OD	4.86	6.81	10.2	15.8
¾ IPS	9.73	12.6	18.5	27.0
¾ IPS	9.73	12.6	18.5	27.0
⅞ OD	10.5	14.1	21.8	33.0
1 IPS	21.4	28.2	41.3	60.8
1⅛ OD	21.4	28.2	41.3	60.8
1¼ IPS	36.9	48.1	70.5	101.
1⅜ OD	36.9	48.1	70.5	101.
1½ IPS	62.0	80.2	114.	160.
1⅝ OD	62.0	80.2	114.	160.
2 IPS	124.	161.	231.	328.
2½ IPS	230.	297.	426.	607.
3 IPS	364.	469.	676.	972.
3½ IPS	539.	704.	1005.	1430.
4 IPS	753.	972.	1385.	1945.

Note: Tonnage values above those underlined give velocities of 300 fpm or less.
*Refrigerant 12.

Refrigerant liquid lines from the receiver to the expansion valve should preferably be designed with a pressure drop of less than 5 psi, and with 10 psi as the maximum. A velocity of 100 to 250 fpm is recommended to prevent a pressure drop great enough to cause vaporization of the refrigerant ahead of the expansion valve. If the evaporator is to be located at a higher elevation than the condenser or receiver, account should be taken of the pressure drop for each foot of static liquid lift. Approximate values are 0.26 psi per foot for ammonia, 0.57 psi per foot for dichlorodifluoromethane, 0.51 psi per foot for monochlorodifluoromethane, and 0.64 psi per foot for monofluorotrichloromethane. Where there is a possibility of vaporization of some of the liquid before reaching the expansion valves, means for subcooling should be provided.

Since a reduction of suction pressure at the compressor results in an appreciable reduction in capacity and more power input per ton of refrigeration, great care should be given to the proper sizing of suction lines between the evaporator and the compressor. Although comparatively high velocities, 500 to 5000 fpm, may be used, the optimum value will depend upon the refrigerant and the operating-pressure range. Since return of the oil to the compressor must be considered in the case of the fluorinated hydrocarbons and methyl chloride, for these refrigerants the minimum velocity should be 500 fpm for horizontal runs and 1000 fpm for vertical runs. For the former, the usual design velocities range between 1000 and 2000 fpm. Too high velocities create noise problems and excessive pressure drops. The total pressure drop in the suction line should be between one and two psi, if the velocity can be kept within the specified limits.

Table 6 .... Maximum Tons of Compressor Capacity for CCl_2F_2 Lines*

(Only for temperatures indicated)

Line Size, Inches	Suction Lines Based on 105 F Condensing Temperature: Psi Pressure Drop per 100 Ft Equivalent Length at 40 F Saturation						Discharge Lines: Condensing Temperature	
	¼	1	2	3	4	5	115 F	90 F
½ OD	0.14	0.20	0.28	0.35	0.41	0.45		
⅝ IPS	0.17	0.24	0.34	0.42	0.49	0.54		
⅝ OD	0.25	0.35	0.51	0.62	0.73	0.81	1.43	1.15
½ IPS	0.35	0.45	0.65	0.79	0.93	1.03	1.87	1.50
⅞ OD	0.55	0.76	1.10	1.34	1.58	1.75	2.97	2.38
¾ IPS	0.68	0.94	1.35	1.65	1.92	2.12	3.26	2.62
1⅛ OD	1.26	1.80	2.57	3.17	3.76	4.15	5.05	4.05
1 IPS	1.43	2.01	2.89	3.54	4.17	4.60	5.29	4.25
1⅜ OD	2.21	3.12	4.45	5.50	6.38	7.05	7.72	6.19
1¼ IPS	2.70	3.82	5.37	6.72	7.68	8.48	9.16	7.35
1⅝ OD	3.40	4.78	6.79	8.42	9.77	10.8	10.92	8.75
1½ IPS	4.05	5.75	8.10	10.12	11.6	12.8	12.5	10.0
2⅛ OD	6.12	8.60	12.1	15.1	17.4	19.2	19.2	15.3
2 IPS	7.66	10.9	15.3	19.2	32.2	24.5	20.6	16.5
2⅝ OD	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9
2½ IPS	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9
3⅛ OD	19.1	27.2	38.2	47.8	55.0	60.7	51.5	39.8
3 IPS	20.9	29.4	42.3	51.8	60.0	66.2	54.5	43.8
3⅝ OD	27.8	39.7	55.7	69.8	80.3	88.7	72.0	57.6
3½ IPS	30.2	43.2	61.0	76.1	87.0	96.0	78.8	63.3
4⅛ OD	38.6	55.2	78.0	97.3	111	123	95.8	77.1
4 IPS	40.7	58.6	83.0	103	118	130	101.6	81.6
5 IPS	71.3	100	141	176	203	224	171.5	137.8
6 IPS	126	183	257	322	366	403	266	214
8 IPS	211	297	422	523	602	664	461	370
10 IPS	352	503	712	887	1024	1130	725	582
12 IPS	550	780	1106	1373	1582	1748	1041	836

* Refrigerant 12.

Compressor discharge or hot gas lines may be designed with velocities from 1000 to 5000 fpm, except for dense gases such as carbon dioxide, where noise considerations will reduce the upper limit. A pressure drop of 2 to 4 psi is recommended for the discharge lines. Extensive tables are available in the literature for the determination of pressure drops through refrigerant lines with various refrigerants. The capacities listed in Tables 5, 6, and 7 are published in *ARI Equipment Standards* 520 (1946), of the Air-Conditioning and Refrigeration Institute, and are used by permission. Table 5 shows the tonnage capacity normally allowed for CCl_2F_2 liquid lines per foot equivalent length of pipe, and Table 6, the maximum tonnage for suction

Table 7 .... Approximate Suction-line Capacity Factors for Equal Pressure Drop of CCl_2F_2 (Refrigerant 12)

Saturated Suction Temperature, F.	50	40	30	20	10	0	−10	−20
Factor	1.09	1.00	0.92	0.86	0.80	0.74	0.66	0.56

and discharge CCl_2F_2 lines. Table 7 presents suction-line capacity factors for equal pressure drop.

ACCESSORIES

Dehydrators, oil separators, strainers, vibration eliminators, sight glasses, and various types of valves are accessories frequently needed for the proper installation and operation of refrigeration systems. Refrigerant-line dehydrators or dryers usually consist of copper containers fitted with tubing connections at either end, and contain a desiccant such as silica gel, activated alumina, or calcium chloride. The liquid refrigerant is circulated through the dryer during operation of the system, and the moisture content of the refrigerant charge is thus kept to a minimum.

Oil separators are installed between the compressor and condenser to prevent excessive oil removal from the compressor crankcase and its passage into the condenser and evaporator. The oil is separated from the gaseous refrigerant by gravity during its passage through a chamber of sufficient size to reduce the velocity. A float-operated valve maintains a maximum oil level in the separator, and additional oil is forced by pressure difference through a line back to the crankcase.

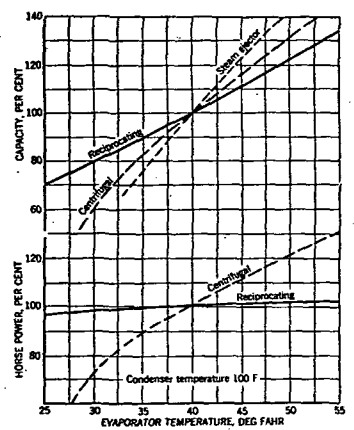


Fig. 16 .... Performance Characteristics of Compression Refrigeration Machines at Constant Speed

Screen strainers are frequently installed in the liquid line piping before solenoid valves and expansion valves, as well as before regulating valves in water lines leading to water cooled condensers. Sight glasses that permit visual inspection of the condition of the refrigerant are sometimes installed on factory-assembled commercial unit systems. It is particularly advisable to place such a fitting before the expansion valve, if the evaporator is located above the condenser.

Flexible vibration eliminators, usually consisting of a bellows design covered with woven copper wire, are sometimes installed in copper lines where units such as compressors are installed on flexible mountings, or where vibration is otherwise a problem. Packed or packless shut-off valves are necessary where it may be required to isolate portions of a system.

EQUIPMENT CHARACTERISTICS AND SELECTION

The various types of compression systems have quite different characteristics of capacity and power with varying evaporator and condenser temperatures, as may be noted from curves in Figs. 16 and 17.

From Fig. 16 it may be observed that power requirements for the centrifugal compressor increase much more rapidly than for the reciprocating compressor, with increase in evaporator temperature. Similarly, the capacities of the steam ejector and centrifugal compressors increase more rapidly than those of the reciprocating compressor with increase in evaporator temperature. Thus, both the steam-jet and centrifugal machines tend to be more self-regulating than the reciprocating. It is also evident from Fig. 16 that the steam-jet equipment is best suited for operation at high evaporator temperatures.

The effect of condenser temperature upon the power and capacity of the different types of compressors is shown in Fig. 17. It may be noted that the power required by the

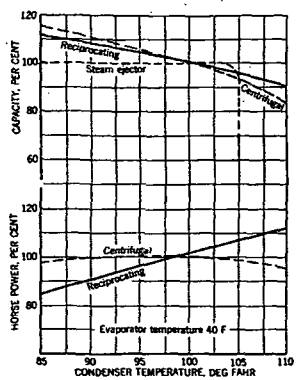


Fig. 17 .... Performance Characteristics of Compression Refrigeration Machines at Constant Speed

Table 8 .... Basis of Equipment Selection

Capacity Tons	Majority Used	Some Used	Few Used
0 to 10	Unit systems in conditioned space.	Unit central systems using duct distribution.	Built-up central systems.
10 to 25	Unit central systems using duct distribution.	Unit systems in conditioned space.	Built-up central systems using reciprocating compressors, adsorption, and absorption systems.
25 to 100	Built-up central systems using reciprocating compressors.	Unit central systems using duct distribution.	Central systems using adsorption systems or centrifugal refrigeration.
100 to 200	Built-up central systems using reciprocating compressors.	Built-up central systems using absorption and centrifugal compressors.	Built-up central systems using steam jet.
200 and Over	Built-up central systems using centrifugal compressors.	Built-up central systems using steam jet or absorption systems.	Built-up central systems using reciprocating compressors.

reciprocating compressor increases rapidly with increase in condenser temperature, while the power curve for the centrifugal compressor is relatively flat. It is also evident that the capacity of the steam-jet compressor is independent of condenser temperature until a certain point is reached, where it drops to zero. As previously stated, steam-jet equipment requires more condensing water than other types of compression systems. Consequently, steam-jet systems are well suited to those applications where condensing water is cheap, or where condensing water is rather high in temperature.

The selection of proper refrigeration equipment for any air-conditioning job is of utmost importance for satisfactory results. The most important factors in the selection of the equipment are:

1. Loads (as determined by the conditions of the space to be cooled).
2. Economics (both initial and operating costs).
3. Codes (local safety codes must be adhered to and influence the type of system to be used).

A broad division of equipment to be used for a particular installation or application may be made on the basis of the magnitude of the load. Current general practice is outlined in Table 8.

Unit or *packaged* systems, consisting of a reciprocating compressor, condenser, evaporator, and fans, are generally used in the smaller sized jobs where electric power is available, as they are manufactured complete, ready to install, and are the most economical (See Chapter 16).

The reciprocating compressor in the built-up central system (see Chapter 19) covers the widest range of application since it is applicable to either the direct-expansion or indirect systems, and can be driven by steam or gas engines, or by electric motors. The quantity of condensing cooling medium required is also less than for any other system, with the exception of the centrifugal compressor, which uses the same amount.

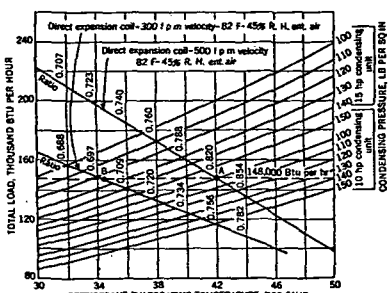


Fig. 18 .... Compressor and Coil Performance

Table 9 .... Typical Operating Conditions for Two Types of Load

Type of Enclosure	Load Btu per Hr.	Ratio Sensible to Total Heat	Air Entering Coil		Operating Balance Point		
			Temp. F	Rel. Hum. %	Evaporator Temp. F	Condenser Pressure Psi	Sensible Heat %
Restaurant							
Sensible	103,000						
Latent	45,000						
Total	148,000	0.695	82	45	34.4	123	69.9
Office							
Sensible	121,000						
Latent	27,000						
Total	148,000	0.820	82	45	42.2	100	82.1

Centrifugal compressors are used for large installations, and usually where the indirect system is required. The driving mechanism can be a steam turbine or electric motor. The steam-jet system is used where steam is available and cooling water can be had in large quantities.

It will be noted by referring to Fig. 16 that all systems using compressors have a common characteristic, namely, that the capacity varies with the evaporating temperature. Not only can the equipment be selected to produce a given result, but the performance can be predicted under varying load conditions by the simple expedient of using the variable of evaporating temperature as the abscissa, and the load or capacity as the ordinate in a series of curves.

Manufacturers of compressors and cooling coils furnish performance data for apparatus that can be plotted in the form of curves similar to those shown in Fig. 18. The performance of a compressor is plotted as a series of curves, each curve being drawn for a given condensing pressure. The performance of a direct-expansion coil at two different air velocities is plotted on the same graph. The operating point will be, of course, where the two curves cross.

Data given in Table 9 illustrate two types of conditioned enclosures having the same total load of 148,000 Btu per hour, but with two different ratios of sensible to total heat. In the case of the office with a ratio of 82 percent sensible to total heat, the operating point *A* in Fig. 18 is found to be 42.2 F evaporating temperature, with a face velocity of 500 fpm. In the case of the restaurant, with a ratio of 69.5 percent sensible to total heat, the air velocity is lowered to 300 fpm, and the evaporating temperature is lowered to 34.4 F as shown in point *B* of Fig. 18. In order to obtain the same capacity, a larger condensing unit is used. This illustration assumes zero pressure drop through the suction line. The pressure drop can be taken into account by shifting the compressor performance curves by the amount of pressure drop expressed in Fahrenheit degrees.

REFERENCES

[1] P. C. Scofield: Air cycle refrigeration (*Refrigerating Engineering*, Vol. 57, June 1949, p. 558).

[2] A. R. Mumford and A. A. Markson: Application and economy of steam jet refrigeration to air conditioning (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 33).

[3] A. A. Berestneff: A new development in absorption refrigeration (*Refrigerating Engineering*, Vol. 57, June 1949, p. 553).

BIBLIOGRAPHY

Refrigerating Data Book, Vol. 1 (American Society of Refrigerating Engineers).

H. J. Macintire: *Refrigeration Engineering* (John Wiley & Sons, New York, 1937).

N. R. Sparks: *Theory of Mechanical Refrigeration* (McGraw-Hill Book Co., New York).

B. F. Raber and F. W. Hutchinson: *Refrigeration and Air Conditioning Engineering* (John Wiley & Sons, New York, 1945).

J. A. Moyer and R. U. Fitts: *Refrigeration* (McGraw-Hill Book Co., New York).

W. R. Hainsworth: Refrigerants and absorbents (*Refrigerating Engineering*, August, September 1944).

B. H. Jennings and S. R. Lewis: *Air Conditioning and Refrigeration* (International Textbook Company, Scranton, 1944).

Jordan and Priester: *Refrigeration and Air Conditioning* (Prentice-Hall, Inc., New York, 1948).

CHAPTER 39

THE HEAT PUMP

History and Development; Basic Circuits; Fundamentals; Types; Heat Sources and Sinks; Performance Characteristics; Temperature Levels; System Balance; Seasonal Performance; Annual Operating Cost; Components; Application; Criteria for Feasibility; Design and Selection; Systems for Larger Buildings; Sensible, Latent, and Low-Side Heat Storage

THE term *heat pump* as applied to a year-round air-conditioning system, commonly denotes a system in which refrigeration equipment is used in such manner that heat is taken from a *heat source* and given up to the conditioned space when heating service is wanted, and is removed from the space and discharged to a *heat sink* when cooling and dehumidification are desired. The thermal cycle is identical with that of ordinary refrigeration, but the application is concerned alike with the cooling effect produced at the evaporator and with the heating effect produced at the condenser. In some applications, both the heating and the cooling effects obtained in the cycle are employed simultaneously.

The heat-pump principle is also applied for purposes other than air conditioning, such as: supply of domestic hot water in residences and commercial buildings, recovery of low-temperature heat as a useful byproduct in industrial operations, and disposal of heat from processes involving evaporation of water or other fluids at depressed temperature and pressure. Defrosting of evaporator coils in refrigerating systems by intermittently admitting hot gas directly from the compressor is another application of the heat-pump method. Descriptions and diagrams of various heat-pump systems and criteria for feasibility are given later in this chapter.

HISTORY AND DEVELOPMENT

Over a century ago, William Thomson (Lord Kelvin)[1] suggested the use of a compressor as a *warming engine* to heat buildings, as an alternate for combustion of fuels. Somewhat prior to 1910 Dr. S. Z. Ferranti, Prof. J. Perry and Mr. T. V. Morley, in England, strongly advocated the undertaking of development in the design and application of heat pumps, but the economic outlook, then unfavorable, deferred further investigation. It was not until a paper on the principles and economics of heat pumps, prepared in October 1928 by T. G. N. Haldane, was presented before the British Institute of Electrical Engineers in December 1929,[2] that development was started in the United States. Haldane reported the trial results from early installations made in Scotland. Application for water heating was then as prominent abroad as use for space heating.

Dr. A. R. Stevenson, Jr. was an early advocate in this country, having first presented the possibility to the Refrigeration and Gas Committee of the American Gas Association in April 1926, and later to the Franklin Institute.[3] The concept held in this country today, that heat pumps for the large majority of space-heating applications should be regarded essentially as *year-round* air-conditioning devices, did not become generally accepted until about 1950.

History in the United States

Pioneering work on heat-pump development in the United States was conducted through the early 1930's mainly under sponsorship of a few investor-owned electric utility companies, with the objective of promoting wider use for their service. The seasonal character of electric load from summer air conditioning, after a record of steady growth, was then already making evident its likelihood of creating new peak demands on utility systems. This prompted the investigation of methods whereby such load could be made more favorable both to the customer and to the electric company.

The first major installation in the United States was made about 1931, for a new 3,800,000 cu ft office building of Southern California Edison Company in Los Angeles. Refrigeration equipment of the already existing air-conditioning system was modified to permit heat-pump operation with outdoor air in a conventional cooling tower serving as heat source. An installation made in Salem, New Jersey[4] in 1934 was the first commercial heat pump in this country designed for complete heating and cooling. A number of buildings occupied by subsidiaries of American Gas and Electric Company in Ohio, West Virginia, Virginia, and Kentucky were equipped during the middle and late 1930's. The home office of Louisiana Power and Light Company at Algiers, Louisiana was equipped in 1936, using purchased city water as source of heat. An office building[5] of United Illuminating Company at New Haven, Connecticut, erected in 1939, was equipped with a water-source heat pump. It utilized as a standby heating system a large water storage tank equipped with immersion electric heating elements. Throughout this early period all installations were custom built, employing standard equipment components modified where necessary to suit the dual-purpose functions. After World War II the use of heat pumps increased, particularly with the introduction of unitary equipment, in which the refrigeration system, fans, pumps, and the necessary automatic controls were assembled within a single housing.

In recognition of the status attained by heat pumps and their growing engineering importance, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS appointed a Technical Advisory Committee on the Heat Pump in 1947. An independent group from the electric utility industry, known as Joint AEIC-EEI Heat Pump Committee, was founded in February 1947. This body has worked closely with equipment manufacturers, giving advice on matters related to suitability of equipment types and control methods, to experience resulting from trial installations, and especially to the characteristics of electric loads. It has also under-

taken research projects in allied areas such as heat sources, utilization of solar energy as a supplementary heat source, and thermal storage methods.

The Joint Committee has lent encouragement to publication of two textbooks[6, 7] on the technology of heat pumps, and to the maintenance of a comprehensive bibliography,[8] a reference library, and installation statistics. Additional bibliographies and installation studies have been made by The Heat Pump Steering Committee of the Southeastern Electric Exchange.[9]

The summer air-conditioning load has high electric power demand and low annual load factor, and in addition, the highest demand peaks will occur on the relatively few extremely hot days during summer months, when the load-carrying capacity of the whole utility system (generation, transmission and distribution) is considerably lower than on cold days in winter. For example, the average annual load factor on a noncoincidental basis for summer air conditioning can be expected to remain between limits of 6 and 15 percent. This is far below the 64 percent present-day annual *load factor for the total electric load* of utility systems. Consequently, the large utility companies are giving support to the advancement of heat-pump technology as a means of broadening the field of application of the heat pump with its beneficial effect on the electric system load factor.[10]

Present Status

Large central heat pumps of modern design, within the capacity range of about 100 to approximately 1000 horsepower of compressor-motor rating, are now operating in a substantial number of buildings. Among noteworthy installations are those in the Oregonian Building and Equitable Building, both in Portland, Oregon; Appalachian Electric Building in Roanoke, Virginia; Columbus and Southern Ohio Electric Building in Columbus, Ohio. Detailed information on their design and performance has been presented in ASHAE papers[11, 12, 13] and in technical magazines.[14]

In the Southdale Shopping Center in Minneapolis, Minn., a 700-ton heat pump installation with its associated deep wells providing the heat sink and heat source, has a coefficient of performance on the heating cycle of approximately 4. Internal combustion engine driven heat pumps in this installation have been found to provide additional heating capacity because of the recoverable heat in the engine cooling water and the muffler.

Compressor types employed in large central systems vary from one large centrifugal unit to as many as eight multicylinder reciprocating units. A single or central system is generally used throughout the building, but in some instances the total capacity is divided among several separate heat-pump systems to facilitate zoning. Both well water and air are used as heat sources. Compression is accomplished in two stages for a few recent projects. Frequently provision is made for heating and cooling service to be supplied simultaneously to separate zones of the building.

Unitary heat pumps, available from or under development by many manufacturers, account for the large majority of residential and small commercial installations. Outdoor air is the heat source in the majority of installations, but models are built also for well water. Capacities cover the range from 2 to about 25 horsepower. Both hermetic and open-type compressors are used. The entire heat-pump circuit may be charged with refrigerant and sealed at the factory, or individual factory-sealed subassemblies may be connected together in the field. Some models are arranged for remote location of the outdoor-air heat exchanger that serves alternately as evaporator and as condenser. In appearance and dimensions, the casings of unit heat pumps closely resemble those of conventional air-conditioning units having equal capacity.

The normal design basis is to select *the size of compressor* and motor for the summer cooling and dehumidification capacity requirement. For heating service with outdoor air as the heat source, supplementary electric resistance heating elements are normally provided for use during such periods of low temperature as may be required by the climate and heating load of the structure. These heaters are sometimes objectionable to the utility because the resulting seasonal electric consumption is low in comparison with the electric input demands they create on the electric service facilities. Consequently, it is important that the application engineer for such installations give careful attention to minimizing the installed capacity of heating elements, as discussed later in this chapter.

With unitary systems, defrosting of the evaporator coils at suitable intervals with minimum interruption of heat supply to the building is usually accomplished automatically by admitting hot refrigerant gas taken directly from the compressor discharge.

Minimum industry performance standards of unitary heat pumps are covered by *ARI Standard 240-57, Unitary Heat Pump Equipment*, issued 1957 by Air-Conditioning and Refrigeration Institute. It applies to factory-made heat pumps and to matched assemblies as defined in the Standard, for residential, commercial and industrial service, but is not applicable to field-modified cooling units converted to heat-pump operation, nor to room air conditioners. The Standard provides that both heating and cooling capacity be expressed in Btu per hour under specific conditions, and includes minimum performance standards such as the ability to operate under maximum load conditions or adverse power and temperature conditions, and describes safety codes to which such equipment should be designed. It also outlines several performance requirements that are considered good practice in the design of either unitary heat pumps or built-up systems.

Window-type, or in-the-wall type heat pumps, now in factory production in ½ to 2 hp size range, resemble conventional units for summer air conditioning, similarly installed. Heretofore, capacity of these units on heating service has generally been less than needed in extremely cold weather to maintain the required temperature within the space served adequately by the same unit in summer. Except in regions having mild climate, they were useful principally for intermediate-season operation or to supplement the main heating system of the building. Some recently introduced models, however, have characteristics more suited to complete year-round operation.

As a result of more than a decade of experience gained in product development, factory production, field operation, and maintenance, heat pumps have become firmly established throughout a large part of the country. Moreover, heat pumps are being adopted for entire housing developments. As many as 1500 units have been installed for a single project.

The estimated number of central system heat-pump units operating at the end of 1958 throughout the United States is approximately 40,000. Installations have been made in practically every state, with the southeastern, southwestern, and west coast states predominating. It is probable that at least two-thirds of total units take heat from outdoor air, and, of the remainder, nearly all use water, with ground coils almost negligible among the more recent installations. Aggregate connected load may well reach 400,000 kilowatts. Approximately 150 domestic water heaters, mostly experimental or pilot-type models, are in operation.

BASIC CIRCUITS

Fundamentals

Since, from the refrigeration standpoint, a heat pump is similar to a conventional refrigeration system, its basic circuit may be represented by Fig. 3 in Chapter 38. Changeover between heating and cooling services may be accomplished by: (*a*) actuating valves in the refrigerant lines, so as to interchange the positions of heat exchangers constituting the evaporator and the condenser, respectively, in the refrigerant flow circuit, or (*b*) by switching the paths of air, water, or other fluid that convey heat from source to evaporator and from condenser to sink, respectively. The interchange function inherent in heat-pump control led to use of the term *reversed-cycle refrigeration* during the early development period of the heat pump, but this inaccurate name is now obsolete.

The operating principle of the heat pump is identical with that of the heat-power thermodynamic cycle governing the conversion between heat energy and mechanical work. It is derived from the Second Law of Thermodynamics (see Definitions, Chapter 1). The operating efficiency, or coefficient of performance (CP), of an elementary perfect refrigeration system for both cooling and heating effects is given by *Equations 2 and 3, Chapter 38*.

In actual systems, the coefficients of performance are defined in the following terms:

The *heating coefficient of performance*, $(CP)_h$, of an installed heat pump may be defined as the ratio of the total *instantaneous* useful heating effect produced by the heat-pump system at stated conditions, to the heat equivalent of the total energy input rate required to drive or operate the system. If total energy input of all auxiliaries such as fans and pumps is not included, it should be so stated.

The *cooling coefficient of performance*, $(CP)_c$, of an installed heat pump may be defined as the ratio of the *instantaneous* useful refrigeration effect produced by the heat-pump system at stated conditions to the heat equivalent of the total energy input rate required to drive or operate the system.

The term *performance factor*, (PF), is similar to coefficient of performance, but is used when referring to values based on an *extended period* of time, such as a day, month, or season. The period of time covered should be given when using this term. If supplemental heat is involved, its effect should also be specified.

Heat Pump Types

Heat pumps for air-conditioning service may be classified according to (*a*) type of heat source and sink, (*b*) heating and cooling distribution fluid, (*c*) type of thermodynamic cycle, (*d*) type of building structure, and (*e*) size and configuration. The more common types are shown in Table 1.

The *air-to-air* type is the most common type of system. It is particularly suitable for factory-built unitary heat pumps and has received favorable acceptance for residential and commercial applications. The diagram shown in Table 1 is typical of the refrigeration circuit employed. In smaller unitary heat pumps, it is common to replace the expansion and check valves with a capillary tube refrigerant control. A few installations have been made in which the forced-convection indoor heat-transfer surface has been replaced by a radiant panel.

In air-to-air heat-pump systems, as shown in second diagram of Table 1, the air circuits may be interchanged by means of dampers (motor-driven or operated manually) to obtain either heated or cooled air for the conditioned space. With this system one heat-exchanger coil is always the evaporator while the other is always the condenser. The conditioned air will pass over the evaporator during the cooling cycle while the outdoor air will pass over the condenser. The change from cooling to heating is accomplished by positioning the dampers which may be motor driven or operated manually.

A *water-to-air* heat pump uses water as a heat source and sink and uses air to transmit heat to or from the conditioned space.

Air-to-water heat pumps are commonly used in large buildings where zone control is necessary, and are also sometimes employed for the production of hot or cold water in industrial applications.

Earth-to-air heat pumps may employ direct expansion of the refrigerant in an embedded coil as illustrated in Table 1, or they may be of the indirect type described above under the water-to-air type.

A *water-to-water* heat pump uses water as the heat source and sink for both cooling and heating operation. Heating-cooling changeover may be accomplished in the refrigerant circuit, but in many cases, it is more convenient to perform the switching in the water circuits such as is illustrated in Table 1.

An *earth-to-water* heat pump (not shown in Table 1) may be like the earth-to-air type shown except for the substitution of a refrigerant-water heat exchanger for the finned coil shown on the indoor side. It may also take a form similar to the water-to-water system shown when a secondary-fluid ground coil is used.

Some heat pumps which use earth as the heat source and sink are essentially of the water-to-air type. An antifreeze solution is pumped through a loop comprised of a pipe coil embedded in the earth and the chiller-condenser.

Many variations are possible in the refrigeration circuit and in the heat-source and sink arrangements. Some of these are described in later sections of this chapter.

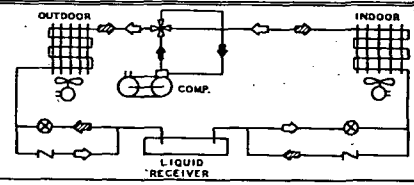
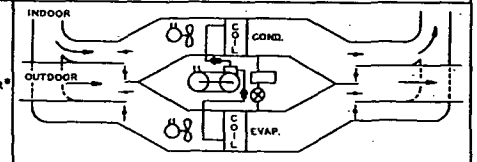
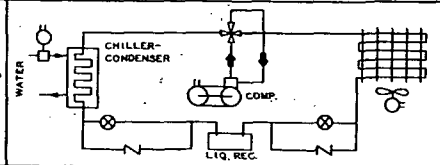
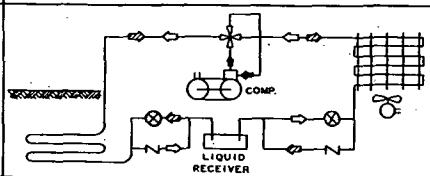
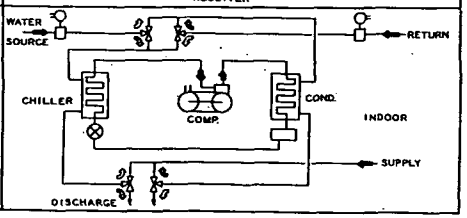
Other types of heat pumps in addition to those listed in Table 1 are possible. An example is one which utilizes solar energy as a source of heat; its refrigerant circuit may resemble the water-to-air, air-to-air, or other types depending on the form of solar collector and the means of heating and cooling distribution which is employed.

Another variation is the use of more than one heat source. Some heat pumps have utilized air as the primary heat source, but are changed over to extract heat from water (e.g. from a well or storage tank) during periods of peak load. The use of solar energy requires another heat source during periods of insufficient solar radiation.

Industrial and Agricultural Heat Pumps

While the principal use of heat pumps may always be in space heating and cooling, the heat-pump principle also has applications in the industrial and agricultural fields, particu-

Table 1 . . . . Common Heat Pump Types

HEAT SOURCE AND SINK	DISTR. FLUID	THERMAL CYCLE	DIAGRAM (▨ HEATING ⇨ COOLING ➡ HEATING AND COOLING)
AIR	AIR	REFRIGERANT* CHANGEOVER	
AIR	AIR	AIR CHANGEOVER*	
WATER	AIR	REFRIGERANT* CHANGEOVER	
AIR	WATER		
EARTH	AIR	REFRIGERANT* CHANGEOVER	
WATER	WATER	WATER CHANGEOVER*	

* ALL SINGLE STAGE COMPRESSION

larly in the process industries where large quantities of low temperature heat may be required. The heat-pump cycle, in many instances, can be effective in the utilization of waste heat.

In a number of processes, low-temperature heating and moderate refrigeration are required simultaneously. The concentration of fruit juices may be accomplished by a heat-pump system.

Fig. 1 illustrates the application of the heat-pump principle for drying. A large portion of the heat given off from the condenser while heating the stream of dry air is recovered by the evaporator from both the sensible and latent heat of the moisture-laden air downstream from the material being dried. This type of heat pump is expected to find particular application in the drying of tobacco and other agricultural products.

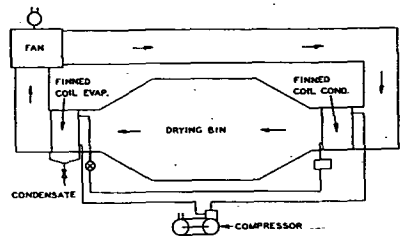


Fig. 1 . . . . Application of Heat Pump to Drying

Other Refrigeration Cycles

Any thermodynamic cycle that is capable of producing a cooling effect may theoretically be used as a heat pump. Other than the ordinary vapor-compression cycle, possible cycles include (a) the heat-operated absorption cycle, (b) the ejector cycle, (c) gas cycles, both open and closed[8] and (d) the thermoelectric cycle.[9]

None of these cycles as heat pumps is currently practicable due to limitations in efficiency, cost or size.

HEAT SOURCES AND SINKS

Table 2 shows the principal media being used with heat pumps as a heat source for heating and as a heat sink for cooling. The most practical choice for a particular application will be influenced primarily by geographic location, climatic conditions, initial cost, availability, and type of structure. Various factors to be considered for each source are given in Table 2. A more detailed discussion of design and selection factors for each source and sink follows.

Air

Outdoor air offers a universal heat-source, heat-sink medium for the heat pump. Extended-surface, forced-convection heat-transfer coils are normally employed to transfer the heat between the air and the refrigerant. Typically these surfaces are 50 to 100 percent larger than the corresponding surface on the indoor side of heat pumps using air as the distributive medium. The volume of outdoor air handled is also usually greater in about the same proportions. The temperature difference during heating operation between the outdoor air and the evaporating refrigerant generally is in the range of from 10 to 25 F deg.

The performance of heating and cooling coils for air is given in more detail in Chapter 23.

When selecting or designing an air-source heat pump, two factors particularly must be taken into consideration; (1) the variation in temperature experienced in a given locality and (2) the formation of frost.

The determination of the heating design temperature for an air-source heat-pump installation is usually more critical than for fuel-fired heating systems because the heat pump capacity is generally more closely matched to the design requirement for heating. This results from the desire not to oversize the installation with respect to cooling because of the large heating requirements. Moreover, the decreased heating capacity of the air-source heat pump at low temperatures establishes the *balance point*, i.e., the outdoor temperature at which the capacity matches the heat requirement.

When the surface temperature of an outdoor air coil is 32 F or lower, frost may form and if allowed to continue, will interfere with heat transfer. Research has shown that with a nominal amount of frost deposit (typically about 2 to 3 lb per sq ft of coil face area) the heat transfer capacity of the coil is not substantially affected.[10] The number of defrosting operations will be influenced by the climate, the air-coil design and hours of operation. Experience has shown that little defrosting is generally required below 20 F and below 60 percent relative humidity.[10] This may be confirmed by psychrometric analysis using the principles given in Chapter 23. However, it should be noted that under very humid conditions when small suspended water droplets may be present in the air, the rate of frost deposit may be about three times as great as would be predicted from psychrometric theory. Under such conditions, a heat pump may require defrosting after as little as 20 minutes of operation. In applying an air-source heat pump, the effect of this condition on loss of available heating capacity should be taken into account.

Early application of air-source heat pumps followed commercial refrigeration practice involving relatively wide fin spacing; (i.e., 4 to 5 fins per inch) on the theory that this would permit faster defrosting. However, experience has proved that with effective hot gas defrosting much closer fin spacing can be tolerated and obtains the advantage of a reduction in the size and bulk of the system. In current practice a fin spacing of 8 to 10 per inch is widely used and some of the newer units have outdoor coils with as many as 13 fins per inch.

Water

Water may represent a satisfactory, and in many cases an ideal, heat source subject to the considerations listed in Table 2. Well water in particular is attractive from the standpoint of its relatively high and nearly constant temperature, generally being about 50 F in northern areas and 60 F and higher in the south. Information on well water availability, temperature, and chemical and physical analysis is generally available from U. S. Geological Survey offices located in many major cities.

Surface or stream water may be utilized, but under winter conditions of reduced temperature, the cooling spread between inlet and outlet must be limited to prevent freeze-up in the water chiller which is absorbing the heat. Some large heat-pump systems have been in operation in Europe where

Table 2 . . . . Heat Pump Heat Sources and Sinks

Heat Source	Air	City Water	Well Water	Surface Water	Waste Water	Earth	Solar
Source classification	Primary	Primary or auxiliary	Primary	Primary	Primary or auxiliary	Primary or auxiliary	Auxiliary
Suitability as heat sink	Good	Good	Good	Good	Variable with source	Usually poor	May be used to dissipate heat to air
Availability (location)	Universal	Cities	Uncertain	Rare	Limited	Extensive	Universal
Availability (time)	Continuous	Continuous—except local shortages	Continuous—unless well runs dry	Continuous	Variable	Continuous, temperature drops as heat is removed, slowly rises when pump stops	Intermittent, unpredictable, except over extended time
Expense (original)	Low, less than earth and water sources except city	Usually lowest	Variable, depending on cost of drilling well	Low	Variable	High	High
Expense (operating)	Relatively low	High, usually prohibitive	Low to moderate	Relatively low	Low	Relatively moderate	Unexplored. Promising for reducing auxiliary operating cost
Temperature (level)	Favorable 75-95% of time in most of United States	Usually satisfactory	Satisfactory	Satisfactory	Usually good	Initially good—drops with time and rate of heat withdrawal	Excellent
Temperature (variation)	Extreme	Variable with location (10 to 25 F deg)	Small	Moderate	Usually moderate	Large—less than for air, however	Extreme
Design information	Usually adequate	Usually adequate	Usually adequate	Usually adequate	Adequate if source is constant in supply and temperature	Inadequate	Practically available
Size of equipment	Large	Moderate	Moderate (except for well)	Moderate	Variable (usually moderate)	Moderate (except ground coils)	Particularly large
Adaptability to mass production	Excellent, can be factory assembled and tested	Excellent	Excellent (except for well)	Excellent	Poor	Fair	Fair
Sources it may augment	Air, earth	Air, earth				Air, water	Air, earth water
Special problems	Least heat available when demand greatest. Coil frosting requires extra capacity, alternate source, or standby heat. May require duct work. Variable air temperature makes control difficult.	Scale on coils. Local use restrictions during shortages. Disposal. Water temperature may become too low to permit further heat removal.	Corrosion, scale may form on heat transfer surface. Disposal may require second well. Water temperature may become too low to permit further heat removal.	Water may cause scale, corrosion, and algae fouling.	Usually scale forming or corrosive. Often insufficient supply. Very limited application. Requires individual design.	Limited by local geology and climate. Installation costs difficult. Requires considerable ground area, may damage lawns, gardens.	Probably will require heat storage and other equipment or condenser side.

winter stream conditions around 35 F have permitted inlet-outlet temperature spreads of only 1 or 2 deg.

Under certain industrial circumstances, use may also be made of waste process water as a source, as for example spent warm water in laundries or warm condenser water from large industrial condensers, for specialized heat-pump operations.

Utilization of water during cooling operation follows, in general, conventional practice with water-cooled condensers.

Water-refrigerant heat exchangers generally take the form of direct-expansion water coolers either of the shell-and-coil type or of the shell-and-tube type and are circuited to permit usage as a refrigerant condenser during the heating cycle and as a refrigerant evaporator during the cooling cycle. Further information on this type of equipment will be found in Chapter 38.

Earth

Earth as a heat source and sink, by heat transfer through buried coils, has not been used extensively. This may be attributed to high installation expense, ground area requirements, and the difficulty and uncertainty of predicting performance.

Compositions of soil vary quite widely from wet clay to sandy soil, and have a predominant effect on thermal properties and attendant overall performance. The heat-transfer process in the soil is primarily one of bulk temperature change with time (e.g., transient heat flow). *Thermal diffusivity* α, the ratio of thermal conductivity to the product of unit density and specific heat, $\alpha = k/\rho c$, is a dominant factor and is difficult to determine at different building sites. The moisture content likewise has an influence, since energy in the ground may also be transported via moisture travel in the soil.

In general, earth coils, usually arranged to be spaced horizontally at from 3 to 6 ft, are submerged below the surface from 3 to 6 ft. Though a lower depth might be preferred, excavation cost requires a compromise. Mean undisturbed ground temperatures generally follow the mean annual climatic temperature for any place.

Many individual studies have been conducted to determine earth heat transfer with respect to heat-pump possibilities, involving theoretical as well as practical considerations (References 20 through 32).

As mentioned in the section Heat Pump Types, two general systems prevail for utilization of earth coils. The first is that of direct expansion of a refrigerant in a buried coil under winter heating conditions with refrigerant vapor condensation under summer conditions, and the other is the transfer of heat between a liquid pumped through a buried coil and the earth (indirect system). The direct system involves special provisions to insure the proper flow of liquid and vapor, that is, pitching downward to prevent trapping of liquid in pockets, etc. The indirect system is less sensitive since the liquid is pumped through the coils under regulated flow rates.

The coil length required, as quoted by some authorities, is 200 to 400 ft of ¾- to 1½-in. pipe per ton of refrigeration for a coil buried at a depth of 4 ft with average pipe spacing 4 ft on centers. To insure continued contact of the coil with ground material, despite earth settling tendencies, the space directly around the coil is sometimes backfilled with sand.

Solar

Although still largely in the research stage, increasing interest is being shown in solar energy as a heat source, either on a primary basis or in combination with other sources. The principal advantage of employing solar radiation as a heat-pump heat source is that, when available, it provides heat at a higher temperature level than other sources, thus resulting in an increase in coefficient of performance.[33] As compared to a solar heating system without a heat pump, the collector efficiency and capacity are materially increased due to the lower collector temperature required.

Research and development in the solar-source heat-pump field has been concerned with two basic types of systems: direct and indirect. In the direct system refrigerant evaporator tubes are embodied in a solar collector, usually of the flat-plate type. Research has shown that when the collector has no glass cover plates the same collector surface can also function to extract heat from the outdoor air.[34] The same surface may then be employed as a condenser using outdoor air as a heat sink for cooling. The refrigeration circuit employed may resemble that shown in Table 1 for an air-to-air heat pump.

An indirect-type system employs another fluid, either water or air, which is circulated through the solar collector. When air is used, the first system shown in Table 1 for an air-to-air heat pump may be employed, the collector being added in such a way that: (1) the collector can serve as an outdoor-air preheater, (2) the outdoor-air loop can be closed so that all source heat is derived from the sun, or (3) the collector may be disconnected and the outdoor air used as the source or sink.[35] When water is circulated through the collector, the heat-pump circuit may be of either the water-to-air or water-to-water type[36] illustrated in Table 1.

On all heat-pump systems employing solar energy as the only heat source, either an alternate heating system or a means of storing heat is required during periods of insufficient solar radiation. Heat storage is discussed in a later section of this chapter.

PERFORMANCE CHARACTERISTICS

Temperature Levels

Temperature differences are required to transfer heat from the heat source to the evaporating refrigerant of a heat pump, and similarly from the condensing refrigerant to the heating fluid. Hence the total temperature lift of the refrigeration system must be increased over that represented by the difference between the heat source and sink temperatures.

Fig. 2 illustrates typical temperature levels for both heating and cooling operation. Specific heat pumps may vary somewhat from the temperature levels shown depending on their design and specific conditions of operation. Equation 3

Fig. 2 . . . . Heat-Pump Temperature Levels

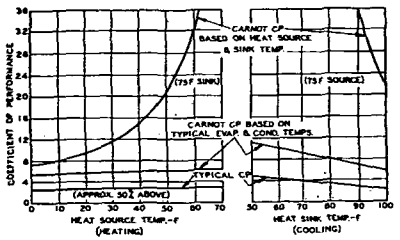


Fig. 3 .... Variation of Coefficient of Performance with Temperature

of Chapter 38 defines the coefficient of performance (CP) of a Carnot cycle heat pump. Applying the relationships given with evaporator and condenser temperatures illustrated in Fig. 2 yields the coefficient of performance (CP) for a Carnot cycle heat pump shown in Fig. 3. An actual heat pump will, as a rough approximation, be about 50 percent as efficient as a Carnot cycle heat pump and will have a coefficient of performance dependent on temperatures, as illustrated in Fig. 3.

System Balance

The performance characteristics of a heat-pump system can be predicted by evaluating and combining the performance characteristics of its individual components.[37] Data available from component manufacturers may be portrayed as shown in Fig. 4.

The conditions of system balance can be established by the following procedure:

1. Choose a combination of evaporator refrigerant temperature t_r and condensing temperature t_c.
2. Determine the compressor refrigerating effect from performance curves similar to *A* of Fig. 4.
3. Determine the compressor power input P_c, in kilowatts, from curves similar to *B* of Fig. 4.

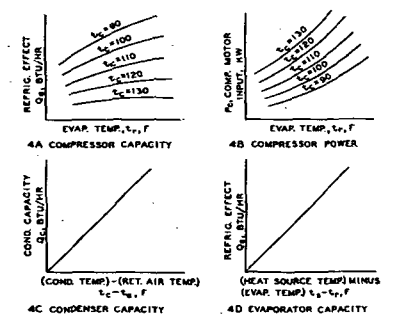


Fig. 4 .... Performance Characteristics of Heat-Pump Components

4. Determine the condenser capacity from

$$Q_c = Q_e + 3413\,P_c - Q_{ec} \qquad (1)$$

where

Q_c = condenser capacity, Btu per hour.
Q_e = compressor refrigeration effect (evaporator capacity), Btu per hour.
Q_{ec} = heat loss from compressor, Btu per hour. (Q_{ec} may have to be approximated if unavailable from compressor manufacturer. For many compressors Q_{ec} is negligible.)

5. Plot Q_c obtained from Equation 1 on a chart similar to *C* of Fig. 4. (See Fig. 5.)
6. Select other condensing temperatures in combination with the original evaporator temperature from Step 1 and repeat Steps 2 to 5 as necessary to determine the condenser capacity at which the system balances. Points *A* and *B* on Fig. 5 represent the results of these calculations. Two points will normally be sufficient to determine the balancing Q_c.
7. Select other evaporator temperatures and repeat Steps 1–6 (See Fig. 5.)
8. For each evaporator temperature find the corresponding heat source temperature t_s from chart similar to *D* of Fig. 4.

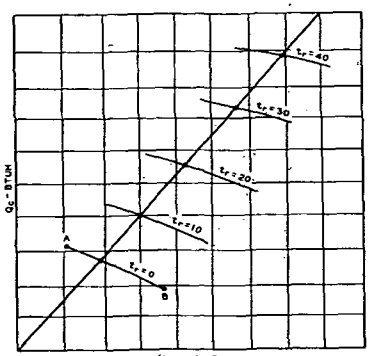


Fig. 5 .... Heat-Pump System Balance

With the conditions of system balance found from the foregoing procedure, it is a relatively simple matter to establish the heating performance characteristics. The net heating effect may consist only of the condenser heat, or, depending upon the system design, may also include heat losses from the compressor and motors, and the heating effect of a refrigerant subcooler coil.

The cooling performance can be determined in a similar manner.

For a heat pump which employs a constant temperature heat source, a few computations will generally establish the balancing conditions for t_r and t_c. For a heat pump having source temperature subject to considerable variation (e.g. air source), the balancing conditions for a wide range of t_r *will be necessary.*

Fig. 6 shows the performance characteristics of a typical heat pump determined either from actual system tests or

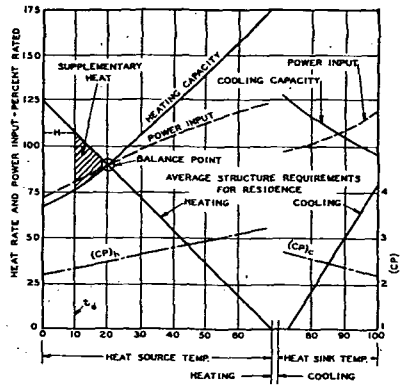


Fig. 6 .... Operating Characteristics of Single-Stage Unmodulated Heat Pump

from an analytical procedure such as already described. Also shown are the heating and cooling loads for a typical residence. The temperature at which the heat-pump capacity and the structure heat requirement are equal is referred to as the *balance point*. If the balance point is above the heating design temperature, t_d, then supplemental heat will be required as denoted by the shaded area.

Seasonal Performance

Fig. 6 illustrates some of the basic factors important in evaluating seasonal power consumption. Following the principles given in Chapter 37 the heating power consumption for any period of time may be estimated from

$$F = \frac{24HD}{3413(t_s - t_d)(\mathrm{PF})}$$

where

F = heating kilowatt hours
H = structure design heat loss, Btu/hr.
D = degree days for specified period.
t_s = outdoor temperature at which structure heat loss is zero (approximate indoor temperature when heat gain from other sources is small), Fahrenheit.
t_d = outdoor design temperature, Fahrenheit.
(PF) = performance factor for specified period.

For a heat pump with a constant heat-source temperature, the heating capacity will be essentially constant and the performance factor (PF) can be taken as the heating coefficient of performance $(CP)_A$.

For an air-source heat pump, with its varying heat source temperature, it is necessary to either break down the period of operation into relatively narrow temperature spans and to compute the power consumption in each, or, alternatively, to find the performance factor (PF) by weighted average of the different coefficient of performance (CP) and operating hours that apply at various temperatures. In either case it is necessary to have data on the time distribution of outdoor-air temperatures. The number of hours that will apply within a given temperature range may be based on past weather data for a specific location. For example, such data are available for a number of cities broken down by month and time of day.[38] However, where the required weather data are not available and when it is not necessary to break the heating season down into shorter periods, generalized data such as portrayed on Fig. 7 may be employed.

Fig. 7 is based on an analysis of 114 weather stations in the United States from Reference 38, in which the heating-season days were correlated with the TAC design temperatures. The TAC design temperatures shown in 1948 to 1958 editions of THE GUIDE were suggested by the ASHAE Technical Advisory Committee on Weather Design Conditions. They were temperatures which were equalled or exceeded during 97½ percent of the hours in December, January, February, and March chiefly for airport stations.

Maximum deviation for any single location from the averages portrayed is on the order of 5 deg at 70 F outdoor temperature and is due mainly to variations in summer climate. Since curves must pass through points representing the winter TAC design temperatures, their accuracy is best in the low-temperature range. The vertical scale also gives the outdoor design temperatures for heating, as given in Chapter 12, which take into account the probabilities of occurrence. Reference 1 of Chapter 12 provides the correlation of these design temperatures with the TAC temperatures.

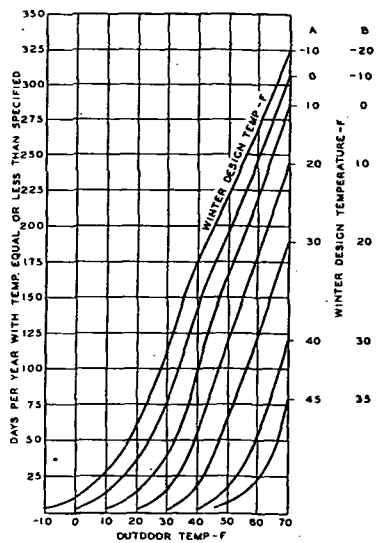


Fig. 7 .... Annual Cumulative Winter Temperature Distribution in the U. S.

Table 3 .... Number of Defrosts per Year[a] (100 Percent Running Time)

Climate (TAC)	−10 to 5	6–8	9–11	12–14	15–17	18–20	21–23	24–26	27–29	30–32	33–35	36–39	39–41
Normally dry	475	450	400	375	350	300	250	200	175	150	100	75	50
Normally damp	800	750	700	625	575	500	425	350	300	225	175	125	100

[a] The figures given must be multiplied by the running-time ratio for the period considered.

Extra power consumption allowance must be made where periodic defrosting of an air-source evaporator is required, and where supplemental heating is employed.

The number of defrosting operations required per month or a complete heating season is difficult to evaluate accurately. It will depend on the detailed design and control of the heat pump, the climate, and the hours of operation. Table 3 gives the range in annual number of defrosts that may be expected on typical small unitary heat pumps.

To obtain a total yearly operating cost, the energy consumption for cooling must be added to that required for heating. Suggested procedures for calculating cooling energy consumption are given in Chapter 46 and 54.

Annual Operating Cost

A tabulation such as is shown by Table 4 may be used for estimating the operating cost for commercial and industrial installations. It is necessary to know the monthly kilowatt electric demands and the corresponding kilowatt-hour consumption for the base electric load, as well as for the heat pump or any other electric equipment, in order to apply the proper electric tariff. The base loads, shown as Columns 1, 2 and 3 of Table 4, may include lighting, elevators, office machinery, exhaust and supply fans, circulating pumps, and similar items which are normally essential in a structure and are not affected by a particular air-conditioning design. The heat-pump kilowatt demand, during the month, which is required to satisfy the net day and night heat loss or heat gain (whichever is the greater), can be obtained from manufacturers' data. These resulting kilowatt demands are multiplied by a coincident factor to obtain the coincident demands, listed in Columns 4 and 7. These coincident demands can be added to the base load demands to obtain the total electric load in Column 10. The coincident factors can best be obtained from published data on similar installations. The procedure for finding the heating and cooling kilowatt hours, Columns 5, 6 and 8, has been previously described.

For residential installations, a detailed monthly estimate generally is not justified unless a monthly demand type of rate is in effect.

In making a power consumption estimate for unitary heat pumps, a detailed breakdown into periods less than a complete season, or the breaking down into small temperature spans, is not ordinarily justified. Manufacturers of this type of equipment provide estimating procedures which are simpler to use, yet are of sufficient accuracy for the purpose intended. These methods usually consist of an empirical equation, tabular data, or plotted curves which combine the influences of the size and thermal properties of the structure, the heating design temperature, the characteristics of the heat pump, and supplemental heat usage.

HEAT-PUMP COMPONENTS

For the most part, the components used in heat pumps and the practices followed bear a direct relation to the low temperature refrigeration art. This section will outline the major components used and point out characteristics or special considerations which apply to the heat-pump field.

Compressors

Reciprocating compressors are used more than any other type on systems in the general range of ½ to 100 tons, and frequently even larger. A brief description of this type of compressor is given in Chapter 38. For the most economical application, it is general practice to select the reciprocating compressor to provide the desired cooling capacity. The cooling capacity is the evaporator capacity and represents the ability of the compressor to pump refrigerant vapor between the existing temperature limits. When heating, the capacity of a particular compressor is the sum of the evaporator capacity (i.e., heat-source coil capacity) plus the heat equivalent of the compressor work.

Figs. 13, 14 and 15 in Chapter 38 shows the capacity and power characteristics of typical reciprocating compressors. A compressor normally employed for comfort cooling use will have a clearance volume (ratio of gas volume remaining in cylinder after compression stroke to total swept-cylinder

Table 4 .... Suggested Procedure for Computing the Operating Cost of Commercial and Industrial Heat-Pump System

Period	Base Load Lighting and Miscellaneous		Cost Base Load	Heat Pump			Supplemental Resistance Heating (If Used)		Total Heating and Cooling	Total Electric Load		Total Cost
				Coincident demand	Heat	Cool						
	Kw (1)	KwH (2)	KwH (3)	Kw (4)	KwH (5)	KwH (6)	Kw (7)	KwH (8)	KwH (9)	Kw (10)	KwH (11)	(12)
June												
July				Fill in one line for each month through May								
May												

volume) of about 5 percent. The capacity drop off at low evaporator temperatures corresponding to low heat-source temperatures is also evident. The drop off in power at low evaporator and condensing temperatures is also evident.

If the compressor has low clearance volume, say 2½ percent, then it is more suitable for low-temperature operation, and will provide, for example, about 15 percent greater refrigerating capacity at an evaporator temperature of 0 F and a condenser temperature of 110 F. However, this compressor has somewhat more power demand under maximum cooling load conditions than does one of medium clearance.

It is obvious that more total heat capacity can be obtained at low outdoor temperatures by deliberately oversizing the compressor. The compressor is oversized in order to obtain more heating capacity so it is necessary to provide some type of capacity reduction to bring the cooling capacity into balance. This can be done by means of two-speed motor drives, cylinder cutouts, or other methods as described in Chapter 38. The disadvantage of this arrangement results from the fact that the greater number of operating hours that occur at the higher suction temperatures must then be served with the compressor in the unloaded condition which generally causes lower efficiency. Therefore, the annual operating cost will tend to rise. It is also true that the additional first cost of the oversized compressor must be economically justified by the gain in heating capacity. One method proposed for increasing the heating output at low temperatures involves the use of *staged compression* in which one compressor may pump from −20 F suction temperature to 40 F condensing temperature and a second compressor compress the vapor from 40 F to 120 F. In such an arrangement it is possible to interconnect any two compressors so that they are in parallel, both pumping from say 45 F to 120 F at the normal cooling rating point, while at some predetermined outdoor temperature on heating they are reconnected so that they pump in staged relationship.

Fig. 8 shows the performance of such a pair of compressors for compressors of both medium and low clearance volume. It is apparent that at low suction temperatures the reconnection into a staged relationship does provide some added capacity. Also, it should be understood that the motor selection involved must be based upon the maximum loading conditions for summer operation even though the low stage compressor has a greatly reduced power requirement under the heating condition.

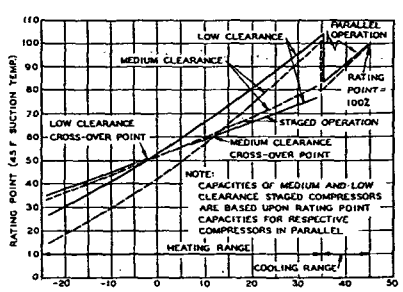


Fig. 8 .... Comparison of Parallel and Staged Operation

The coefficient of performance will be approximately the same whether they are coupled in parallel or compound-staged at any set of operating conditions depending somewhat on the motor characteristics when lightly loaded.

A *rotary* compressor has characteristics similar to a reciprocating compressor except that it is by nature possessed of low clearance and high volumetric efficiency. From this standpoint it is well suited to heat-pump service, providing about 30 percent greater capacity at 0 F—110 F lift than a medium-clearance reciprocating compressor. However, this characteristic tends also to increase the power demand at the maximum cooling load conditions. At this time, reciprocating compressors are most widely used for heat pumps, with rotary compressors restricted to the first stage of a staged-compression system and not used during the cooling cycle.

Heat Transfer Components

Refrigerant-to-air and refrigerant-to-water heat exchangers, as previously described in the section Heat Sources and Sinks, are similar to heat exchangers used in current air-conditioning practice. A refrigerant subcooler coil may be employed either in conjunction with an indoor air coil, or, on systems with a ventilation air supply, to preheat ventilation air. A substantial gain in capacity and coefficient of performance can result.

Refrigeration Components

Refrigerant piping, receivers, expansion devices and refrigeration accessories in heat pumps are usually the same as those used in other types of refrigeration and air-conditioning systems (see Chapter 38).

A *reversing valve* is used to change the system from the cooling to the heat operation. This change-over requires the use of a valve, or valves, in the refrigerant circuit,* except where the change is accomplished in fluid circuits external to the refrigerant circuit (see Table 1). Reversing valves are usually pilot-operated by means of solenoid valves which admit head and suction pressures to move the operating elements.

Expansion devices for controlling the refrigerant flow are normally thermostatic expansion valves as described in Chapter 38. Special requirements may sometimes be presented. If the circuiting is so arranged that the refrigerant line upon which the control bulb is placed can become the compressor discharge line, the resulting pressure developed in the power element of the valve may be excessive, requiring the use of a special control charge or pressure-limiting element. When a thermostatic expansion valve is applied to an outdoor air coil, a special *cross-charge* is desirable to limit the superheat at low temperatures, and thereby obtain a better utilization of the coil.

When an expansion valve is attached to a coil that is operated as a condenser, a bypass with a check valve is normally provided as indicated in Table 1.

On small factory-built systems, capillary tubes are normally used as expansion devices. While a single capillary tube on both heating and cooling is sometimes employed, better efficiency and performance can be obtained by using a more restrictive capillary tube for heating than for cooling. This may be accomplished by using two capillary tubes in either a series or parallel arrangement with a check valve to bypass one for cooling or to block one for heating, respectively.

On an air-source heat pump that must operate over a wide range of evaporating temperatures, a capillary tube will tend to pass refrigerant at an excessive rate at low back pressures, causing liquid floodback to the compressor. In some cases suction line accumulators or charge-control devices are employed to minimize this effect.

A *refrigerant receiver* is commonly used, as in other types of refrigeration systems, to provide a storage place for liquid refrigerant. It is particularly useful in a heat pump in order to take care of the unequal refrigerant requirements between heating and cooling.

Control Components

Heat pump control systems and components are of the same general type used in other types of heating and cooling equipment. Reference should be made to Chapter 43.

Practically all heat pumps for residential heating and cooling are controlled automatically from the temperature of the conditioned space. *Room air thermostats* are usually of the type that combine the control of both the heating and cooling function, the selection of function either being accomplished automatically in response to air temperature or manually by a selector switch. When supplementary heaters are employed, usually electric resistance heaters, it is common to control them with the second stage of a two-stage heating control. This restricts their usage to makeup of the difference between the heating requirement and the heat-pump capacity (see Fig. 6). To avoid unnecessary usage and excessive electrical demand, particularly when the thermostat setting is raised suddenly, it is customary to use one or more outdoor thermostats to limit the amount of resistance heat used at the higher outdoor temperatures. For information on electric resistance heaters see Chapter 17.

In residential heat pump applications, night-set-back, common with fuel-fired systems, is not ordinarily employed because substantial excess capacity is required for warm-up and because possible operating cost savings, if any, are deemed to be small.

On the larger type systems, a proportional action type of control is sometimes used for controlling compressor and supplementary stages in steps.

A variety of *defrosting control* schemes have been used to sense the need for defrosting air-source heat pumps and to initiate and terminate the defrost cycle.

A timer is sometimes used and set to cause defrosting at predetermined intervals for instance, about every two hours. After initiation of the defrost cycle defrosting can be terminated either by the use of a control sensing the coil pressure or a thermostat located so as to measure the temperature of the liquid refrigerant in the outdoor coil. When the temperature (or corresponding saturation pressure) of the liquid leaving the outdoor coil rises to about 40 F the completion of defrosting is assured. Termination of defrosting may also be obtained by use of a second time interval control.

Another method of starting the defrost cycle is to use a pressure control which reacts to the air pressure drop across the coil. Under conditions of frost accumulation the air flow will be reduced and the increased pressure drop across the coil will initiate the defrost cycle. Again the preferred method of terminating the defrost cycle is to use a refrigerant temperature control measuring the temperature of the liquid refrigerant in the coil.

A third method for defrosting involves a temperature differential control in which two temperature sensing elements are used; one responsive to the outdoor-air temperature and the other responsive to the temperature of the refrigerant in the coil. As frost accumulates, the differential between outdoor temperature and refrigerant temperature will increase, causing a defrost cycle to be initiated. The system will be restored to operation when the refrigerant temperature in the coil reaches a specified temperature indicating that defrosting has been completed. When the outdoor-air temperature decreases, the differential between outdoor-air temperature and refrigerant temperature decreases, and the defrost cycle is initiated sooner.

Other Components

Supplementary *resistance heaters*, commonly used with many unitary type heat pumps may be incorporated either within the unit or external to it. They may take any of the forms discussed in Chapter 17. When installed in the distribution ductwork, they are frequently controlled to temper the air during defrosting operation on units employing *reverse-cycle* defrost.

APPLICATION

Criteria for Feasibility

Factors to be considered in determining the practicability of employing a heat pump include the following:

1. Presence of a cooling load.
2. Relative amounts of heating and cooling.
3. Availability of suitable heat source and sink.
4. Equipment requirements and installed cost.
5. Operating cost.

These criteria may be applied whether the equipment to be considered is of the factory-built, unitary type or the field-assembled type. In fact, an analysis of these factors, in addition to determining feasibility of the heat pump for a particular application, should also be useful for choosing the equipment type.

Unless there is a need for cooling, heating with a heat pump is ordinarily not economically feasible. Under certain circumstances this may not be true, but usually a heat-pump system will have a higher first cost than a fuel-fired heating system.

With the almost universal need for comfort cooling, particularly in commercial establishments, but also to an increasing extent in residences, the feasibility of a heat pump becomes more favorable. Only a portion of the required equipment will then have to compete with a separate heating system. In fact, in large buildings, the existence of simultaneous heating and cooling loads in different zones for a large portion of the time is often uniquely suited to a heat-pump system.

The relative amounts of heating and cooling required may greatly influence the first cost and the suitability of a heat pump. In the usual case the size of the heating load is the predominating factor, particularly with structures in colder climates having low internal heat gain. The application of insulation and adequate sealing against unwanted air infiltration reduces the ratio of the heating to cooling loads.

The availability of a suitable heat source and sink generally involves making the optimum selection from the standpoint of overall economics. Factors to be considered were presented earlier in this chapter in the section Heat Sources and Sinks.

The heat source selected will usually also serve as a heat sink. It should be recognized that in applications requiring heating and cooling in different zones simultaneously, the heat source or sink need only handle the net heat input or output of the system.

To *estimate the equipment requirements and installed cost* it is necessary to make at least a tentative equipment selection. Before this can be done it will, first be necessary to determine the heating and cooling loads. For small structures this generally entails only the calculation of the peak heating and cooling loads. For larger buildings, a more detailed analysis may be required in order to take into account the loads at other than the design conditions, the time periods at which loads occur, ventilation schedule, variations in loads due to internal heat gains from lights, equipment and occupancy, and simultaneous heating and cooling loads in different zones.

It is implicit in a feasibility study of this kind that the objective is to obtain a comparison between the cost of a heat-pump system and a combination air-conditioning and fuel-fired heating system. Therefore, in order to estimate the first cost of both systems, a preliminary design and selection of each must be made.

It does not necessarily follow that the system with the lowest first cost is the optimum choice. An increase of initial cost for any system may be justified by savings in operating cost or by other benefits derived.

The principal factors to be taken into account in estimating *operating cost* are given in the section Performance Characteristics. The application or feasibility study of a heat-pump system with regard to operating cost involves the comparison between it and alternative types of systems considered. Operating cost of fuel-fired systems may be estimated by the procedures given in Chapter 37.

Heating and Cooling Distribution

The distribution of the heating and cooling effect from a heat-pump system is an important application consideration, particularly because of the relatively low temperatures involved as compared to fuel-fired heating systems.

In a commercial-type structure, any system of distribution which is satisfactory for cooling will generally be satisfactory for heating with a heat pump. Conversely, any system satisfactory for heat distribution with a heat pump will be satisfactory for cooling, except, however, that panel systems will seldom have sufficient capacity to handle normal cooling loads satisfactorily.

Heat distribution in the space can be accomplished by forced air at high, medium, or low pressures; primary air with induction units; individual fan and coil units; a split system composed of forced air and panels; or panels alone. All these methods of heating, except panels alone, are as applicable to cooling as to heating.

The distribution from the heat pump to space units can be by use of water or air. Quantities circulated, pipe or duct sizes, transfer units, and other characteristics of the distribution system will resemble very closely standard practice for cooling systems.

Heating and cooling distribution in residential heat-pump systems generally follows the same principles as for summer cooling. Refer to Chapter 46. It has been found with residential air distribution systems, comfort may be achieved with the air volumes and temperatures encountered with heat pumps when proper attention is given to the number and location of air outlets.[1]

Design and Selection

Irrespective of the size and type of building and whether unitary or field-erected equipment is being considered, the first steps that should be performed are those described under Criteria for Feasibility. In applying unitary heat pumps, the manufacturers of these units ordinarily provide sufficient data and instructions for proper application. The typical procedure followed for both unitary and field-erected heat pumps is outlined below:

1. Determine heating and cooling loads. It is desirable also to determine the amount that loads may be reduced by additional insulation, sealing, and shading. Savings effected in equipment due to distribution and to operating costs may more than offset the added cost involved. In larger buildings, loads should be determined for the different conditions of operation including simultaneous heating and cooling.
2. Establish heat source and sink. See previous section on this subject.
3. Determine type of distribution system.
4. Choose major components: compressors, heat exchangers, and air handlers. For unitary equipment, establish equipment size, type, and make.
5. Establish design of refrigeration system.
6. Choose auxiliary equipment: fans, pumps, valves, etc.
7. Determine equipment location. Since the heat pump operates without fuel and combustion products, considerable flexibility is permitted in its location. A single-package unit, may be located in a basement, crawl space, attic, roof, garage, or carport, in the conditioned space, or out-of-doors. For remote-type, or split units, one section is generally located outdoors and the other in a convenient indoor location. Considerable flexibility in locating equipment of a field-erected heat pump in one or more areas is possible.

   Factors to be considered in equipment location are: (*a*) Accessibility to heat source and sink, (*b*) Convenient location with respect to indoor distribution system, usually duct work, (*c*) Availability of condensate drain, (*d*) Required accessibility and sufficient space for servicing, (*e*) Convenience to electric power wiring, (*f*) Suitability of structure for supporting weight of equipment, (*g*) Effect of noise and vibration transmission, and (*h*) Appearance.
8. Determine control system and components.
9. Estimate first cost and operating cost.

Illustrative Systems for Larger Buildings

Fig. 9 shows a system employing well water as a heat source and sink. In this particular case a heat exchanger is used between the well water circuit and the fluid circuit to transmit heat to the heat pump, heat exchangers, and the indoor conditioning coils. Although this requires an extra step of heat exchange, it avoids the necessity for pumping well water throughout the building and permits a high rate of flow in the building water circuit with resulting improvement in heat transfer. The use of a water preconditioning coil as shown to preheat, or precool, ventilation air often reduces the size of refrigeration capacity required as well as the cost of operation. It should be noted that both heated and cooled water can be furnished simultaneously to indoor-air-handling or remote-room units as may be required by the existence of heating and cooling loads in different zones at the same time.

Fig. 10 illustrates an air-to-air heat pump system which is similarly suited to a structure having simultaneous heating and cooling loads. The refrigeration subcooling surface shown is used to preheat ventilation air and thereby effect substantial economies in equipment size and operating cost.

HEAT STORAGE

The use of thermal storage in a heat-pump system can improve its performance characteristics. Installations of heat pumps with thermal storage have been made principally in a few large building systems.[2, 3, 4]

All materials possess the property of thermal storage in

greater or lesser degree. In the case of a building, the structural materials are almost always in the process either of absorbing heat from or delivering heat to the interior space. This effect is more pronounced in cooling operation where greater air temperature variation is tolerated. Storage tends to reduce the rate of temperature change and helps in some measure to reduce the peak equipment requirements. In this sense every heating and cooling system can be said to involve heat storage in some degree.

Many attempts have been made, particularly in recent years, to increase the heat-storage effect by using special heat-storage materials as part of the heating or cooling system. The result is to reduce the size of the heating or cooling equipment necessary to take care of peak demands.

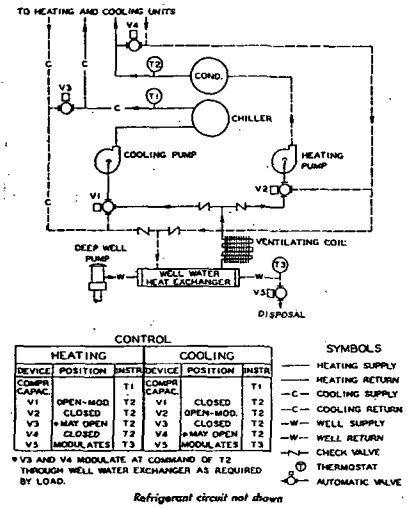


Fig. 9 .... Flow Diagram of Central-Type Heat-Pump System

In the case of the heat pump, a provision for heat storage can serve not only to reduce the size of the heat pump necessary for a given load, but also to provide a more desirable electrical load by shifting part of the load to the time of day when the cost of power is least. The off-peak electric hot water heater is a common example of such a heat-storage application.

In general, there are two types of heat-storage systems that have been employed: (1) sensible heat-storage systems and (2) latent heat-storage systems. Usually the latter is actually a combination of the two, making use of some sensible heat storage in addition to the latent effect. Heat storage in a heat-pump system may be utilized on the *high side*, when heat is available at a temperature suitable for direct heating; or on the *low side* as an intermittent heat source at temperatures lower than the heated space.

Table 5 .... Properties of Various Specific Heat Type Storage Materials

Material	Specific Heat Btu per (lb) (cu ft) (F deg)	Density lb/cu ft	Heat Capacity Btu per (cu ft) (F deg)
Water	1.00	62.4	62.4
Iron	0.11	490.0	53.9
Concrete	0.27	140.0	37.8
Brick	0.20	120.0	24.0
Gravel	0.20	100.0	20.0

Sensible Heat Storage

Water, with the highest sensible heat storage capacity on a weight basis, is an obvious first choice and has the advantage of low cost, easy storage, and good heat exchange properties. The danger of freezing can be reduced by various types of antifreeze solutions. In some cases, however, freezing may be desirable. The maximum temperature is limited by the cost of maintaining water under pressures when over 212 F, but this temperature is high enough for most heating systems. Rock and gravel beds have also been employed, and have a heat content per unit volume approximately 20 percent of the value for water. The earth has been utilized for heat storage, both for heating and cooling in a number of instances. A problem with most solid media such as these is one of heat transfer into and out of the medium.

Table 5 lists the thermal properties of some of the common materials used, or considered, for sensible-heat storage.

Latent-Heat Storage Systems

Because of the relatively large volumes required for all sensible-heat storage systems, many attempts have been made to utilize the latent heat of fusion of certain materials. Most materials have some capacity to absorb heat when changing from a solid to a liquid, and some materials have a large capacity to do so. Ordinary ice changing to water is a prime example of this kind and the possibility exists of utilizing it as low-side storage. For most heating applications, however, the temperature of this change is too low to be useful.

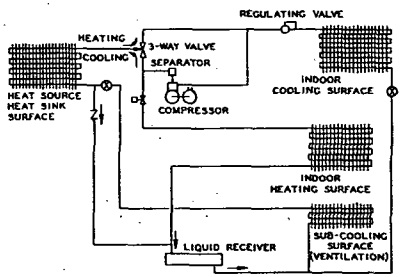


Fig. 10 .... Flow Diagram of Air-to-Air Heat Pump for Simultaneous Heating and Cooling

Table 6 .... Materials for Heat Storage Listed According to Latent Heat per Unit Volume[a] of All Materials Having Melting Points between 72 F and 176 F, Listed in Chemical Handbooks

Material	Formula	Melting Point F	Heat Content G-cal per g	Specific Gravity	Latent Heat of Fusion per Unit Volume (Theoretical)	
					G-cal per cc	Thousands of Btu/cu ft
Nitrogen pentoxide	N_2O_5	85	76.7	1.63	125.0	14.0–
Gallium	Ga	86	19.0	5.88	113.8	12.7–
Sodium phosphate, dodecahydrate	$Na_2HPO_4{\cdot}12H_2O$	94	66.8	1.52	101.5	11.4
Sodium sulfate, decahydrate	$Na_2SO_4{\cdot}10H_2O$	90	57.1	1.46	83.4	9.4
Water	H_2O	32	79.1	1.00	80.0	9.0
Nickel nitrate	$Ni(NO_3)_2{\cdot}6H_2O$	132	36.4	2.05	74.7	8.4
Calcium chloride	$CaCl_2{\cdot}6H_2O$	86	40.7	1.68	68.4	7.7
Osmium tetroxide	OsO_4	104	13.5	4.91	66.3	7.5
Zinc nitrate	$Zn(NO_3)_2{\cdot}6H_2O$	98	31.1	2.06	64.3	7.2
Calcium nitrate	$Ca(NO_3)_2{\cdot}4H_2O$	108	33.9	1.82	61.7	6.9

[a] Nine materials having melting points between 72 and 176 F listed according to latent heat of fusion. Selected as highest on a list of 100 organic and inorganic materials listed in chemical handbooks.

Prolonged searches have been made for materials which pass through a change of state as reliably and regularly as ice and water, but at temperatures high enough to be used for direct application to space heating. In order to be much superior to water, the latent heat capacity of such materials must be high. Most organic materials are eliminated from consideration on this basis alone. Furthermore, the absorption and release of heat must occur in predictable fashion. Many materials which have been tried exhibit the defect of subcooling, delayed change of state, or decomposition through time and repeated cycles. A study of all materials, both inorganic and organic, listed in chemical handbooks resulted in a list of over 100 materials having a fusion temperature between 72 F and 176 F. Most of these materials are extremely unpromising as heat storage media because of cost, corrosiveness, instability, or unavailability. For many of them, data on their heat of fusion are unavailable. All of these materials, where such data are available, have been listed in order of heat capacity per unit volume. Table 6 gives the nine materials which were highest on this list, and it is interesting to note that there are few common or low cost materials among this group. Others, notably the first two, are academic curiosities.

The material disodium phosphate is high on the list and has been used in a number of experimental installations. Results for the most part indicated a considerable unreliability at least in the particular form of the material which was used. Such factors as the rate of heat addition or release, size and shape of container, presence of nucleating media, and temperature gradients proved important in determining the percentage of heat storage actually available compared with the theoretical. In several small house heating installations employing heat pumps with condenser-side storage, the amount of disodium phosphate actually used was 3300 lb with theoretical latent heat storage capacity of 400,000 Btu. Indications were that the material was only partially effective in actual heat-storage performance. A method which makes use of a mixture of this material suspended in oil has been successfully tested in the laboratory, and through use of forced circulation of the oil, has exhibited high rates of heat transfer.[9]

Low-Side Storage

With a heat-pump system the possibility also exists of storing heat at relatively low temperature on the low side (heat source side) of the system. The freezing of water has already been mentioned as one possibility. Heat pumps with solar energy as the heat source have been studied with low-side storage.[10]

A limitation present with this type of storage system as compared to high-side storage is that the peak heating capacity of the system is limited to the heat-pump capacity when operating with the low-temperature storage reservoir as the heat source. In many structures, particularly in colder climates where the benefits of heat storage are potentially greatest, this is less than the peak heating load.

REFERENCES

[1] William Thomson: On the economy of heating and cooling of buildings by means of currents of air (*Glasgow Phil. Soc. Proc.*, Vol. 3, December 1852).

[2] T. G. N. Haldane: The heat pump—an economical method of producing low-grade heat from electricity (*Journal of Institution of Electrical Engineers*, London, June 1930).

[3] A. R. Stevenson, Jr.: Refrigeration (*Journal Franklin Institute*, Vol. 208, August 1929).

[4] Philip Sporn and D. W. McLenegan: An all electric heating cooling and air conditioning system (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, August 1935).

[5] Experience with a reversed-cycle heating system (*Power*, August 1941).

[6] Philip Sporn, E. R. Ambrose, and Theodore Bannister: *Heat Pumps* (John Wiley & Sons, New York, 1947).

[7] E. N. Kemler and Sabert Oglesby, Jr.: *Heat Pump Applications* (McGraw-Hill Book Co., New York, 1950).

[8] *Bibliography of the Heat Pump through 1951* (*Edison Electric Institute Publication* No. 53-4).

[9] *Heat Pump Bibliography* (compiled by Southern Research Institute for Southeastern Electric Exchange).

[10] Philip Sporn: *Electric Heating and Year-Round Air Conditioning and the Heat Pump* (*Edison Electric Institute Bulletin*, May 1955).

[21] J. D. Kroeker and R. C. Chewning: Heat pump in an office building (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 221).

[22] J. D. Kroeker, J. H. Bonebrake, and J. A. Melvin: Heat pump application to a newspaper plant (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 467).

[23] Philip Sporn and E. R. Ambrose: Two-year performance of a heat pump system furnishing year-round air conditioning in modern office building (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 483).

[24] J. D. Kroeker and R. C. Chewning: Costs of operating the heat pump in the Equitable building (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 157).

[25] E. G. Hellier and H. C. Weingartner: Methods of frozen juice concentration (*Refrigerating Engineering*, July 1950, p. 673).

[26] T. F. Thomas: The air cycle heat pump (*Engineer*, August 22, 1947, p. 180; August 29, 1947, p. 205).

[27] W. F. Friend: Future electronic, solar, and nuclear developments (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 563).

[28] W. F. Stoecker: How frost formation on coils affect refrigeration systems (*Refrigerating Engineering*, February 1957, p. 42).

[29] *Heat Pump Electricity Usages and Other Operating Characteristics as Experienced under Various Weather Conditions (Southern Research Institute Report* No. 2371-288-SVIII, July 22, 1955).

[30] L. R. Ingersoll and H. J. Plass: Theory of the ground pipe heat source for the heat pump (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 339).

[31] E. W. Guernsey, P. L. Bets, and N. H. Skau: Earth as a heat source or storage medium for the heat pump (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 321).

[32] A. B. Algren: Ground temperatures as affected by weather conditions (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 363).

[33] G. S. Smith and Thomas Yamanchi: Thermal conductivity of soils for design of heat pump installations (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 355).

[34] L. R. Ingersoll, F. T. Adler, H. J. Plass, and A. C. Ingersoll: Theory of earth heat exchangers for the heat pump (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 167).

[35] G. S. Smith: Factors useful in ground grid design for heat pumps (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 189).

[36] Merl Baker: Design and performance of a residential earth heat pump (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 371).

[37] W. A. Hadley and Raymond Eisenstadt: Moisture movement in soils due to temperature difference (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 395).

[38] G. S. Smith: Intermittent ground grids for heat pumps (ASHAE TRANSACTIONS, Vol. 63, 1956, p. 473).

[39] W. A. Hadley: *Operating Characteristics of Heat Pump Ground Coils (Edison Electric Institute Bulletin*, Vol. 17, December 1949, p. 457).

[40] Joint AEIC-EEI Heat Pump Committee: *Research Results Concerning Earth as a Heat Source or Sink (Edison Electric Institute Bulletin*, September 1953).

[41] D. M. Vestal and B. J. Fluker: Earth as heat source and sink for heat pumps (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 41).

[42] D. M. Vestal, Jr. and B. J. Fluker: *A Proposed Procedure for the Design of a Heat Pump Buried Coil (Texas A & M Bulletin*).

[43] R. C. Jordan and J. L. Threlkeld: Availability and utilization of solar energy (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 177).

[44] Philip Sporn and E. R. Ambrose: The heat pump and solar energy (Association for Applied Solar Energy, *Proceedings World Symposium on Applied Solar Energy*, November 1955).

[45] C. P. Davies, Jr. and R. I. Lipper: Sun energy assistance for air-type heat pumps (ASHAE TRANSACTIONS, Vol. 64, 1958).

[46] F. H. Bridgers, D. D. Paxton, and R. W. Haines: Performance of a solar heated office building (ASHAE TRANSACTIONS, Vol. 64, 1958).

[47] F. R. Ellenberger, A. B. Hubbard, W. R. Foote, F. Burggraf, and J. J. Martin, Jr.: Evaluating heat pump performance (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 87).

[48] *U. S. Weather Bureau Forms* E-1-D, E-4-D, E-6-D (U. S. Department of Commerce).

[49] Alfred Koestel and G. L. Tuve: Performance and evaluation of room air distribution systems (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 533).

[50] Philip Sporn and E. R. Ambrose: Heat pump system using water storage (*Mechanical Engineering*, Vol. 69, November 1947, p. 899).

[51] Philip Sporn and E. R. Ambrose: The heat pump, an all-electric year-round air conditioning system (*Heating and Ventilating*, January 1944, p. 68).

[52] T. L. Etherington: A dynamic heat storage system (ASHAE TRANSACTIONS, Vol. 64, 1958).

BIBLIOGRAPHY

W. E. Johnson: Economic and technical aspects of the heat pump (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 201).

J. D. Kroeker, R. C. Chewning, and C. E. Graham: Heat pump results in Equitable building (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 345).

J. F. Sandfort: Thermodynamic criteria for heat pump performance (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 363).

G. S. Smith: Climatology as an aid in heat pump design (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 499).

C. F. Kayan: Electrical analogger application to the heat pump process (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 361).

W. F. Stoecker: How frost formation on coils affects refrigeration systems (*Refrigerating Engineering*, February 1957, p. 42).

Applied Solar Energy Research (Stanford Research Institute, 1955—contains numerous heat pump references).

Heat Pump Data Sheets on Residential and Commercial Installations (Edison Electric Institute Publication No. 56-12).

CHAPTER 40

EVAPORATIVE APPARATUS FOR HEAT REJECTION

Water Cooling Methods, Water Use and Conservation, Rivers and Lakes, Spray Ponds, Atmospheric and Mechanical Draft Towers, Cooling Tower Theory, Design Conditions, Selection and Evaluation, Tower Location, Operation and Maintenance, Evaporative Condensers

REFRIGERATING systems and many other industrial processes generate heat that must be removed and dissipated. Small quantities of heat can be rejected directly to the atmosphere by convection and radiation. When large quantities are involved, the most effective procedure is to transfer the heat to cooling water flowing through a heat exchanger.[1] The water, if cheap and plentiful, may be wasted to a sewer or open waterway such as a stream or lake. The source of the cooling water may be wells, a public water system, or a lake or river.

WATER COOLING METHODS

Water Use and Conservation

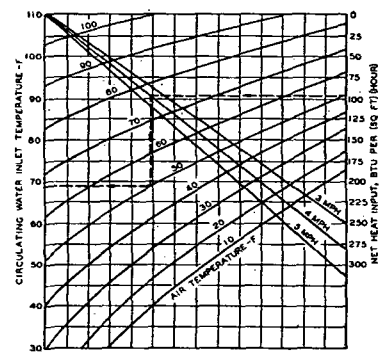
The rapidly expanding demand for water is depleting what was once looked upon as an inexhaustible natural resource. The water table is receding in most localities. Communities are finding their present water systems inadequate to supply the increasing demands of domestic and industrial users. Heavy industrialization along flowing streams causes pollution. The temperature of the water that may be discharged into the stream is frequently limited in order to protect wild life. Many cities do not have adequate sanitary or storm sewers to handle cooling water. As a result of the increasing demand and the lack of an adequate supply or disposal system, users are being forced to conserve water by using recirculating systems. Heat must be removed from the circulating water by some type of water cooling apparatus.

Rivers and Lakes

Plants located on flowing streams can use a once-through system, discharging the heated water downstream from the intake to prevent mixing. Intake screens and sometimes settling basins, are needed to remove debris. The intake structure must be designed to handle flow conditions varying from flood stage to the minimum flow during periods of drought. The operation can be hampered during cold weather by ice clogging of the intake or by ice jams restricting the flow of the stream. Since water treatment is seldom economical, scale and corrosion may be a problem. Intermittent chlorination may be used, however, to control the growth of slime and algae on heat exchanger surfaces. If the stream temperature becomes too high, it may be necessary to cool the effluent before discharging it into the stream.

Lakes or ponds may also be used as a source of cooling water.[2] The water temperature in lakes or streams tends to follow the average dry-bulb temperature of the surrounding air. The temperature gradient of a lake will vary with the season of the year. Applying a heat load will raise the average water temperature and decrease the temperature gradient from the surface to the bottom although stratification will remain. The cold water intake should, logically, be located as far below the surface of the water as practicable, and remote from the hot water return to prevent bypassing. Dikes or piers have been built into lakes to prevent bypassing where cold water intakes are adjacent to hot water returns, but they offer little advantage unless they extend into a lake for a distance of at least 300 ft.

Warm water discharged near the surface spreads out as a thin film. The temperature near the surface of a cooling pond will average about 4 F deg above the bottom temperature. A suction pipe located at the bottom of the pond will not withdraw water at the temperature of the lowest stratum. A hydraulic gradient is set up which withdraws water from all strata, and the intake temperature will be about 2 F deg below the surface temperature. The heat dissipation is a function of wind velocity and the temperature difference between water surface and the dry-bulb temperature. Fig. 1 is a typical performance chart for lake cooling, with wind velocity measured 5 ft above water level. Throne[1] has developed additional charts for other conditions. The heat dissipation of a



Reprinted from Power Magazine, September 1951, page 88.

Fig. 1 . . . . Performance Chart for Lake Cooling

lake increases from 0 to 10 Btu per (sq ft) (hr) (F deg temperature difference) of air and lake water surface, for duty up to 150 Btu per (sq ft) (hr) and wind velocities up to 6 mph with air temperatures varying from 20 to 80 F.

Spray Ponds

Heat is dissipated from the surface of a lake by evaporation, radiation, and convection. The spray pond divides the water into small drops, greatly extending the water surface, and brings it into intimate contact with the air.[1] The heat transfer is largely due to evaporative cooling as explained in the section Cooling Tower Theory. The driving force is the difference in enthalpy rather than temperature difference. The water temperature tends to approach the wet-bulb rather than the dry-bulb temperature of the air. This offers an inherent advantage in making it possible to cool the water to a temperature lower than the dry-bulb.

Cooling occurs in a spray pond as the water is propelled upward and then falls to the surface of the pond. The pond itself acts largely as a collecting basin. A typical nozzle arrangement is shown in Fig. 2. The design criteria in Table 1 provide a guide that can be used in laying out a pond.

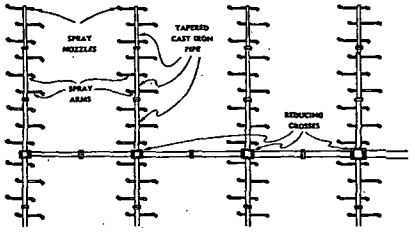


Fig. 2 .... Typical Nozzle Arrangement for a Spray Pond

The wind varies constantly in both direction and velocity. The wet-bulb temperature is usually changing also. These variations result in a continually changing temperature of the water as it strikes the surface of the pond. The pond acts as a reservoir so that the water temperature at the suction of the pump represents an average of past operating conditions. These characteristics make it difficult to test a spray pond or to develop ratings. As a result, spray ponds are not used when water temperatures must meet narrow limitations. They are usually designed for a 10 to 15 F deg cooling range and about a 10 deg approach of cold water temperature to the wet-bulb temperature.

The water leaving a spray nozzle will rise to a height of approximately 1 ft per psi of nozzle pressure. It will strike the surface of the pond in a circular pattern having a radius of about 10 ft. The cooling occurs within an *active filled volume* having a height equal to the elevation of the nozzles above the surface plus 1 ft per psi nozzle pressure, and a plan area extending 10 ft beyond the outer nozzles. The heat is transferred to the quantity of air passing through the *air area* which is the projected area of a vertical plane through the *active filled volume* and broadside to the direction of the air movement. The length of air travel is the horizontal distance the air moves through the *filled volume*.

Table 1 .... Spray Pond Design Data

Conventional Up-Spray System

	Units	Standard	Min	Max
Water capacity per nozzle	gpm	35 to 50	25	60
Nozzles per 12 ft length of pipe		6	4	6
Height of nozzles above water level	ft	6	5	12
Nozzle pressure	psig	6	5	7
Size of nozzles and nozzle arms	in.	2	1½	2
Distance between spray lateral piping	ft	25	13	38
Distance nozzles from pond side unfenced	ft	25 to 35	20	50
Distance nozzles from pond side fenced	ft	15 to 20	15	25
Height of louver fence	ft	12	12	12
Depth pond basin	ft	4 to 5	2	—
Friction loss allowed per 100 ft pipe	ft	1 to 3	—	—
Design wind velocity	mph	5	3	—

A reasonably accurate estimate of spray pond performance can be based on the outgoing wet-bulb temperature of the air passing through the filled volume. This temperature, obviously, cannot exceed the hot water temperature. It can approach the hot water temperature if it passes lengthwise through a long spray pond, but will have a small temperature rise when passing broadside through a narrow pond. The values in Table 2 can be used to obtain a reasonably accurate estimate of the cold water temperature with respect to the calculated outgoing wet-bulb temperature.

Wind passing through a pond carries away entrained water as drift. This creates a nuisance in the area on the leeward

Table 2 .... Degree Adjustments to be Applied to Leaving Air Wet-Bulb Temperature to Find Cooled Water Temperatures of Spray Ponds[a]

Cooling Range F deg	Entering Wet-Bulb[b] Temp. F	Adjustment in F Degrees		
		Length of Air Travel[c] (Feet)		
		100	50	25
10	80 WB	−3	+2	+4
	70 WB	−2	+3	+5
	60 WB	−1.5	+3.5	+5.5
15	80 WB	−5.0	+1	+5
	70 WB	−4	+2	+6
	60 WB	−3.5	+2.5	+5.5
20	80 WB	−7	0	+6
	70 WB	−6	+1	+7
	60 WB	−5.5	+1.5	+7.5

[a] Cooled-water temperature = wet-bulb temperature of leaving air plus the values shown.
[b] Wet-bulb temperature of air entering *spray-filled volume*. See text.
[c] Length of air travel through *spray-filled volume*.

side of the pond. This must be considered when selecting the location of the pond. The drift loss can be minimized by enclosing the spray pond within a louver fence 10 to 12 feet high. Fogging, which can occur during cold weather presents another hazard to buildings or roadways. A pond should be located with due regard for seasonal variations in wind directions to provide maximum cooling during the hot summer months, and minimum hazard to roadways during the winter months.

Lower water temperatures can be obtained from a spray pond than from a cooling lake, and a much smaller area is needed. This reduces the initial cost when a natural lake is not available. This saving in cost is offset by the cost of piping, supports and spray nozzles. Pumping costs may also be higher due to the back pressure imposed by the spray nozzles.

Atmospheric Wind Towers

An atmospheric spray-filled tower (Fig. 3) is essentially a spray pond with louvered walls and an elevated spray system

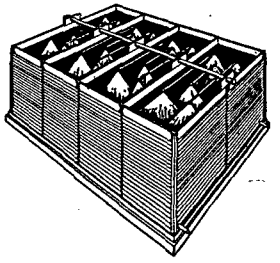


Fig. 3 .... Spray Filled Atmospheric Cooling Tower

which usually sprays downward. While the spray pond uses large nozzles handling 25 to 50 gpm each, the spray tower will have ¼ in. to ¾ in. nozzles handling 2 to 5 gpm each. Water loadings vary from 0.6 to 3.0 gpm per sq ft of plan area, and the heights vary from 6 to 15 ft. The design conditions are generally based on a 3 mph wind, but the down-spray nozzles have an aspirating effect that can induce a downward air movement of up to 400 fpm. Despite this positive air movement, wind is needed to carry the hot vapors away and prevent recirculation.

Atmospheric spray towers seldom exceed 50 ft in length. They frequently serve small refrigerating systems or cool jacket water of internal combustion engines. The performance can be estimated by calculating the outgoing wet-bulb temperature, as with the spray pond. The cold-water temperature will be approximately equal to the outgoing wet-bulb temperature, and performance can be estimated on that basis. An inadequate spray system will increase the water temperature several degrees. Spray towers require less basin area, less piping, and no more mechanical equipment than spray ponds. These savings may be largely offset by the extra cost of the structure.

The atmospheric deck tower contains wooden latticework decks regularly spaced from top to bottom. This filling interrupts the falling water, greatly increases the exposed water surface, and prolongs the time of contact with the air. Towers, such as shown in Fig. 4, are 20 to 50 ft high and 8 to 16 ft wide. The hot water is distributed at the top by a spray system or, more frequently, by open troughs. Except for the vagaries of the wind, the performance is quite predictable. This is done by the method described later for cross-flow mechanical-draft towers.

The cooling range of spray ponds or spray towers seldom exceeds 15 F deg and it is difficult to obtain less than a 10 F deg approach unless the cooling range is shorter than that. These performance limitations do not apply to the atmospheric deck tower. The design can control performance by variations in tower height, water loadings, and density of the filling.

The drift loss of a deck tower is considerable, and the nuisance is similar to that caused by the spray pond or spray towers. Both spray and deck towers are frequently located on building roofs, but the drift nuisance is seldom tolerated in congested areas. Water cooling systems that depend on the wind for air movement are not suitable for many services where the temperature requirements are more exacting.

Mechanical-Draft Towers

The mechanical-draft tower is equipped with fans to provide a positive and constant air flow. Since performance does not depend on the wind, it is possible to design mechanical-draft towers for exacting conditions. The fans may operate to provide forced or induced draft, depending on their location at the inlet or outlet of the tower. The tower may be cross-flow as shown in Fig. 5 or counter-flow as shown in Fig. 6. The addition of the fan makes it possible to design wider towers that are more compact than the long, narrow atmospheric towers. Early mechanical-draft towers were frequently of the spray-filled, forced-draft type. These spray-filled towers are rarely used at present except in congested areas where fire codes prohibit the use of wood filling.

The mechanical-draft tower is subject to recirculation of the exhaust vapors. These vapors will usually rise vertically on a still day and cause no trouble. Wind blowing across a

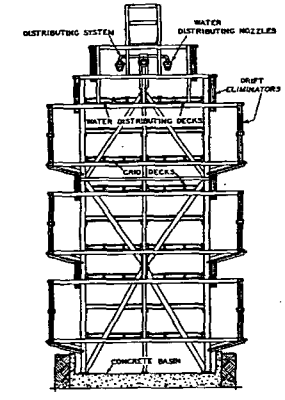


Fig. 4 .... Atmospheric Deck Tower

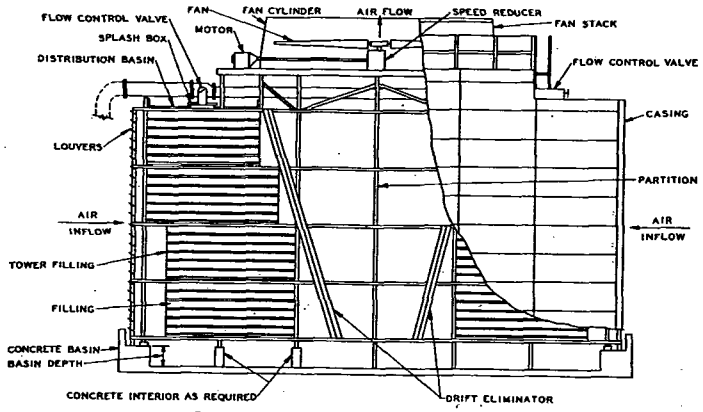


Fig. 5 .... Crossflow Mechanical Draft Tower

tower creates a low-pressure area on the leeward side. The exhaust vapors may then move downward and flow back into the tower. This increases the entering wet-bulb temperature and affects the performance accordingly. Recirculation is worse with forced-draft towers because of their low exit velocities. As a result, most towers built since 1940 are of induced-draft type. The forced-draft towers, however, have the advantage of locating the mechanical equipment away from the humid exhaust-air stream.

Air velocities usually vary from 300 to 400 fpm in counter-flow towers. The lower friction losses in cross-flow towers make it practicable to operate them at velocities up to 600 fpm. The water loading of counter-flow towers varies from 1.2 to 8 gpm per sq ft of plan area. The upper limit is largely determined by the blanketing effect of the distribution system. Since this blanketing is less pronounced in cross-flow towers, the loadings, based on plan area of the filling, can go up to 14 gpm per sq ft or higher.

Mechanical-draft towers require less plan area than ponds or atmospheric towers because of the higher water loadings. This is quite important in roof installations where the available space may be limited. This feature can be of great importance also in large industrial applications where space may be at a premium. Reduction in plan area will usually result in a saving in piping costs also.

Most manufacturers guarantee a drift loss that will not exceed 0.2 percent of the circulating rate, and the usual losses are probably less than that figure. The nuisance from drift is therefore much less with mechanical-draft towers.

COOLING TOWER THEORY

Basically, a water-cooling tower is a heat exchanger in which heat flows from the water to the air (1) by a flow of sensible heat from the warm water to the cooler air, and (2) by an exchange of latent heat resulting from the evaporation of a small part of the circulating water to increase the humidity ratio of the air by a corresponding amount. The general principles involved are similar to those encountered in the processes of diffusion in absorption and extraction equipment.[4, 5]

The generally accepted concept of cooling tower performance is based on enthalpy potential as the driving force. This is based on the relationship in an air-water mixture proposed by Lewis[6] showing that the numerical value of the coefficient of sensible heat transfer when divided by the coefficient of diffusion, approximately equals the specific heat of air at constant temperature. The relationship, involving three numerical values, each having different units, also explains why the wet-bulb temperature of a mixture of air and water

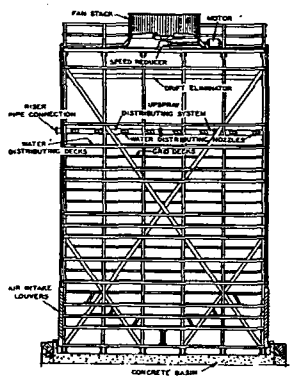


Fig. 6 .... Counterflow Induced Draft Cooling Tower

vapor approximates the temperature of adiabatic saturation. The relationship was applied by Merkel[7, 8] to combine the two coefficients into one overall coefficient based on enthalpy potential.

Details of the theoretical analysis have been published by various authors[9, 10, 11, 12] and those interested in the derivation of the equations should refer to these references. The derivations are based on the following assumptions:

1. Each particle of water is surrounded by a film of air that is at the temperature of the water and saturated with moisture.
2. The specific heat of water is unity at the temperatures encountered.
3. A constant wet-bulb temperature at varying humidity ratios represents constant enthalpy.
4. There is no weight loss of water due to evaporation during the cooling process.
5. The Lewis relationship applies.

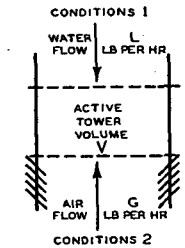


Fig. 7 .... Operations in a Typical Water Cooling Tower

Considering the cooling tower shown in Fig. 7 on the basis of these assumptions, the heat transfer per square foot of plan area in an incremental volume dV is

$$(G)(dh) = Ka(h'' - h_a)dV \tag{1}$$

Integration gives the familiar equation

$$\frac{KaV}{G} = \int_1^2 \frac{dh}{h'' - h_a} \tag{2}$$

It is more convenient to analyse cooling tower performance on the basis of water temperature, so

$$dh = \frac{L}{G}\,dt \tag{3}$$

and

$$\frac{KaV}{L} = \int_1^2 \frac{d\theta}{h'' - h_a} \tag{4}$$

where

- a = overall average wetted area (surface of water drops plus wetted tower surface), square feet per cubic foot of active tower volume.
- G = weight rate of flow of air, pounds of dry air per hour.
- h_a = enthalpy of air-vapor mixture, Btu per pound of dry air.
- h'' = enthalpy of saturated air-vapor mixture at water temperature, Btu per pound of dry air.
- K = overall energy unit conductance, Btu per (hour) (square foot overall average wetted area) (Btu enthalpy difference per pound of dry air).
- L = water rate, pounds per hour.
- θ = temperature of water in tower, Fahrenheit.
- θ_1 = temperature of inlet water, Fahrenheit.
- θ_2 = temperature of outlet water, Fahrenheit.
- V = active tower volume, cubic feet.
- Z = height of filled volume, feet.

Either Equation 2 or 4 applies, but Equation 4 is more commonly used to reduce performance conditions to a numerical value called the *Number of Transfer Units* (NTU). A *transfer unit* is related to a set of performance conditions which are such as to make the right-hand side of the equation equal unity. This is a condition in which the temperature change equals the mean driving force. The (NTU) evaluates the degree-of-difficulty by expressing the rate of temperature change per square foot of plan area per Btu of mean driving force.

The (NTU) applies to a vertical column of tower having one sq ft of plan area, a height Z and a volume V. Since V and Z are numerically equal

$$\frac{Ka}{L} = \frac{1}{Z}\int \frac{d\theta}{h'' - h_a} \tag{5}$$

This is the Unit-Volume Coefficient, obtained by dividing (NTU) by height. It evaluates the rate of temperature change per cubic foot of tower per square foot of plan area per Btu of mean driving force. This refers to temperature change without regard to the amount of water being cooled. Transposing L

$$Ka = \frac{L}{Z}\int \frac{d\theta}{h'' - h_a} \tag{6}$$

This is the overall coefficient and evaluates the Btu transferred per cubic foot of tower per square foot of plan area per Btu of mean driving force.

Equation 4 may be evaluated by a process of mechanical or graphical integration with the aid of a temperature-enthalpy diagram such as is shown in Fig. 8. The water enters the tower at temperature θ_1, having enthalpy h_1''. As it is cooled, the enthalpy of the air film surrounding it follows the saturation curve to temperature θ_2 where the enthalpy is

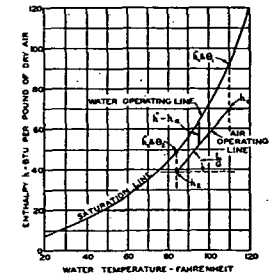


Fig. 8 .... Temperature-Enthalpy Diagram for Air-Water Vapor Mixture Showing Operating Lines for Example 2

Table 3 .... Sequence of Mechanical Integration Tower Performance

1 Water Temp. θ	2 Enthalpy of Film h''	3 Enthalpy of Air h_a	4 Enthalpy Diff. $h''-h_a$	5 $\frac{1}{h''-h_a}$	6 $\frac{\Delta\theta}{A\,(h''-h_a)}$	7 $\Sigma\,\frac{\Delta\theta}{h''-h_a}$	8 Cooling Range
84	48.22	38.61	9.61	0.1042			
					0.1042	0.1042	1
85	49.43	39.81	9.62	0.1041			
					0.1039	0.2081	2
86	50.66	41.01	9.65	0.1038			
					0.1034	0.3115	3
87	51.93	42.21	9.72	0.1030			
					0.1025	0.4140	4
88	53.23	43.41	9.82	0.1020			
					0.1013	0.5153	5
89	54.56	44.61	9.95	0.1006			
					0.1002	0.6155	6
90	55.93	45.81	10.12	0.0988			
					0.4640	1.0795	11
95	63.32	51.81	11.51	0.0868			
					0.3965	1.4760	16
100	71.73	57.81	13.92	0.0718			
					0.3220	1.7980	21
105	81.34	63.81	17.53	0.0570			
					0.2537	2.0517	26
110	92.34	69.81	22.53	0.0445			

h_1''. Air enters the tower at wet-bulb temperature t_1' having an enthalpy of h_1, which is plotted vertically below h_1''. Since the heat removed from the water equals the heat added to the air, the enthalpy increase per pound of dry air equals the temperature change of the water multiplied by the liquid/gas (L/G) ratio. The enthalpy of the air, therefore, increases along a straight line having a slope equalling L/G and terminating at h_2 which is vertically below h_1''. The driving force at any section of the tower is $h'' - h_a$, the vertical distance between the two operating curves.

Example 1: It is desired to cool 300 gpm (150,000 lb of water per hr) from 110 to 84 F with 3050 cfm (125,000 lb of dry air per hr) with an entering wet-bulb temperature of 75 F. Calculate the (NTU) showing the successive steps in tabular form.

Solution: The sequence of steps in the mechanical integration is shown in Table 3.

Water temperature is entered in Column 1 in even, but not necessarily equal, increments. One-degree increments are used from 84 to 90 F where the potential difference is small, and the effect of the reciprocal is large. Little accuracy is sacrificed by using 5-degree increments from 90 to 110 F where the potential difference is larger and its reciprocal smaller. Column 2 shows the enthalpy of saturated air corresponding to the temperatures in Column 1. Air enters the tower at a wet-bulb temperature of 75 F, having an enthalpy of 38.61 Btu per lb of dry air. This value is entered at the top of Column 3. The enthalpy of the air increases by $\Delta h = \frac{L}{G}d\theta$. The L/G ratio is $\frac{150{,}000}{125{,}000} = 1.2$ so Δh equals 1.2 Btu from 84 to 90 F and 6 Btu from 90 to 110 F. These increments are added to the initial enthalpy to obtain the successive values in Column 3. The enthalpy potential difference in Column 4 is the difference between Columns 2 and 3. This driving force appears in the denominator of Equation 2 so the reciprocal of Column 4 is entered in Column 5. The (NTU) within each increment equals the temperature change multiplied by the average of the entering and leaving values from Column 5. The products are entered in Column 6. The summation of Column 6 in Column 7 gives the (NTU) for the cooling ranges in Column 8. Hence, the mechanical integration reduces the given conditions to the numerical value of 2.0517 *Transfer Units*.

Example 2: The cooling tower designed to handle the conditions in *Example 1* has a fill height of 8 ft and a plan area of 96 sq ft. Calculate the unit-volume and overall coefficients and explain their significance.

Solution: The (NTU) of 2.0517 as determined in *Example 1* signifies the tower must cool the water 2.0517 deg per Btu of mean driving force. This cooling is accomplished in 8 ft of fill height, so the unit-volume coefficient is $\frac{2.0517}{8} = 0.2565$. This means that each cubic foot of tower must cool the water 0.2565 degrees per Btu of mean driving force. The water loading is $\frac{150{,}000}{96} = 1563$ lb per (sq ft) (hr). The overall coefficient is $0.2565 \times 1563 = 400.91$. This signifies that the tower must transfer 400.91 per cu ft (hr) (Btu of mean driving force).

Column 7 of Table 3 shows how the (NTU) increases with cooling range. A plot of Columns 7 vs. 8 can be used to determine the (NTU) for intermediate cooling ranges. This curve can also be used to determine temperature distribution because equal increments of transfer units correspond to equal increments of tower height. The linear relationship between (NTU) and height provides the basis of the procedure used to analyze crossflow cooling towers.[13]

Since the water temperature varies both horizontally and vertically in a cross-flow tower a double integration is required. This is accomplished by using equal increments of *transfer units*, representing equal increments of distance, and calculating the corresponding temperatures. Counter-flow integration progresses from the bottom of the tower upward. The cross-flow integration starts at the top of the air inlet, and progresses downward and inward.

Required and Available Coefficients

Any set of performance conditions can be accomplished by an infinite number of cooling tower designs. Calculating the (NTU) serves to evaluate the degree-of-difficulty of the problem. The *unit-volume* coefficient is more specific, but it also applies to an infinite number of towers of a given height. The *overall coefficient* has the most significance when comparing towers of different designs that are being considered for a given duty. It evaluates the effect of variations in air rate, height, and plan area on the *required* coefficient. A higher *required* coefficient indicates that more effective filling is needed. The effectiveness of a filling is determined almost entirely by the density of the splash surface and wetted surface per cubic foot of filled volume.

These calculations evaluate the problem, not the tower. The *available* coefficient of a specific cooling tower is determined from its operating characteristics. The tower must be tested at various conditions and the data from each test reduced to a corresponding coefficient. A cooling tower coefficient is not constant, but varies with operating conditions. Its characteristic is established by correlating variations of the coefficient with corresponding operating conditions.

The characteristic of a cooling tower closely approximates the relationship expressed by the general equation:

$$Ka = cL^nG^m \qquad (7)$$

The cooling tower manufacturer uses this procedure to determine the characteristic of each cooling tower design.

An examination of the steps taken in the solution of *Example 1* will show that various air rates could have been specified. Increasing the air rate would have reduced the degree-of-difficulty and resulted in a smaller (NTU) for a given set of conditions. Decreasing the air rate would have the opposite effect. Therefore, the *required* coefficient for a given set of conditions varies with the air rate. A tower will operate at a given set of conditions when the *required* coefficient equals the *available* coefficient as determined from Equation 7.

DESIGN CONDITIONS

The design conditions used in selecting a cooling tower are (1) the circulating water rate, (2) cooling range, (3) entering wet-bulb temperature, and (4) the approach of the cold-water to the wet-bulb temperature. These conditions are selected by considering the function of the cooling tower in the overall system. The system includes a process having a heat load that is to be transmitted to the circulating water through a heat exchanger. The cooling tower rejects the heat load to the atmosphere. Certain design conditions of the system affect the design of the cooling tower. They include (1) the heat load, (2) the temperature at which it is to be removed, and (3) the wet-bulb temperature of the atmosphere to which the heat is finally rejected. The cooling tower and heat exchanger are then selected to serve these conditions. The relationship of the cooling tower to the system is shown schematically in Fig. 9, with the process represented by two heat sources operating in series.

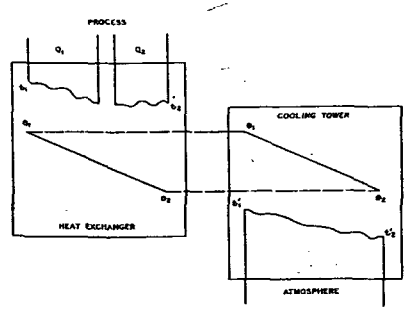


Fig. 9 .... Relation of Cooling Tower to Process

Heat Load

No definite sequence can be followed in selecting the design conditions because each item affects the choice of the others. The heat load comes closest to being a fixed quantity so can be considered first.

The amount of heat rejected by typical processes is shown in Table 4. The heat is rejected at a constant temperature when vapors are condensed, as illustrated by Q_1 in Fig. 9. A second type of heat load, represented by Q_1, consists of reducing the temperature of a fluid. Each type of heat load has specific requirements that affect overall design. The temperature at which the heat is removed from a process may be as important in the overall design as the heat load itself. As this temperature level decreases, the degree-of-difficulty increases at an accelerated rate, calling for larger and more expensive equipment. Higher temperatures usually result in higher operating costs, and loss of capacity in the process.

Table 4 .... Heat Rejection of Typical Processes

Equipment	Btu per min per ton	Btu per kw hr	Btu per bhp-hr
Refrigeration Compressors (reciprocating)	250		
Refrigeration Compressors (centrifugal)	300		
Refrigeration Absorption System	550		
Steam Jet Refrigerating System	550		
Steam Electric Power Plant:			
500 kw		11,210	
1000 kw		10,750	
5000 kw		8,150	
7500 kw		7,700	
10,000 kw		7,020	
Diesel Engine Jacket & Lube Oil:			
Four-cycle, Supercharged			2600
Four-cycle, Non-supercharged			3000
Two-cycle, Crank-case Compressor			2000
Two-cycle, Pump Scavenging (large unit)			2500
Two-cycle, Pump Scavenging (high speed)			2200
Natural Gas Engine:			
Four-cycle			4500
Two-cycle			4000

Table 5 . . . . Condenser Design Data

Gas	Desired Pressure in Condenser	Gas Temperature in Condenser, F	Leaving Hot Water Temperature, F	
			Best Condenser Design	Average Condenser Design
Steam	28 in. vac	101.2	97	93
Steam	27 in. vac	115.1	110	105
Steam	26 in. vac	125.4	120	114
Ammonia	185 psia*	96.0	92	88
Carbon dioxide	1030 psig*	86.0	83	80
Methyl chloride	102 psig*	100.0	96	92
Dichlorodi-fluoromethane	117 psig*	100.0	96	93
	126 psig*	105.0	100	97
	136 psig*	110.0	104	101

* Head pressure.

Condensing Systems

Each vapor has a pressure at which condensation occurs for any given temperature. This pressure increases with temperature up to the critical temperature of the vapor, above which no amount of increased pressure causes condensation. Condensation in a process occurs at constant temperature because a constant pressure will exist in the heat exchanger. The condensing pressure is usually the controlling factor in the process, although this is obtained by designing for the corresponding temperature. Table 5 lists typical condensing data for commonly used vapors.

A transfer of heat from the vapor to the circulating water requires a temperature differential between the two media. Since the vapor must condense at a constant temperature, the condensing temperature t_2 in Fig. 9 must be higher than the hot water temperature θ_1. It is for this reason that the condensing temperature is largely determined by the temperature of the warm water leaving the heat exchanger, and the entering cold-water temperature has only a small effect. The terminal temperature difference is controlled by the amount of heat transfer surface in the condenser. Economically sized condensers are usually designed to operate with 3 to 10 F deg temperature difference between the vapor and hot water.

Fluid Cooling

Different conditions exist when cooling a fluid such as milk, natural gas, or distillate. When this is done in a counter-flow heat exchanger, two terminal differences are involved. The fluid can be cooled to a temperature approaching that of the entering cold water, and the circulating water can be heated to a temperature approaching the hottest temperature of the fluid. A terminal temperature difference must exist at both the hot and cold end of the exchanger, but the temperature of the fluid leaving the exchanger can be colder than the hot water temperature.

Water Circulating Rate

The heat load on the system is determined by the process being served, and when this value is fixed, the circulating water rate determines the temperature rise in the heat exchanger. The cooling range in the cooling tower is obviously equal to the rise in the exchanger when the two are operating in equilibrium. The type of heat load and the characteristics of a cooling tower govern the selection of the optimum circulating rate. Increasing the circulating rate while holding the other operating conditions of a cooling tower constant, will reduce the cooling range. The cold-water temperature will increase, but the hot-water temperature will decrease to a greater extent. A high circulating rate and a short cooling range is advantageous when condensing vapors because the lower hot-water temperature has more effect on condensing temperature than the smaller increase in cold-water temperature. This advantage can be offset by greater power requirements for pumping, and possibly by the increased cost for larger piping.

Cooling a fluid presents a different problem. The cooling tower must deliver the cold water at a temperature that is below that of the cooled fluid, and this is more easily accomplished with a low water rate. The process may involve several heat loads, one of which requires a low temperature and the other can be removed at a much higher level. The size of a cooling tower is governed more by the cold-water temperature than by the cooling range. Once the cold-water temperature is fixed, it is desirable to increase the range as much as possible. A single cooling tower serving several heat loads in series is usually more economical than a number of individual towers.

A refrigerating system rejecting 250 Btu per ton imposes a load on the cooling tower of 30 (F deg) (gallons) per ton. The standard practice is to design the towers for 3, 4 or 5 gpm per ton, corresponding to cooling ranges of 10, 7½ and 6 F deg, respectively. Towers serving steam condensers usually operate with a range of 12 to 19 F deg. Oil refineries usually design for a cooling range of 25 F deg or more.

Heat Exchanger

The cooling tower operates in conjunction with the heat exchanger in transferring heat from the process to the atmosphere. Increasing the size of either one will reduce the required size of the other. The optimum conditions are determined by considering the initial and operating costs of several combinations of cooling towers and heat exchangers.

Process Characteristics

The type of process must be considered when selecting the design conditions of the system. This is particularly true of the temperature at which heat is removed from the process. Ammonia refrigerating systems are frequently designed to operate at 185 psig head pressure, corresponding to 96 F condensing temperature. This will require cooling water leaving the condenser at about 90 F. It will be impractical in most localities to select a cooling tower that will not exceed that temperature during brief periods. An increase in the temperature of the circulating water will raise the head pressure and increase the power requirement of the compressor. The increased head pressure also reduces the refrigerating capacity, but the system can be designed to perform acceptably at such times. A carbon dioxide refrigerating system represents a more serious situation. The critical temperature of the refrigerant is 88.4 F so the system will not function if that temperature is exceeded. The design conditions are also influenced by the process served by a refrigerating system. The maximum refrigerating load does not necessarily coincide with peak wet-bulb temperatures so a reduction in capacity may not be serious. That may not be true in the case of air conditioning if the maximum capacity is needed when wet-bulb temperature may be at its highest.

The efficiency, capacity and operating costs of a steam electric generating station all improve as the condensing temperature decreases. Designing for maximum performance at the most adverse conditions will result in a plant having excess capacity most of the time. The most adverse conditions occur during summer daylight hours, and peak power loads formerly came during winter evening hours. This gave the happy combination of peak demand during favorable operating conditions, and a low demand when the conditions were adverse. The power industry is profiting from the expansion of air conditioning, with the result that the peak load is beginning to coincide with the most adverse operating conditions. A generating plant can be designed to accept higher operating costs during such periods, but higher operating temperatures can reduce capacity below the required demand.

The capacity of an industrial operation such as an oil refinery will vary with the temperature of the cooling tower. Such a process may be so involved that it is not practicable to vary the output with fluctuation of the weather. These plants usually are compelled to operate at a constant rate that can be maintained from day to day. All processes using cooling water are faced with variations in water temperature. Higher temperatures increase operating costs and reduce capacity. These can be overcome by installing a larger plant or a larger cooling system, with the result that one portion of the equipment will have excess capacity during most of the year. The overall design must be based on (1) the minimum permissible capacity under the most adverse conditions and (2) an economic study showing the effect of changes in the size of the components on initial cost and operating expense.

Wet-Bulb Temperature

The wet-bulb temperature is beyond the control of the designer. It fluctuates during the day, and throughout the year. The annual averages should be studied and correlated with the characteristics of the process. Peak wet-bulb temperatures are usually of short duration so have little effect on annual operating costs, but the effect of these conditions in limiting capacity must be considered. The design wet-bulb temperature should be selected on the basis of the number of hours of the four summer months of June through September (or of the year) that it can be exceeded during an average year. A common practice in air conditioning is to select a wet-bulb temperature that will not be exceeded more than 5 percent of the time during the four summer months. Various groups are cooperating in compiling weather data for this type of study. A limited portion of the data are tabulated in Table 2, Chapter 13. More complete summer weather data, statistics, charts, maps, and technical analysis have been prepared by Albright.[13]

When the circulating rate and water temperatures have been selected, the size of the cooling tower will be increased by selecting a higher wet-bulb temperature. Selecting the design wet-bulb temperature makes it possible to consider various combinations of cooling towers and heat exchangers. The overall design will, of necessity, consider the worst possible conditions. This will correspond to the maximum wet-bulb temperature for the locality, and the highest permissible process temperature. All components should be selected with these conditions in mind, but testing will be easier if the same size equipment is designed and guaranteed around a lower wet-bulb temperature. Table 6 shows that an ammonia refrigerating system might be designed for a 78 F wet-bulb temperature. That temperature will prevail 12 hours a summer, and for brief periods of time. Designing the same equipment around a lower wet-bulb temperature greatly increases the amount of time that tests can be conducted at or near the design conditions.

Table 6 . . . . Alternate Design Conditions for Ammonia Refrigerating System, Showing Annual Hours the Design Wet-bulb Temperature is Attained

Wet-Bulb Temp. F	Hot Water Temp. F	Head Press. psi	Hours Attainable
78	92.0	185	12
75	88.5	174	161
70	86.0	166	750

SELECTION AND EVALUATION

The characteristics of the various types of water cooling equipment will influence the final selection. The requirements of the process, and physical conditions of the plant will impose further restrictions. These factors will determine the design conditions to be used and indicate the general type of equipment that will be most desirable. A wide variety of cooling towers will, nevertheless, be capable of meeting the design conditions. Competitive designs may be similar, but they will not be identical. The same design can be considered in various frame sizes, representing different plan areas or heights. An increase of frame size will enable a tower to meet the conditions with less total air flow, and will reduce horsepower requirements. As a general rule, an increase of frame size increases the initial cost but reduces operating costs. Optimum size is determined from an economic evaluation that will include such items as the following:

CAPITAL INVESTMENT

1. Total erected cost of cooling tower.
2. Basin, grillage, and value of the space occupied.
3. Cost of pump and prime mover.
4. Electric wiring to pump and fan motors.
5. Electrical controls and switch gear.
6. Cost of piping from tower inlet to point on tower nearest to plant.
7. Sump, screens, overflow, and make-up lines not furnished by manufacturer.
8. Shut-off and control valves not furnished by manufacturer.
9. Walkways, ladders, etc., providing access to tower.

OPERATING COSTS

1. Demand charge, if it enters into cost.
2. Energy costs for fans and pumps, based on 3000 to 7000 hr per yr.
3. Amortization of investment, over a period varying from 3 to 30 years.
4. Maintenance and repairs.

Other factors to consider are safety features, conformity to building codes, general design and rigidity of structures, availability of spare parts, experience and reliability of manufacturer, and operating flexibility for economical operation at varying loads or seasonal changes. Failure to perform in accordance with the guarantee usually proves costly to the

purchaser regardless of subsequent adjustments made by the manufacturer. The capability of a cooling tower should be evaluated prior to purchase, and acceptance tests conducted during the warranty period.

TOWER LOCATION

Small cooling towers are frequently located on the roof or offset of a building, but large towers must be located on the ground. Proper location and orientation are needed in either case to insure satisfactory performance. A most important factor is to provide for a free flow of air to and from the tower. The air flow to a tower can be obstructed by adjacent structures or by screening walls in the case of roof installations. Such walls must be provided with openings that are at least equal to the inlet area of the tower. If the opening in the wall is not immediately adjacent to the tower inlet, the wall must be set back far enough to allow the air to flow around or under the tower to the inlet.

Downdrafts are created by wind blowing over a cooling tower or adjacent structures. The exhaust air is carried downward and can re-enter the tower from the leeward side. Since such recirculation raises the entering wet-bulb temperature a location should be selected that results in the least loss of performance during the hottest weather. Recirculation is usually the least with an end wind unless the tower is over 300 ft long. Adjacent structures and topography affect the wind. They should be considered carefully together with U. S. Weather Bureau data for the locality. A cooling tower should not be located where the prevailing summer wind will carry heat to it from an external source or from another cooling tower.

A properly designed cooling tower will have a drift loss not more than 0.2 percent of the circulating rate. A small amount of water in the exhaust air is unavoidable, but it can be objectionable if the droplets fall on parked automobiles, nearby windows or power lines. Evaporation leaves a deposit of the dissolved solids.

The moisture content of the air passing through a cooling tower is increased due to the evaporative cooling of the water. Condensation is unavoidable if the vapors strike a cool surface and cause a hazard that can be serious during freezing weather. During periods of high relative humidity, fogging occurs as the exhaust air mixes with the atmosphere and is cooled below its dew point. Fogging is usually worse in the spring or fall when the ambient air is humid and cool but not too cold. Warm air during the summer is capable of absorbing the excess moisture without producing a fog. Less water evaporates in the tower during extremely cold weather because the humidity ratio is reduced, so the fog is dissipated rather rapidly.

The drift nuisance is usually confined to an area immediately adjacent to the tower although high winds can carry it several hundred feet at times. Fogging, however, is less localized and can occur close to the tower or settle to the ground a considerable distance away.

Noise problems occur when cooling towers are located in congested areas. This applies particularly to small towers rather than large industrial installations. The sound is created by the falling water or fans and air movement. The water noise does not carry far but is quite noticeable adjacent to a tower. The sound of the fans may be audible at a considerable distance away from the tower if it is not deflected by intervening structures. Deflecting screens are of some value, and acoustical treatment is quite effective but expensive. It is commonly believed that objectionable noise can be eliminated by a reduction in the tip speed of the fan. This is seldom successful because the sound is a function of horsepower as well as tip speed. Both of these must be reduced to lower the sound level.

OPERATION AND MAINTENANCE

Most cooling tower manufacturers issue operating and maintenance instructions. These bulletins should be placed in the hands of operating personnel for their guidance.

Water Treatment

The general discussion of this subject in Chapter 55 offers detailed information on scale deposition, corrosion, and control of biological fouling. The dissolved solids in a circulating system are concentrated as water is lost by evaporation. The evaporation averages 0.8 percent of the circulating rate for each 10 F deg of cooling range. Excessive concentration is prevented by wasting a portion of the circulating water as blowdown. The make-up water, replacing losses due to evaporation, drift and blowdown, introduces dissolved solids into the system. An equal amount of solids must be removed by the drift and blowdown. Expressing all terms as a percentage of the circulating rate, the number of concentrations in the system is:

$$\text{No. of Conc.} = \frac{\text{Evaporation} + \text{drift} + \text{blowdown}}{\text{Drift} + \text{blowdown}} \quad (8)$$

The form of Equation 8 shows that a small amount of blow down is highly effective, but higher rates are of little advantage. Drift loss alone will maintain about 15 concentrations. This is rapidly reduced to 3 or 4 concentrations with 1 percent blowdown, but an increase to 100 per cent blowdown brings the system down to one concentration. Draining the system periodically is a poor substitute for continuous blowdown, because the concentration builds up to the maximum within a few hours.

Towers operating continuously are usually arranged for blowdown by maintaining a water level in the collecting basin that allows the desired quantity to be wasted at the overflow connection. Placing the blowdown connection in the hot-water line adjacent to the tower inlet prevents unnecessary waste when the pump is stopped. This is recommended for intermittent operation.

Many small installations rely on blowdown for water conditioning, and use no supplementary water treatment. Larger installations find it desirable to use chemical treatment in the system, and most of these processes require a controlled blowdown rate. Chapter 55 discusses the chemicals used to control scale, corrosion and biological fouling.

Wood Deterioration

Wood is one of the most durable materials, but it can be destroyed by chemical action or by organisms producing decay. Until 1950, the deterioration of cooling tower lumber was looked upon as the result of chemical action, but later investigations indicate it is largely due to micro-biological attack.[14] Deterioration can be expected if sodium carbonate is present in the water. This was formerly classed as a chemical attack although sodium carbonate is a rather poor pulping agent. The deteriorated wood almost invariably contains wood-destroying organisms, so the actual cause of the deterioration can be chemical, micro-biological or a combination of the two.

Chlorine is frequently used to control the growth of slime and algae that foul the circulating system. This is a powerful oxidizing agent that attacks the wood if used excessively. The action bleaches the surface of the wood and loosens the fibers, allowing them to wash away and collect in the heat exchangers or on the screens. A more common type of surface attack, however, is caused by soft rot organisms. The more durable species of wood contain fungicidal materials that are removed by oxidizing agents such as chlorine. These materials are also removed by normal leaching.

Decay organisms are widespread in nature, serving a useful and necessary purpose in sewage disposal systems or in the disintegration of dead trees in the forest. The same process is highly undesirable when it occurs in structural lumber. It is prevented by impregnating the lumber with preservatives that are toxic to the organisms. The use of treated lumber in cooling towers has been increasing since 1953. Service records accumulated since that time indicate that the service life of cooling tower lumber can be economically extended by preservative treatment. The use of treated wood increases the initial cost of a cooling tower about 10 percent. Preservatives can be applied by the double diffusion method to existing cooling towers built of untreated lumber.[14]

Winter Operation

The basin and piping must be drained, or otherwise protected, if a cooling tower is shut down during freezing weather. Towers that are shut down frequently are preferably operated with a *dry basin*. The tower is installed at an elevation that allows the basin to drain into an indoor tank to which the circulating pump is connected.

Water-cooling towers operating in freezing weather are subject to ice formation, particularly at the air inlet. This is not influenced by the temperature of the circulating water since the icing occurs where fine drops splash out and strike relatively dry surfaces in the entering air stream. Stopping the fans temporarily allows the water to fall vertically across the inlet, and much of the ice is melted. Ice that is not melted in this manner can be removed by reversing the fans for 15 or 20 minutes. Air flow is obstructed by the ice, causing water temperatures to increase drastically. Since the water temperature increases when icing occurs, it is not possible to use thermostatic controls to stop or reverse the fans.

CONTROL OF COOLING TOWERS

Other than the usual electrical interlocks with the refrigeration system, few automatic controls are used on cooling towers. Sometimes the fan is turned off automatically to reduce capacity in cool weather. Towers not of the dry-basin type may have drain valves controlled by outdoor thermostats to prevent freeze-up of the water in the pan.

EVAPORATIVE CONDENSERS

The evaporative condenser combines the functions of the condenser and cooling tower by using a minimum amount of water on a surface, cooling it to approximately the wet-bulb temperature of the surrounding atmosphere.

The end view of a typical evaporative condenser is shown in Fig. 10. The fan draws the air over a condenser surface which is kept wet by a water spray. The discharge refrigerant gas from the compressor enters the top of the condenser coil. The liquid refrigerant is drained from the bottom of the coil into a liquid receiver, and then circulates through the remaining portion of the system in the usual way.

The water is circulated through the spray nozzles and drops into the sump where its level is maintained by means of a float valve. The eliminator plates are placed in the path of the water-air mixture so as to remove the entrained water. Since the air leaving the unit is almost completely saturated, care must be taken in locating discharge ducts to prevent condensation.

Evaporative condensers are available in sizes up to 300 tons or more. These units use only a small portion of the water required for a waste-water cooled condenser. The water is vaporized by the heat of the refrigerant. Each pound of water used extracts approximately 1000 Btu from the refrigerant, whereas under commonly used rating conditions where the water temperature rise is 20 F deg, each pound of water extracts only 20 Btu from the refrigerant. The water used including the loss by entrainment in the discharge air, the blow down, and the evaporation, amounts to about 3 to 5 percent of that which would be required for a water-cooled condenser.

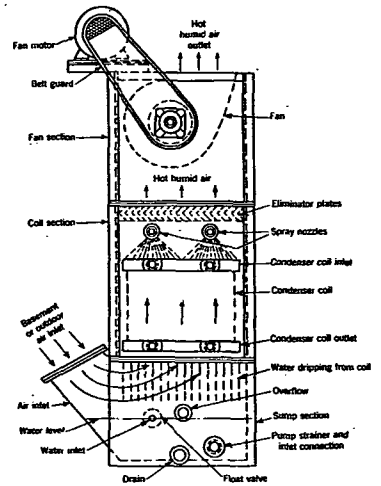


Fig. 10 .... Schematic View of an Evaporative Condenser

The evaporative condenser requires more maintenance, occupies greater space (must be located where air is available), and has a higher first cost than the water-cooled condenser, but where the use of water is restricted or expensive, the evaporative condenser has become widely accepted. Compared with a water-cooled condenser and cooling tower, which have about the same heat dissipating capacity, the evaporative condenser has the advantage of lower cost and smaller space requirements.

Control of Evaporative Condensers

An electrical interlock is usually provided to prevent the refrigeration system from operating until the pump and fan

of the evaporative condenser have been started. In some cases, an automatic damper is used on the inlet of the evaporative condenser to provide proportional control of head pressure on the refrigeration system. See Chapter 43 for general information on controls.

REFERENCES

[1] W. D. Collins: *Temperature of Water Available for Industrial Use in the United States* (U. S. Geological Survey, *Water Supply Paper* No. 520F).

[2] R. F. Throne: Cooling performance of an industrialized lake surface (*Power*, September 1951).

[3] S. Hori, U. A. Patchett, and L. M. K. Boelter: Design of spray cooling ponds (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October 1942, p. 624).

[4] W. H. Walker, W. K. Lewis, W. H. McAdams, and E. R. Gilliland: *Principles of Chemical Engineering* (McGraw-Hill Co., New York, 1937, p. 480).

[5] D. Q. Kern: *Process Heat Transfer* (McGraw-Hill Co., New York, 1950, p. 563).

[6] W. K. Lewis: The evaporation of a liquid into a gas (*ASME Transactions*, Vol. 44, 1922, p. 325).

[7] F. Merkel: Verdustungs kuhlung (*Forschungarbeiten*, No. 275, 1925).

[8] H. B. Nottage: Merkel's cooling diagram as a performance correlation for air-water evaporative cooling system (ASHVE TRANSACTIONS, Vol. 47, 1941, p. 429).

[9] W. H. McAdams: *Heat Transmission* (McGraw-Hill Co., New York, 1933, p. 157).

[10] A. L. London, H. B. Nottage, and L. M. K. Boelter: Determination of unit conductance for heat and mass transfer by the transient method (*Industrial and Engineering Chemistry*, April 1941, p. 467).

[11] Joseph Lichtenstein: Performance and selection of mechanical-draft cooling towers (*ASME Transactions*, 1943, p. 779).

[12] D. R. Baker and L. T. Mart: Analysing cooling tower performance by the unit-volume coefficient (*Chemical Engineering*, December 1952, p. 196).

[13] J. C. Albright and D. R. Baker: *Summer Weather Data* (The Marley Co., 1944).

[14] D. R. Baker: Studies relative to protection of cooling tower lumber (*ASME Oil, Gas and Power Division Proceedings*, June 1954).

CHAPTER 41

EVAPORATIVE AIR COOLING AND HUMIDIFICATION

Apparatus for Humidification, Spray Generation, Spray Distribution, Unit Humidifiers, Air Washers, Humidification with Air Washers, Dehumidification and Cooling with Air Washers, Evaporative Air Cooling, Types of Direct Evaporative-Cooling Equipment, Design Considerations, Controls, Maintenance

EVAPORATIVE air cooling and humidification increase the moisture content of an air stream or the air in a selected space, with or without a change in dry-bulb temperature.

Evaporative air cooling applies the well-known phenomenon of evaporating water at ambient temperature into an air stream. If no heat is added during the process, the dry-bulb temperature of the air stream is reduced along the thermodynamic wet-bulb temperature line with a consequent increase in latent heat and moisture content.

Humidification, used industrially and in residential air-conditioning and heating systems, may be accomplished by evaporative air-cooling apparatus, by the direct introduction of water spray into the air stream or space either by heated evaporating pans or steam jets. Some equipment can be used for both cooling and humidification.

APPARATUS FOR HUMIDIFICATION

Humidification may be direct or indirect, i.e., water in droplets or vapor form may be introduced directly into the humidified space, or previously humidified air may be circulated in or through the space. Evaporative air cooling is always indirect although the cooling may be accomplished by wetted surface or secondary cool surface.

The classification of apparatus used for humidification and evaporative air cooling is shown in Table 1.

HUMIDIFICATION SPRAY GENERATION AND DISTRIBUTION

Spray generation is obtained by (1) compressed air, (2) impact, (3) hydraulic separation, and (4) mechanical separation. Atomization can be accomplished by the use of a compressed air jet to reduce the water particles to a fine spray. With the impact method, a jet of water under pressure impinges directly on the end of a small round wire. Where hydraulic separation is employed, a jet of water enters a cylindrical chamber and escapes through an axial port with a rapid rotation which causes it to separate in a fine cone-shaped spray. In the mechanical separation process, water is thrown by centrifugal force from the surface of a rapidly revolving disc and separates into particles sufficiently small to be utilized in certain types of mechanical humidifiers.

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

Air-jet distribution utilizes the atomizing jet to distribute the spray through the space. Induction distribution utilizes the aspirating effect of an impact or centrifugal spray jet to induce air flow through a casing or duct for distribution of the spray. Fan propulsion uses a power-driven fan to distribute the spray.

Atomizing Humidifiers

In several types of atomizing humidifiers nozzles are placed within the room. Compressed air effects complete atomization of the water which is converted into vapor by the heat of the room air. In some of these nozzles an aspirating effect draws the water into the nozzle and atomizes it. Others are operated on a combined air and water pressure. It is usual for nozzles of the water-pressure type to be controlled by a diaphragm valve actuated by the pressure of the atomizing air.

High-Duty Humidifiers

Water is supplied at about 150 psig to an impact-type spray-generating nozzle located in a cylindrical casing. A drainage pan provides for the collection of unevaporated water which is returned to a filter tank. A powerful air current from a fan mounted above the unit, flows through the cylinder and is charged with moisture. Air discharges from an opening below at a high velocity in a complete and nearly horizontal circle. The spray is evaporated and the resulting vapor is diffused and distributed.

Spray Humidifiers

This type uses a spray nozzle in a cylindrical casing equipped with a drainage pan. The aspirating effect of the nozzle induces a moderate air current through the casing which distributes the entrained spray. The general method of circulating and returning the water is similar to that employed for high-duty humidifiers. A suitable pump and centrally-located filter tank are required.

Self-Contained Humidifiers

In this type mechanical or centrifugal means are used to generate and distribute spray. They may be used either singly or in groups. In large installations, where suitable connections are provided to permit the cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

UNIT HUMIDIFIERS

The term unit humidifier denotes an assembly of elements the principal function of which is to humidify. The essential

element of a unit humidifier is an atomizer or evaporator. To this may be added a fan, a heater, outlet vanes or diffusers, and a housing to enclose the various parts.

Unit humidifiers fall into four general classifications, depending on the method of causing evaporation. These are as follows: (1) *Nozzle Type*, (2) *Rotary Type*, (3) *Cascade Type*, and (4) *Heater Type*.

In the *nozzle* type of humidifier water is sprayed into the air and evaporation is effected by adiabatic exchange of energy. Units of this type in simplest form, spray a fine mist of water directly into the air in a space. They are used to a great extent in the textile industry.

The *rotary* type of humidifier generates and distributes by mechanical separation using rotating discs. In all other respects, this type of humidifier is similar to the nozzle type. It has the advantage over the nozzle type of being less likely to become clogged.

In the *cascade* type the humidification takes place by water falling in sheets over a series of baffles or trays. This type is usually furnished with a fan, air heater, and air filter, all enclosed in a housing.

In the *heater* type of humidifier the water is heated either to the boiling point or to a temperature at which the water vapor readily passes into the air stream. In the simplest form the heating element using steam, hot water, gas, oil, or electricity is placed in a pan or vessel of water, and the vapor passes from the surface of the water to the stream of air.

A modification of this type of humidifier is the combination of spray nozzle and heater type in which the water is sprayed over a hot surface and evaporated. It has the disadvantage of accumulating scale on the surfaces of the vessel or heating surface.

Unit humidifiers are usually controlled by a two-position room humidistat operating a controlled device which functions to start and stop the humidifying action. This may be a valve on the water supply to the humidifier. The fan, if used, normally is also turned off by the humidistat. The controlled device operates the source of heat in the heater type of humidifier. See Chapter 43 for a general discussion of humidity controls.

AIR WASHERS

Spray-Type Washers

A spray-type air washer consists essentially of a chamber or casing containing a spray nozzle system, a tank for collecting the spray water as it falls, and an eliminator section at the discharge end for removal of entrained drops of water from the air. A pump recirculates water at a rate greatly in excess of the evaporation rate. Intimate contact between the spray water and the flowing air causes heat transfer between the air and the water, resulting in either humidification or dehumidification, depending upon the method of operation and the relative temperatures of air and spray water.

To prevent backlash of spray ahead of the washer chamber, and to provide more uniform air distribution, perforated inlet plates, horizontal louvers or vertical eliminator baffles may be provided in the washer inlet. Inlet diffusion plates may be required when the air and water spray are in the same direction because eddy currents may be present. Eliminator baffles are used where one or more of the water sprays opposes the air flow and is sufficiently close to the entering end that water might come out. At the outlet end of the washer, suitable eliminator plates are placed and may or may not be provided with flooding devices. These plates or blades, for the removal of entrained moisture, may cause four to six directional air changes and present as many as four effective hooked edges for trapping the water. Blades usually have bends at 30 deg to the direction of air flow. Sharper bends add substantially to the resistance without improving the elimination materially. Vertical type baffles may be provided with flooding devices where the air contains large quantities of particulate matter or lint, as in cotton mills, to prevent build-up on the baffle blades.

Table 1 . . . . Classification of Apparatus Used for Humidification and Evaporative Air Cooling

Designation	Humidification		Evaporative Air Cooling		
	Indirect	Direct	Primary Wet Surfaces	Dry Surface or Secondary Cool Surface	Two Stage
Residential	1. Evaporating pans with "wick action" water transport. 2. Externally heated evaporator pans (electric or steam heat). 3. Porous, extended surface pad types.	1. Atomizers or sprays.	1. Drip type, usually nonmetal pads. 2. Slinger type, usually metal pads. 3. Rotary type, usually metal screens.	1. Blast coils cooled by cooling tower water.	
Commercial or Industrial	1. Air washers, spray and fill types.	1. Pneumatic atomization. 2. Pressure atomization. 3. Centrifugal atomization. 4. Steam jets.	1. Drip type, usually non-metal pads. 2. Slinger type, usually metal pads. 3. Spray or washer type. 4. Rotary type, usually metal screens.	1. Blast coils cooled by cooling tower water.	1. Secondary cool surface cooling followed by primary wet surface or spray cooling.

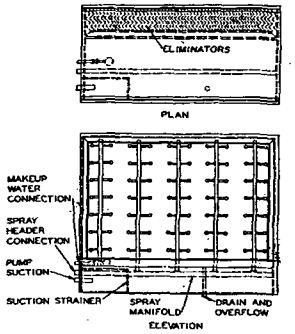


Fig. 1 . . . . Typical Single-Bank Air Washer

Figs. 1 and 2 show the essential construction features of conventional spray-type air washers. Intimate contact between the air and water is secured (1) by breaking the water into fine drops, (2) by passing the air over the surfaces continuously wetted by water, or (3) by a combination of the two.

Essential requirements in the air washer operation are: uniform distribution of the air across the spray chamber, air velocities of from 400 to 700 fpm in the washer chamber, an adequate amount of spray water broken up into fine droplets at pressures of from 15 to 30 psig, good spray distribution across the air stream, sufficient length of travel through the spray and wetted surfaces, and the elimination of free moisture from the outlet air.

Expected performance, physical size, length, number of sprays, etc., vary greatly depending on the functions of the

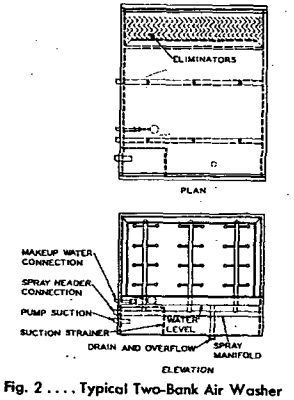


Fig. 2 . . . . Typical Two-Bank Air Washer

installation. In general, the width and height of an air washer will depend on the space available. Washers of nearly equal height and width are desirable from an air flow and economic standpoint, although these proportions are not necessary. The length of washers varies considerably. A space of approximately 2½ to 4½ ft between spray banks is used, and the first and last banks of sprays are located about 1 to 1½ ft from the entering or leaving end of the washer. In addition, air washers may be furnished with heating or cooling coils within the washer chamber. These coils may affect the overall length of the washer.

Spray water requirements for spray-type air washers used for washing or evaporative cooling vary from 4 gpm with a single bank to 10 gpm for double banks per 1000 cfm. Pumping heads will usually be in the range from 55 to 80 ft of water

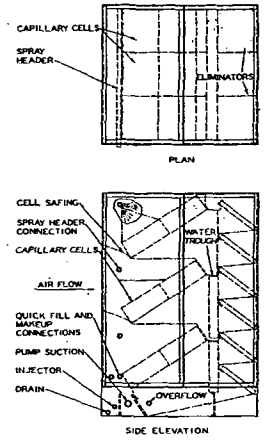


Fig. 3 . . . . Typical Fill-Type Air Washer

depending on spray pressure, height of apparatus, pressure losses in pipe and strainers, etc.

The resistance to air flow through an air washer varies with the type and number of baffles, eliminators, and wetted surfaces; number of spray banks and their direction; air velocity; size and type of other resistances such as cooling and heating coils; and other factors such as air density. Resistance may be as low as ¼ in. or high as 1 in. water column. It is therefore necessary that the manufacturer be consulted with regard to the resistance of any particular washer design combination.

Fill-Type Washers

Other air washer types achieve contact between air and water by passing the air through cells packed with glass fibers or sheets of specially designed media. See Fig. 3. These washers have the cells arranged in tiers over which water is distributed at low pressure. The cells are followed by conven-

tional blade type or glass mat eliminators. With water drainage provided for each cell tier, most of the stratification found in spray-type washers is avoided.

An essential requirement of operation of this type of air washer is good distribution of water over the face of the cells. When compared with conventional spray-type washers, the water requirements are much lower, approximately 4 gpm per 1000 cfm for ordinary washing and evaporative cooling, and the pumping head is approximately one-half the head of conventional spray-type washers.

Humidification with Air Washers

Air humidification can be accomplished in three ways with an air washer. These are: (1) use of recirculated spray water without prior treatment of the air; (2) preheating the air and washing it with recirculated spray water; and (3) using heated spray water. In any air washing installation the air should not enter the washer with a wet-bulb temperature less than 35 F. This is a precaution to eliminate the danger of freezing the spray water.

Method 1. Except for the small amount of energy added from outside by the recirculating pump in the form of shaft work, and for the small amount of heat leakage from outside into the apparatus, including the pump and its connecting piping, the process would be strictly adiabatic. Evaporation from the liquid spray would therefore be expected to bring the air immediately in contact with it to saturation adiabatically; and, since the liquid is recirculated, its temperature would be expected to adjust to the thermodynamic wet-bulb temperature of the entering air.

It does not follow from the foregoing reasoning that the whole air stream is brought to complete saturation, but merely that its state point should move along a line of constant thermodynamic wet-bulb temperature as explained in Chapter 3. The extent to which the leaving air temperature approaches the thermodynamic wet-bulb temperature of the entering air, or the extent to which complete saturation is approached, is conveniently expressed by a ratio known as humidifying effectiveness or saturating effectiveness, and is defined in Equation 1 as

$$e_s = \frac{t_1 - t_2}{t_1 - t'} \times 100 \qquad (1)$$

where

- e_s = humidifying or saturating effectiveness, percent.
- t_1 = dry-bulb temperature of the entering air, Fahrenheit.
- t_2 = dry-bulb temperature of the leaving air, Fahrenheit.
- t' = thermodynamic wet-bulb temperature of the entering air, Fahrenheit.

The following may be taken as representative of the humidifying or saturating effectiveness of a spray-type air washer for the arrangements stated:

Arrangement	Length Ft	Effectiveness—%
1 bank downstream	4	50-60
1 bank downstream	6	60-75
1 bank upstream	6	65-80
2 banks downstream	8 to 10	80-90
2 banks opposing each other	8 to 10	85-95
2 banks upstream	8 to 10	90-98

The humidifying or saturating effectiveness of a washer is dependent upon the essential items of design mentioned in the section Air Washers. Other conditions being the same, a low velocity of air flow is conducive to high humidifying effectiveness.

Method 2. The preheating of the air increases both the dry- and wet-bulb temperatures, and lowers the relative humidity, but does not alter the humidity ratio (pounds of water vapor per pound dry air). At a higher wet-bulb temperature, but the same humidity ratio, more water can be absorbed per pound of dry air in passing through the washer, assuming that the humidifying effectiveness of the washer is not adversely affected by operation at the higher wet-bulb temperature. The analysis of the process occurring in the washer itself is the same as that explained under *Method 1.* The final desired conditions are secured by adjusting the amount of preheating to give the required wet-bulb temperature at the entrance to the washer.

Method 3. Even if the heat is added to the spray water, the mixing occurring in the washer itself may still be regarded as adiabatic. The state point of the mixture should move in a direction determined by the specific enthalpy of the heated spray as explained in Chapter 3. It is possible, by elevating the water temperature, to raise the air temperature, both dry-bulb and wet-bulb, above the dry-bulb temperature of the entering air.

In each of the methods, 1, 2, or 3, the air leaving the air washer may require reheating to produce in the conditioned space the required dry-bulb temperature and relative humidity.

Dehumidification and Cooling with Air Washers

Lowering of the wet-bulb temperature of an air-vapor mixture can be accomplished by an air washer if the temperature of the spray water is lower than the wet-bulb temperature of the air. Moisture removal is obtained when the spray water temperature is lower than the dew point of the entering air. In these cases the final dry-bulb temperature and relative humidity of the leaving air are dependent upon the design factors of the air washer.

Both sensible and latent heat are removed in the process of dehumidification by cold spray water. Extraction of sensible heat occurs during the entire time that the air is in contact with the spray medium. Latent heat removal takes place as condensation occurs. Therefore, the lower the spray temperatures, the greater the amount of moisture removal per pound of dry air, all other conditions remaining the same.

Where a limited supply of cold water is available, multiple-stage washers may be used to great advantage. In such washers the cool water is pumped through the multiple spray systems in series and counterflow to the air flow. Such an arrangement brings the delivery air in contact with the coldest water, securing a maximum amount of cooling and saving water.

When using cold well water or water from city water mains, care should be used to secure accurate data on the water temperatures. Table 2 lists some approximate water main averages which may be used as a guide but should be verified from local records. This is particularly true with city water main temperatures. In the case of well water temperatures, Fig. 4 shows the approximate temperatures of water to be expected from wells at depths of 30 to 60 ft.

Air washers for dehumidifying and cooling usually have separate recirculating pumps for each washer. These pumps deliver a mixture of cold and recirculated water under the control of a three-way valve.

Table 2 .... Average Maximum Water Main Temperatures*

State	City	Temp. F	State	City	Temp. F	State	City	Temp. F	State	City	Temp. F
Ala.	Birmingham	84	Iowa	Cedar Rapids	78	N. Y.	Albany	68	Texas	Amarillo	70
	Mobile	80		Des Moines	77		Buffalo	78		Austin	90
Ark.	Little Rock	83		Sioux City	62		Jamaica	56		Beaumont	86
Ariz.	Phoenix	82	Kans.	Concordia	57		Mt. Vernon	74		Dallas	86
	Tucson	80		Kansas City	86		New Rochelle	75		El Paso	85
Calif.	Anaheim	60		Topeka	88		New York	72		Fort Worth	85
	Berkeley	69		Wichita	72		Rochester	70		Galveston	90
	Fresno	80	Ky.	Louisville	85		Schenectady	60		Houston	83
	Fullerton	75	La.	Baton Rouge	85		Syracuse	74		Port Arthur	83
	Glendale	68		New Orleans	90		Utica	69		San Antonio	78
	Los Angeles	80		Shreveport	88		Yonkers	70		Wichita Falls	85
	Oakland	69	Me.	Augusta	60	N. C.	Asheville	74	Utah	Logan	44
	Ontario	70	Md.	Baltimore	75		Charlotte	85		Salt Lake City	65
	Pasadena	82	Mass.	Boston	80		Raleigh	92	Va.	Fredericksburg	75
	Pomona	75		Cambridge	70		Winston-Salem	82		Lynchburg	73
	Riverside	78		Fall River	76	N. M.	Albuquerque	65		Norfolk	80
	Sacramento	72		Lowell	50	Ohio	Akron	76		Richmond	85
	San Bernardino	65		Lynn	68		Canton	50	Wash.	Olympia	58
	San Diego	84		New Bedford	70		Cincinnati	84		Seattle	62
	San Francisco	71		Salem	68		Cleveland	77		Spokane	51
	Whittier	75		Worcester	76		Columbus	84		Tacoma	57
Colo.	Denver	75	Mich.	Detroit	77		Dayton	60	W. Va.	Charleston	85
Conn.	Bridgeport	66		Flint	70		Lakewood	82		Huntington	78
	Hartford	73		Grand Rapids	84		Springfield	72		Wheeling	78
	New Haven	76		Highland Park	77		Toledo	83	Wis.	LaCrosse	54
	Waterbury	72		Jackson	56	Okla.	Oklahoma City	82		Madison	58
D. C.	Washington	84		Kalamazoo	53		Tulsa	86		Milwaukee	75
Del.	Wilmington	83		Lansing	64	Ore.	Eugene	60		Racine	68
Fla.	Jacksonville	86		Saginaw	82		Portland	65			
	Miami	82	Minn.	Duluth	55	Pa.	Altoona	74	Province	City	Temp. F
	Tampa	77		Minneapolis	85		Erie	75			
Ga.	Atlanta	85		St. Paul	80		Johnstown	74	Alta.	Calgary	64
	Macon	80	Mo.	Jefferson City	82		McKeesport	82			
Idaho	Boise	60		Kansas City	84		Philadelphia	85	B. C.	Vancouver	60
Ill.	Chicago	78		Springfield	82		Pittsburgh	86			
	Cicero	76		St. Joseph	84	R. I.	Providence	68	Ont.	London	50
	Evanston	73		St. Louis	85						
	Moline	83		Springfield	74	S. C.	Charleston	83		Toronto	63
	Peoria	67	Nebr.	Lincoln	70		Greenville	81			
	Rockford	59		Omaha	85		Spartanburg	78	P. E. I.	Charlottetown	48
	Springfield	82	Nev.	Reno	70	S. D.	Rapid City	65			
Ind.	Evansville	88	N. H.	Manchester	76				Que.	Montreal	78
	Gary	75	N. J.	Jersey City	63	Tenn.	Chattanooga	84			
	Indianapolis	84		Newark	75		Knoxville	90		Quebec	68
	South Bend	61		Paterson	78		Memphis	85			
	Terre Haute	82		Trenton	79		Nashville	90			

* These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown. Some values were supplied by H. E. Degler, Marley Company. Some were obtained from City Water Department records. The highest values given by the various authorities are usually those listed.

Performance of an air washer, when cooling and dehumidifying, depends upon a number of factors, among which are:

(1) Air velocity through spray chamber area.
(2) Spray nozzle pressure.
(3) Size factor (larger washers are more efficient than smaller ones).
(4) Spray density (gpm/sq ft/spray bank).
(5) Water temperature rise.
(6) Air wet-bulb temperatures—entering and leaving.
(7) Water-to-air weight ratio.
(8) Water temperatures—entering and leaving.
(9) Length of air washer in direction of flow.
(10) Number and arrangement of spray banks.

Where increase of overall heat transfer between the air

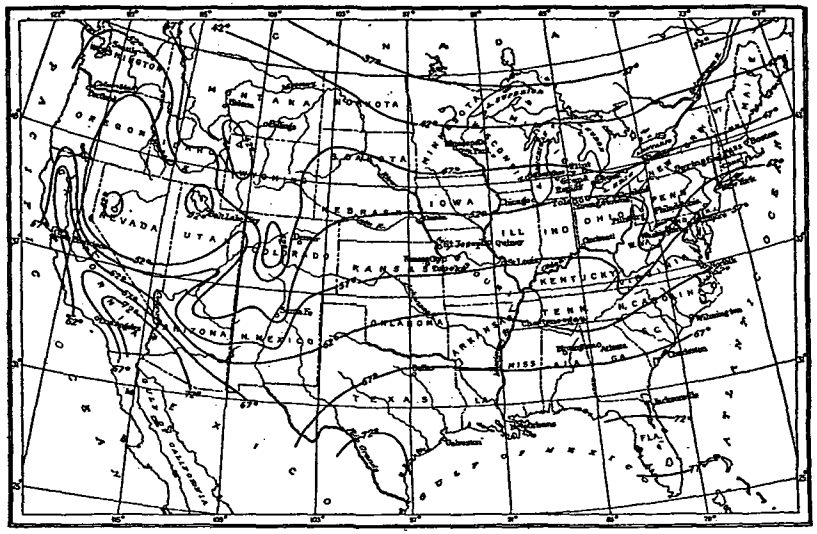


Fig. 4 .... Approximate Well Water Temperatures at Depths of 30 to 60 Ft[1]

and water is required, multi-stage washers are used. These washers are equivalent to a number of washers in series, and the water has to be pumped from one stage to the other in a direction counter to the air flow.

The most common air washer arrangement for cooling and dehumidifying air has two spray banks and is eight to nine feet long. If the air washer has ability to cool and dehumidify the entering air to a wet-bulb temperature equal to the leaving water temperature, it is convenient to assign to such a washer a performance factor of 1.0. The actual performance factor of any washer is the actual enthalpy change divided by the enthalpy change in a washer of 1.0 performance factor.

Calculation of the required performance factor, F_P, for any washer application involving cooling and dehumidifying can be made from Equation 2:

$$F_P = (h_1 - h_2)/(h_1 - h_3) \qquad (2)$$

where

h_1 = Enthalpy at entering air wet-bulb temperature, Btu per pound.

h_2 = Enthalpy at leaving air wet-bulb temperature at actual condition, Btu per pound.

h_3 = Enthalpy at wet-bulb temperature leaving a washer with $F_P = 1.0$, Btu per pound.

Knowing the performance factor of a particular air washer, the actual conditions of operation can be graphically determined as shown in Fig. 5. Points 1 and 2 are plotted on the saturation curve representing total heat at the entering and leaving air wet-bulb temperatures, t_1' and t_2'. Point 5 represents the condition at which leaving air wet-bulb and leaving water temperatures would be the same. This point is determined by solving Equation 2 for the unknown quantity, h_3.

$$h_3 = h_1 - \frac{(h_1 - h_2)}{F_P}$$

A diagonal line is drawn through Point 5 with a negative

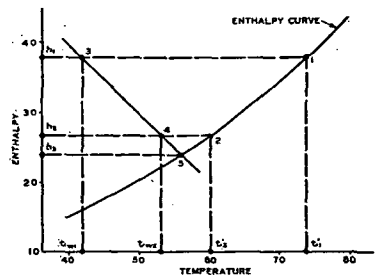


Fig. 5 .... Graphical Solution of Conditions Produced by an Air Washer

slope equal to the water-to-air weight ratio. Points 3 and 4, at which this diagonal line intersects horizontal lines through points 1 and 2, show the required entering and leaving water temperatures t_{w1} and t_{w2}. A check of the solution can be made from the fundamental heat balance equation,

Heat absorbed by the water = heat removed from the air.

The graphical method as described can be used to arrive quickly at solutions of air-washer cooling and dehumidifying problems where there are a number of unknown factors, including quantity of water to be used and the entering and leaving water temperatures. It can be adapted to problems of heating and humidifying as well as to water cooling (indoor washer-cooling tower) and multi-stage applications.

The actual performance factor of a particular washer must, however, be obtained from the manufacturer's data and depends on the many factors listed previously.

Control of Air Washers

Chapter 43 discusses methods of controlling air washers to meet system or space requirements.

EVAPORATIVE AIR COOLING

Evaporative air cooling is undoubtedly the oldest method used in man's attempt to produce comfort in hot climates. In pre-biblical times,[2, 3, 4] wetted grass mats and porous jars utilized the evaporative process to cool air or water. The early settlers of Southwestern United States found the Indians using these same devices. In the 1930's a rush of inventions appeared which mechanized evaporative cooling and various devices were developed and manufactured to utilize evaporation directly in the air stream or indirectly to cool the air in a heat exchange process.

Evaporative air cooling equipment may be placed in two general classes, direct and indirect. The first of these uses primary wet surfaces which evaporate water directly into the fresh air supplied to the space being cooled. This may be done with a series of sprays as in an air washer or by use of an extended wetted surface material such as aspen wood excelsior, glass fibers, metal wire, or expanded paper. As these devices are relatively simple and economical, they represent the vast majority of equipment in use today.

The second class of evaporative air cooling is termed indirect. The indirect cooler uses the evaporative principle to cool air or water in a device similar to a cooling tower. The resultant cool air or water is then used by means of a secondary heat exchanger to cool the air in the occupied space. The simplest form of this consists of a cooling tower, a circulating pump, and a chilled water coil. The pump circulates the chilled water from the cooling tower sump through the blast coil in the building ventilating system. More complex arrangements of this principle have been used. By applying recirculating, regenerating principles, temperatures below the initial wet-bulb may be produced.[5, 6] The complexity and cost of these devices has made them relatively obsolete with the advent of economical refrigeration equipment.

Types of Primary Wet-Surface Evaporative Cooling Equipment

Currently, the drip or desert-type coolers (see Fig. 6) constitute the largest percentage of evaporative coolers. These devices use evaporative pads, usually made of aspen wood fibers, and a water circulating pump to lift the sump water

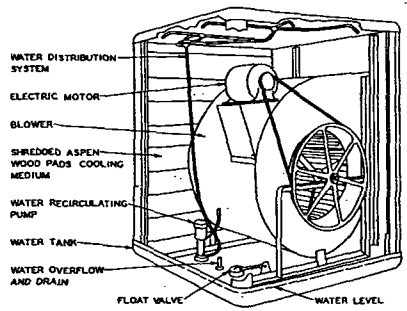


Fig. 6 .... Typical Drip-Type Evaporative Cooler

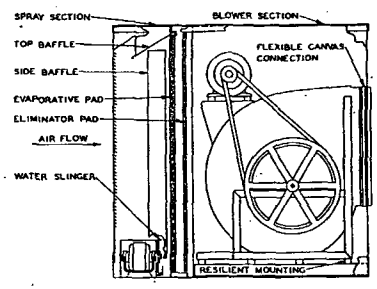


Fig. 7 .... Typical Spray-Type Evaporative Cooler

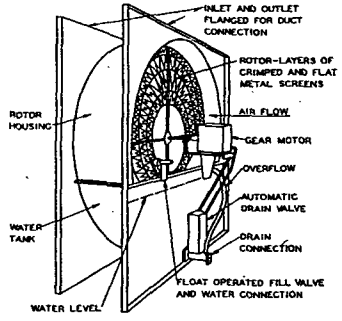


Fig. 8 .... Typical Rotary-Type Evaporative Cooler

up to a distributing system from which it runs down through the pads and back into the sump. A fan (usually included within the unit) pulls the air through the evaporative pad and delivers it to the space to be cooled.

The spray-type cooler (see Fig. 7) contains an evaporative pad usually made of glass fiber, a water slinger, a centrifugal vaporizer, or spray nozzles to throw a fine spray into the air stream and the evaporator pads. Eliminators are usually required on the downstream side of the evaporator pads. These devices are supplied with or without the air-handling fan.

The rotary-type cooler (see Fig. 8) is a device which continually wets and washes the evaporator pad by rotating it through a water bath and into the air stream. The evaporator pads are usually of copper wire construction to provide the necessary structural and corrosion resistance.

The air washers previously discussed in this chapter are also used for direct evaporative cooling.

Control of sump water within these evaporative coolers consists of a float valve for level control and a bleed-off device. Overflow stand pipes, automatic valves or recirculating pump bypass connections are used for bleed-off. An evaporative air cooler operating with an efficiency of 80 percent is generally considered optimum. In this case, efficiency is the same as humidifying or saturating effectiveness found from Equation 1.

Design Considerations

Various methods for designing evaporative cooling systems are in common use. Simplified rule of thumb methods use an assumed air change rate as a basis. Good engineering practice requires an evaluation of the building heat gain, internal loads, and the cooling season wet-bulb temperature pattern.

The suitability of evaporative air conditioning for a particular application is, in the first instance, determined by the climatic conditions. The lowest temperature theoretically attainable with the most common systems of the direct or single-stage type is that of the wet-bulb temperature of the entering air. The process is one of adiabatic saturation and in practice is almost always operated with 100 percent outdoor air. Since the lowering of the dry-bulb temperature of the air during the evaporative process is always accompanied by a corresponding increase in moisture content, it follows that the greatest application of evaporative cooling for conditions for human occupancy is in areas in which periods of high temperature are always accompanied by low outdoor relative humidities. The areas in which the climate is suitable for the reliable production of optimum comfort conditions by evaporative cooling are limited. Despite this, evaporative air conditioning is extensively used over areas with less favorable climates as a means of improving conditions for human occupancy. It finds wide application also in industrial applications, particularly in those in which an improvement of dry-bulb temperatures can be used to advantage, and in which high humidities are desirable or at least not objectionable.

Proper analysis of the cooling season climatic data is most important. The most acceptable approach appears to be based on an analysis of the average high wet-bulb temperatures of the area for the periods when cooling is needed. Criteria established on this basis must, of necessity, take into account the extent to which departures from optimum conditions both in degree and duration can be tolerated.

There is no latent heat load on evaporative cooling equipment as with refrigeration cooling, since it operates on a 100 percent outdoor air basis. The air quantity handled by a typical evaporative cooling system is higher than with refrigerated cooling or heating systems. These relationships can be seen by comparison of *Example 1* with the sample air-conditioning cooling load calculation, *Example 11* of Chapter 13.

Example 1: An evaporative cooling system is to be installed in the one-story office building shown in Fig. 6 of Chapter 13. Outdoor design conditions are assumed to be 95 F dry-bulb and 65 F wet-bulb. The heat gains, calculated in *Example 11* of Chapter 13, which are used in the design of this type of system are:

All walls, roof and doors	78,500 Btu/hr
Glass areas	5,970
Occupants	17,000
Lighting	62,700
Total sensible heat load =	164,170 Btu/hr

Find the required air quantity, the temperature and humidity ratio of the air leaving the cooler (entering the office), and the temperature and humidity ratio of the air leaving the office.

Solution: A temperature rise of 10 F deg in the cooling air is assumed. Equation 20 from Chapter 13 may then be used to calculate the air volume required to be supplied by the evaporative cooler:

$$Q_m = \frac{q_s}{1.08(t_i - t_s)}$$

$$= \frac{164{,}230}{1.08 \times 10} = 15{,}200 \text{ cfm}$$

where

Q_m = required air quantity through equipment, cubic feet per minute.
q_s = instantaneous sensible heat load, Btu per hour.
t_i = indoor air dry-bulb temperature, Fahrenheit.
t_s = room supply air dry-bulb temperature, Fahrenheit.

This air volume rate represents a 2.6 minute air change for a building of this size.

The evaporative air cooler is assumed to have a saturating effectiveness of 80 percent. This is the ratio of reduction of the dry-bulb temperature to the wet-bulb. The dry-bulb temperature of the air leaving the evaporative cooler is found from Equation 3:

$$t_2 = t_1 - e_s(t_1 - t') \qquad (3)$$

For the conditions of this problem Equation 3 becomes:

$$t_2 = 95 - 0.8(95 - 65)$$

$$t_2 = 71 \text{ F}$$

The humidity ratio of the cooler discharge W_2 is found from the psychrometric chart $W_2 = 0.01185$ lb per lb dry air.

Equation 8, Chapter 13, may be adapted to solve for the remaining unknown, the humidity ratio of the air leaving the space being cooled W_3, from Equation 4:

$$q_s = Q_m \times 4840(W_3 - W_2) \qquad (4)$$

where

q_s = latent load, Btu per hour.
Q_m = rate of entry of air to the space from the cooler, cubic feet per minute.

The humidity ratio of the air leaving the space, W_3, from Equation 4, is therefore:

$$W_3 = \frac{q_s}{Q_m \times 4840} + W_2$$

The only latent load of the space for this design is the occupancy load, 21,250 Btu/hr as calculated in *Example 11* of Chapter 13. Substituting this and the other known quantities in Equation 4:

$$W_3 = \frac{21{,}250}{15{,}200 \times 4840} + 0.01185 = 0.01214 \text{ lb/lb dry air.}$$

The remaining values of wet-bulb and relative humidity for the problem may be found from the psychrometric chart. Fig. 9 illustrates the various relationships of the outdoor air, supply air to the space, and the discharge air.

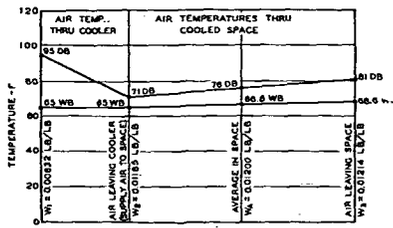


Fig. 9 .... Conditions Within Evaporative Cooler and Cooled Space for Example 1

The design of evaporative air-cooling systems is similar to other heating or cooling systems. It should be kept in mind that considerably more air will be handled as no recirculation is used and, therefore, duct, fan, and grille sizes will be larger. This problem has been solved in a number of ways. Large, open buildings such as supermarkets have been successfully cooled and heated by the use of four evaporative air coolers for cooling, two of which are fitted with duct heaters for space heating. In this way a balance between the summer cooling and winter heating air quantity is obtained. In climates which are not particularly suited to direct evaporative air cooling, a combination of refrigeration and evaporative cooling may be economical.

There are many industrial demands for cooling where direct evaporative cooling is economical. Successful application of this type has been made for electric-motor cooling, turbine cooling, and spot cooling in the factories such as textile mills, steel mills, and foundries. Laundries, with their large exhaust air loads, may also take advantage of evaporative cooling. In cattle and poultry barns evaporative cooling has been used with excellent results even in climates which give poor results with human occupancy. This is possible because the blood temperature of these animals is higher than human.[3] Greenhouses use evaporative cooling to reduce temperature and raise humidity.

Two Stage Evaporation Air Cooling

On industrial or commercial installations two stage evaporative cooling will sometimes be found to be effective and economical. Such a system uses a cooling tower and a blast coil ahead of the evaporative air cooler. Using a 6 or 8 row coil and a well designed cooling tower, the air discharged from the blast coil can be within 15 F of outdoor wet bulb. This part of the cooling takes place at constant humidity ratio. Passing the cooled air through an evaporative air cooler further reduces the dry-bulb temperature, the process following the thermodynamic wet-bulb temperature line on the psychrometric chart. Use of this two stage system will result in smaller required air quantities and consequent low duct costs. Careful design can frequently allow the same duct and grille system to be used for heating and maintain proper duct and register velocities.

Control of Evaporative Air Coolers

Control of evaporative air-cooling systems may be accomplished by ordinary dry-bulb thermostats or a combination of a dry-bulb thermostat and a humidistat. These controls are set to turn on at a dry-bulb temperature and off at a pre-set high humidity. When two-speed fans are used, automatic control of fan speed may be employed to provide better control results.

Evaporative air-cooling systems can also be used for ventilation on mild days and at night when air temperatures are relatively low and straight ventilation produces comfortable results. A valve on the water supply to the evaporative air cooler may be operated by a thermostat in the outdoor air to accomplish this automatically. See Chapter 43 for general information on controls.

Maintenance

Much of the discussion of maintenance and operation problems at the end of Chapter 40, Evaporative Apparatus for Heat Rejection, is pertinent to evaporative air cooling and humidification equipment as all these devices use the evaporation of water as the cooling mechanism.

Some air washers and all evaporative air coolers contain extended surface, evaporator pads of one sort or another. Because of these pads, control of the scale-forming mineral content of the water is required or the extended surface pads will soon plug up with scale and the air flow will be reduced. Another effect of scale deposit is to cause channeling of the water and air and so give local high velocities and consequent entrainment. Some scale will be deposited on these pads during drying after shut-down. If replacement or cleaning is required more frequently than once a year the scale control method is usually at fault. Chemical water treatment with polyphosphates is helpful to prevent hard scale. Regular blowdown or bleed-off of sump water is usually the best control to prevent corrosion and hard scale deposit.

REFERENCES

[1] W. D. Collins: Temperature of water available for industrial use in the United States (*U. S. Geological Survey Water Supply Paper* No. 520 F).

[2] Air conditioning in Iran (*Heating and Ventilating*, June 1947, p. 93).

[3] D. L. Fiske: Refrigeration is not new (*Refrigerating Engineering*, October 1932, p. 201).

[4] George Cecil: Keeping cool in India (*The Heating and Ventilating Magazine*, June 1927, p. 61).

[8]K. P. Brace: Regenerative cooling for hot, dry climates (*Heating and Ventilating*, May 1932, p. 36).

[9]J. R. Watt and R. A. Bacon: Indirect evaporative air cooler (*Heating, Piping and Air Conditioning*, May 1956, p. 149).

[10]R. S. Ash: Evaporative cooling for farm animals (*Air Conditioning, Heating and Ventilating*, October 1956, p. 109).

BIBLIOGRAPHY

H. M. Hendrickson: How air washers perform when cooling (*Heating, Piping and Air Conditioning*, March 1954, p. 119).

H. M. Hendrickson: How to calculate air washer performance when cooling (*Heating, Piping and Air Conditioning*, September 1954, p. 116).

T. W. Reynolds: Evaporative cooling in air conditioning applications (*Heating and Ventilating*, March 1944, p. 53).

F. W. Hutton: Useful tool—properly engineered evaporative cooling (*Heating, Piping and Air Conditioning*, July 1950, p. 96).

R. S. Farr: Evaporative cooling for comfort (*Refrigerating Engineering*, May 1953, p. 527).

M. H. Irons: Evaporative-cooling for textile management (*Mechanical Engineering*, February 1943, p. 115).

G. T. Lang: Evaporative cooling for textile mills (*Heating and Ventilating*, December 1945, p. 89).

W. L. Fleisher and J. H. Burges: Air conditioning the textile mill (*Refrigerating Engineering*, May 1954, p. 33).

W. L. Fleisher: Evaporative cooling for comfort (*Refrigerating Engineering*, December 1936, p. 409).

R. S. Ash: Cooling industrial hot spots with evaporative coolers (*Heating, Piping and Air Conditioning*, December 1955, p. 94).

R. S. Ash: Summer cooling for greenhouses (*Air Conditioning, Heating and Ventilating*, August 1957, p. 67).

Evaporative Cooling—A Symposium (Presented at ASHAE 1955 Semi-Annual Meeting).

CHAPTER 42

DEHUMIDIFICATION BY SORBENT MATERIALS

Types of Sorption Dehumidifiers; Liquid Sorption Systems: Liquid Sorbents, Machine Operation and Design; Solid Adsorption Systems: Solid Desiccants, Machine Operation and Design; Sorption Dehumidifiers for Elevated Pressures; Applications for Sorption Dehumidifiers; Moisture Load Calculations; Vapor Transfer to Dehumidified Space

DEHUMIDIFICATION as used herein is the reduction of the water-vapor content of a given volume of air or other gas. The term thus describes a special case of dehydration which covers the removal of moisture in any form from matter. The degree of dehumidification required varies greatly with different applications, and is one of the prime considerations influencing the choice of a method.

Dehumidification may be accomplished by chilling, as described in Chapters 19, 23, and 41, by the use of sorbents, or by compression in combination with chilling, sorbents, or both chilling and sorbents.

Sorbents are substances which have the property of extracting and holding other substances (usually gases or vapors, e.g., water vapor) brought into contact with them. All materials are sorbents to a greater or lesser degree. The weight of water held by a substance will increase or decrease, depending upon whether the vapor pressure of the water held by the substance is less or greater, respectively, than the partial pressure of water vapor in the surrounding atmosphere. As generally used, however, the term sorbents refers to those materials having a capacity for moisture which is large compared to their volume and weight. Such materials are divided into two general classifications:

1. *Absorbent*—A sorbent which changes either physically, chemically, or both, during the sorption process. Calcium chloride is an example of a solid absorbent, while liquid absorbents include solutions of the halogen group (lithium chloride, lithium bromide), and the ethylene glycols.

2. *Adsorbent*—A sorbent which does not change physically or chemically during the sorption process. Certain solid materials, such as activated alumina, silica gel, activated bauxites, and activated charcoal have this property. The action of adsorbents, most of which adsorb some gases and condensible vapors besides water vapor, is selective. Thus, in the case of a mixture containing both water and organic vapors, silica gel would remove the water vapor in preference to the organic vapors, while the reverse would be true in the case of activated carbon. The selective property of adsorbents is made use of in some instances for the removal of objectionable and contaminating vapors from an air or gas mixture. (See Chapter 7.)

LIQUID ABSORPTION SYSTEMS

Liquid Absorbents

A liquid absorbent has the property of absorbing moisture from, or adding moisture to, the air, depending upon the vapor-pressure difference between the air and the solution. The equilibrium vapor pressure of the solution is dependent upon the temperature and concentration of the solution.

Absorbents for a liquid dehumidifying system should have the following characteristics:

1. Suitable vapor-pressure characteristics. The absorbent should not crystallize at temperatures 10 to 15 deg below the temperature range used in the operating cycle.
2. Noncorrosive, odorless, nontoxic, and nonflammable.
3. Stability. It should not break down over the range of use.
4. Low viscosity and good heat-transfer characteristics.
5. Widely available and economical to produce.
6. Capable of being regenerated at temperatures obtainable with low-pressure steam.

The equilibrium contact dew point is plotted against contact temperature for various concentrations of a typical organic liquid absorbent in Fig. 1. The same data for a typical inorganic absorbent is shown in Fig. 2. The concentration of the liquid absorbents used in commercial equipment is dependent on the temperature of the cooling medium, and on the latent and sensible heat-removal requirements.

The equilibrium contact dew points in Figs. 1 and 2 show the dew point of air in equilibrium with the liquid absorbent. Reference should be made to the published performance data for commercial apparatus for the solution of air-conditioning and air dehumidification problems. Fig. 3 shows a rating curve for a liquid absorbent dehumidifier using lithium chloride solution.

Liquid Absorption Operation

The operating cycle of a liquid absorption system with extended-surface contactor coils is shown by Fig. 4. For dehumidifying operation the strong absorbent solution is pumped from the sump of the unit and sprayed over the contactor coils.

The hygroscopic solution at the required temperature and concentration is in intimate contact with the air which is flowing over the coil surface in the same direction as the liquid absorbent. Equipment is also available for flow countercurrent to the flow of liquid absorbent. Moisture is absorbed from the air by the solution due to the vapor-pressure difference between the air and the liquid absorbent. The moisture content of the outlet air can be maintained at a constant condition by automatically regulating the flow of water through the cooler with a modulating water valve.

The heat generated in absorbing moisture from the air is called the heat of condensation of the water vapor in the solution. This consists of the latent heat of condensation of water vapor and the heat of solution or the heat of mixing. The heat of mixing varies with the liquid absorbent used and with the concentration and temperature of the absorbent. The solution is maintained at the required temperature by cooling with city, well, refrigerated, or cooling tower water, or refrigerant flowing inside the tubes of the contactor coil. The quantity of water required is a function of the water temperature and the total heat, either sensible, latent, or both, removed from the air by the hygroscopic solution.

The dry-bulb temperature of the air leaving the liquid absorbent contactor is a function of the temperature of the liquid absorbent and the amount of contact surface between the air and the solution. In most commercial equipment the

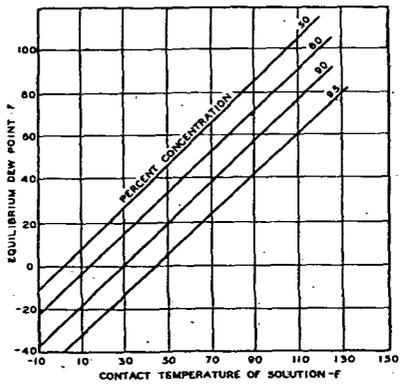


Fig. 1 .... Temperature—Equilibrium Dew-Point Characteristics for Typical Organic Absorbent

dry-bulb temperature of the air leaving the dehumidifier will be within 1 to 5 deg of the liquid absorbent temperature.

Cooling and heating coils are installed in the ductwork after the dehumidifier if the air is to be cooled or heated.

The liquid absorbent is maintained at the proper concentration for moisture removal by automatically removing from the liquid the water vapor absorbed from the air. A small percentage of the solution, usually 10 to 20 percent of the flow to the contactor coils, is passed over the regenerator coil where the liquid is heated with steam or other heating medium. The liquid absorbents commonly used may be regenerated with steam at 2 to 25 psig. The vapor pressure of the liquid absorbent at temperatures corresponding to 2

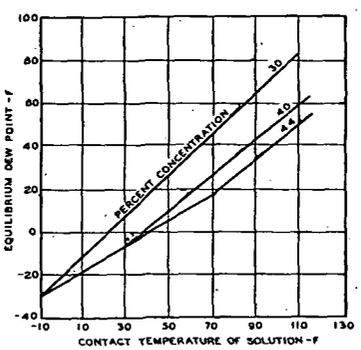


Fig. 2 .... Temperature—Equilibrium Dew-Point Characteristics for Typical Inorganic Absorbent

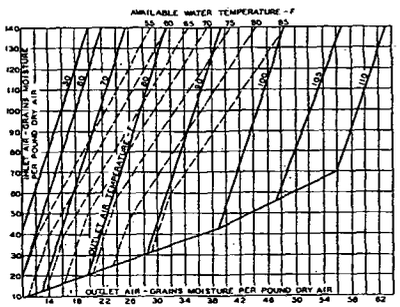


Fig. 3 .... Performance Data for Liquid-Absorbent Dehumidifier Using Lithium Chloride

psig steam is considerably higher than that of the outdoor air. The hot solution at the relatively high vapor pressure is in contact with outdoor air in the regenerator where water is absorbed from the solution by the scavenger air due to the vapor-pressure difference between the outdoor air and the hot solution. The hot-moist air from the regenerator is discharged to the outdoors and the concentrated solution flows to the sump where the process is repeated.

The steam flow to the regenerator coil is regulated by a control responsive to the concentration of the solution circulated over the contactor coils. This may be a level control, specific gravity control, boiling point control, or a similar instrument.

For humidifying operations the liquid absorbent is maintained at the required temperature by adding heat in proportion to the water absorbed by the air from the solution. Water is automatically added to the solution to maintain the proper concentration.

SOLID ADSORPTION SYSTEMS

Solid Desiccants

The ability of an adsorbent (desiccant) to remove water vapor from a gas is explained by the fact that the vapor pressure of the water in the adsorbent (when in the re-

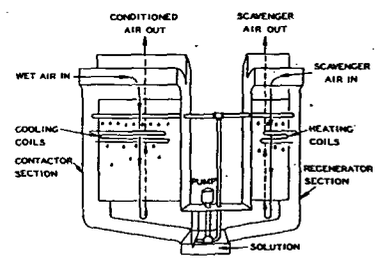


Fig. 4 .... Flow Diagram for Typical Liquid-Absorbent Dehumidifier

activated condition) is less than the partial pressure of the water vapor in the surrounding atmosphere. For instance, when an active adsorbent is brought into contact with a gas of high humidity, there is a tendency for the vapor pressure of the water in the adsorbent to reach equilibrium with the partial pressure of the water in the surrounding gas, with the result that water is extracted by the adsorbent and its weight increased, while the moisture content of the gas is correspondingly reduced. (The adsorbent is said to be saturated for a given condition when equilibrium is attained.) The weight of water a given adsorbent will extract is dependent upon the relative humidity (ratio of the partial pressure in the gas to the saturation pressure at a given temperature) and the temperature of the adsorbent. The process is reversible; if the temperature of the adsorbent is raised until the vapor pressure of the adsorbed water becomes greater than the partial pressure of the vapor in the surrounding atmosphere, water will be released by the adsorbent. After the adsorbent cools to room temperature, for instance, the vapor pressure of the water in the adsorbent falls below the partial pressure of the vapor in the atmosphere, and the adsorbent will again start extracting water. The elimination of water by the addition of heat is known as reactivation, and is a means of regenerating the adsorbent so that it may be used repeatedly.

Adsorption is proportional to the amount of surface (internal and external) of the sorbent. The materials that are used commercially as solid adsorbents have a porous structure of submicroscopic dimensions, which gives them extensive surface area. An adsorbent should meet the following requirements in order to be satisfactory for dehumidification purposes:

1. Have a high adsorptive capacity under normal atmospheric conditions.
2. Be chemically stable, resisting contamination from impurities.
3. Be physically rugged to resist breakdown from handling and use.
4. Be capable of reactivation at temperatures generally obtainable.
5. Be heat-stable at reactivation temperatures.
6. Have a weight per unit volume such as to avoid excessive bulk.
7. Be available at reasonable cost.

There are a number of solid adsorbents available, which are used in solid adsorption systems. The reader is referred to the various manufacturers for detailed information regarding performance of the various desiccants. These include activated aluminas and bauxites, silica gels, and alumino-silicates marketed under various trade names. Reactivation temperatures for these desiccants are generally between 200 and 600 F.

Machine Operation and Design

Dehumidification by a solid desiccant such as silica gel or activated alumina may be performed either under static or dynamic operation. As the name implies, the static method entails no artificial means for circulating the gas to be dried into or through the desiccant. Instead, the air immediately surrounding the adsorbent is initially dried and subsequently, through convection and diffusion, water vapor from spaces and objects further away pass into the air surrounding the desiccant and then to the desiccant where it is adsorbed. Obviously this type of dehumidification is best suited for small containers. This method of preservation was used extensively by the armed services during World War II for overseas shipments and domestic storage.

On the other hand, dynamic dehumidification entails the forced passage of the air being treated through the desiccant bed. This method is used in installations such as dehumidified warehouses and ships of the "Mothball Fleet" where large volumes of air must be dried.

The only requisites for a dynamic dehumidifier are a desiccant bed, a fan to force the humid air through this bed, and a heater to periodically reactivate the adsorbent. As the air passes into the activated desiccant, it surrenders a certain amount of its water vapor. The rate of moisture pickup and the humidity condition of the leaving air are functions of a great many variables, some of which will be discussed later. The ratio of the amount of water adsorbed by the desiccant in a given time to the amount of water vapor in the air entering the desiccant bed during that time is known as *adsorption efficiency*. A characteristic of adsorbents in dynamic use is that this adsorption efficiency remains constant and at a relatively high level from the beginning of an adsorption cycle until some later point in the cycle at which time the efficiency begins to drop. This point is known as the *breakpoint*, and the time from the beginning of adsorption to this point is known as the *breakpoint time*. In the ideal case the breakpoint would coincide time-wise with the sudden rise in the dew-point temperature of the effluent air. Although additional drying can be effected beyond the breakpoint, good commercial practice dictates that the desiccant be regenerated at or near this point. Adsorption carried beyond the breakpoint continues at an increasingly slower rate until the adsorbent is completely saturated. This point is known as *completion*.

Adsorbents, such as silica gel and activated alumina, contain, even when reactivated, a small amount of water. This is usually about 5 to 7 percent of the dry weight of the desiccant and is known as residual moisture. Attempts to remove it result in a physical change in the substance and a reduced adsorptive capacity. The term *useful concentration* is used to designate the percent of moisture in the adsorbent over and above the residual quantity. It is generally based on the dry weight of the desiccant.

When regeneration of the adsorbent is desired, the heater is energized and the direction of air flow through the bed is usually reversed. The dry-bulb temperature of the effluent air rises rapidly at first, and then virtually levels off for a period of time. This period of level or slowly increasing temperature represents the period during which the major portion of the heat input is being used to boil off the adsorbed water. This temperature pattern continues until most of the water contained in or on the desiccant is released. When the latent heat requirements begin to diminish, the heat input goes into sensible heat gain to the passing air stream. This is reflected in a rather sharp increase in the dry-bulb temperature of the effluent air. This period, measured from the beginning of desorption, has been designated *temperature-rise time*. Although additional regeneration (at a slower rate) can be attained by continuing the heat addition process beyond the temperature-rise time, once again good commercial practice calls for reactivation to be ended near this point. Regeneration past this point until the adsorbent is in moisture equilibrium with the air stream is known as *complete desorption* or *desorption to completion*. The energy expended in the heater per unit weight of water desorbed for any given time is called *economy of desorption* and usually has the dimensions of kilowatt hours per pound of water desorbed.

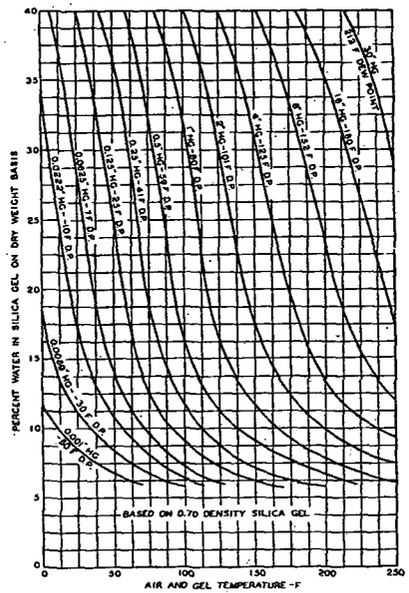


Fig. 5.... Silica Gel—Water-vapor Equilibrium Curves

During the process of adsorption there is a liberation of heat which results in an elevation of the effluent air temperature. This heat is equivalent to the latent heat of vaporization of the adsorbed liquid plus an added quantity known as the *heat of wetting*, which is defined as the heat developed when a liquid and a solid surface contact one another. As the adsorbed vapor condenses, the latent heat is converted to sensible. All of the released heat, known cumulatively as the heat of adsorption, is dissipated into the desiccant, the enclosure, and the passing air stream. In comfort air conditioning it is often necessary to cool this effluent air prior to its introduction to the conditioned space, but in most other dehumidification applications this heat is not objectionable, and no provisions are made for its removal. For this reason the process is known as adiabatic adsorption, meaning that the released heat is taken up for the most part by the passing air stream, that the adsorbent chamber is insulated, and that no attempt is made to cool the effluent air. Although ordinary air-conditioning dehumidification only approaches the adiabatic process, it is called adiabatic to differentiate it from the isothermal adsorption process in which the outlet gas is cooled to inlet temperature.

It is appropriate now to discuss the factors which influence the rate and amount of water-vapor adsorption by the desiccant. Although such factors as the shape and volume of the desiccant pores and the surface tension and wetting ability of the condensed vapor are important, the foremost considerations when working with a given adsorbent and a given vapor, for example, airborne water vapor, are the temperature of the adsorbent and the dry-bulb and dew-point temperatures of the air. The latter temperature is important since it is a function of the partial pressure of the water vapor in the air. The relationships existing between these variables are best illustrated by the Silica Gel–Water-vapor equilibrium curves shown in Fig. 5. In this set of curves, silica gel and air temperature versus percent water in silica gel (dry weight basis) form the abscissa and ordinate, and air dew-point temperatures (with the corresponding vapor pressures) are the parameters. Several important characteristics of the adsorbent are brought out by these curves. For example, note that each of the curves becomes asymptotic to the X-axis somewhere between 5 and 6 percent moisture content. This is because, under normal reactivation temperatures, there is always this amount of water in the gel. Known as residual moisture, it is omnipresent and an attempt to remove it results in a physical change in the desiccant and a reduced adsorptive capacity. The term *useful concentration* is used to designate the percent moisture in the gel over and above the residual amount. The variation in this useful concentration during a test measures the magnitude of adsorption or desorption since the residual amount theoretically remains unchanged assuming no excessive activation temperatures. Note that at constant air and gel temperature the amount of moisture adsorption varies directly with the air dew-point temperature, i.e., as the air approaches saturation condition (dry-bulb temperature equal to dew-point temperature) the equilibrium moisture content of the gel increases. It can also be seen that at saturation the percent water in the gel is in the neighborhood of 40 percent of the dry weight of the gel for all temperatures.

With a fixed dew-point temperature, the moisture concentration decreases as the temperature of the air and gel is increased.

Some of the many variables that influence the results of a dynamic dehumidification operation are:

1. Type of desiccant.
2. Grain size of desiccant.
3. Shape of bed.
4. Area of bed normal to air flow.
5. Depth of bed.
6. Dry weight of desiccant.
7. Packing of the desiccant in the bed.
8. Weight of air flowing per unit of time.
9. Temperature of entering air.
10. Moisture content of entering air.
11. Contact time between air and adsorbent; a function of inlet face velocity and bed depth.
12. Barometric pressure.
13. Pressure drop through bed.
14. Duration of cycles.
15. Reactivation temperature.
16. Rate and magnitude of heat supply during reactivation.
17. Heat storage capacity of the bed.
18. Temperature gradient of the bed.
19. Air leakage from the apparatus.

Solid adsorption dehumidifiers are mostly of the stationary dual-bed type in which one desiccant bed is adsorbing while the other is being reactivated by electricity, gas, or steam.

Fig. 6 shows a schematic flow diagram for a typical dual-bed dehumidification unit. There are two types of units: (a) Shallow-bed; (b) Deep-bed.

Shallow-bed machines have comparatively thin desiccant beds and moisture removal is not complete in one pass through the bed. Low dew points may be maintained by progressive lowering of the room dew point as increasingly drier air enters the machine. The machine capacity must be in excess of internal moisture load and infiltration of moisture from external sources. Increased dryness can be obtained by reducing outlet air flow and recirculation of excess air from the dry air outlet to the humid air inlet on

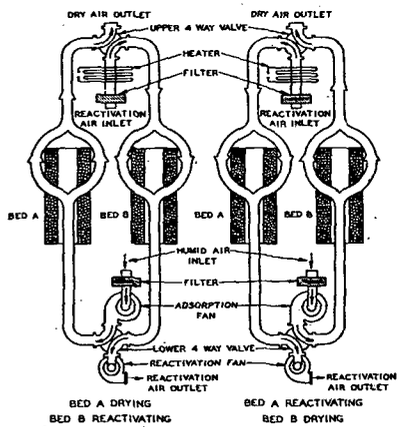


Fig. 6.... Typical Dual-Bed Dehumidification Unit Air-Flow Diagram

the machine. Fig. 7 shows a typical cycle diagram for a shallow-bed machine.

Deep bed machines have desiccant beds 10 in. and more in depth. Deep bed units are used when extreme dryness is required in one pass through the dehumidifier and the air cannot be recirculated. Fig. 8 shows a typical cycle diagram for a deep-bed dehumidifier.

Most fixed-bed machines work on a predetermined time cycle. Smaller machines usually have a nonadjustable cycle. Larger units have adjustable time cycles that can be changed for various operating conditions.

The average hourly moisture removal capacity of a dehumidifier can be determined as follows:

$$W = \frac{60Q(G_i - G_o)}{7000v} \qquad (1)$$

where

W = moisture removed, pounds per hour.
Q = air volume, cfm.
60 = conversion of cubic feet per minute to cubic feet per hour.
G_i = moisture content of air entering dehumidifier, grains of moisture per pound of dry air.

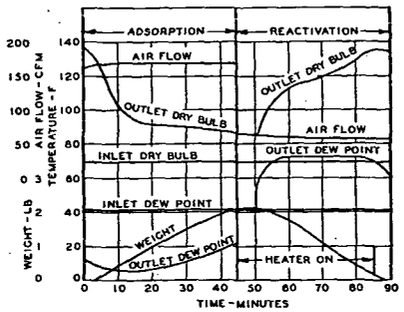


Fig. 7.... Typical Cycle Diagram for Shallow-Bed Dehumidifier

G_o = moisture content of air leaving dehumidifier, grains of moisture per pound of dry air.
v = specific volume of the air, cubic feet per pound.
7000 = conversion from grains to pounds.

Moisture removal capacity may be increased by (1) higher inlet dew point; (2) for a constant entering dew point, lower temperature and greater saturation; (3) larger air quantity, which causes a decrease in the dryness of the effluent air.

If the air to be dehumidified is very warm, and especially where a very low dew point is required, it is advantageous to install a precooler to reduce the temperature of the inlet air. In this way the working temperature in the adsorber is lowered, and the overall performance appreciably increased. Some equipment manufacturers install cooling coils in the adsorbent beds for the same purpose, while others divide the adsorbent bed and install coolers between the sections.

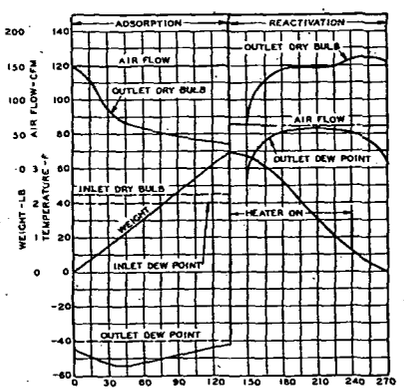


Fig. 8.... Typical Cycle Diagram for Deep-Bed Dehumidifier

MOISTURE CONTENT OF AIR AT ONE ATMOSPHERE

DEW POINT F	MOISTURE: GRAINS PER LB DRY AIR	POUNDS PER LB DRY AIR	GRAINS PER CU FT DRY AIR	PPM*	VOLUME PERCENT DRY AIR
110	400	0.0600	25	60,000	8
		0.0500		50,000	
100	300		20		7
		0.0400		40,000	6
90			15		
	200	0.0300		30,000	5
80					4
	150	0.0200	10	20,000	
			9		3
70			8		
	100	0.0150	7	15,000	
	90				2
60	80		6		
	70	0.0100	5	10,000	1.5
	60	0.0090		9,000	
50	50	0.0080	4	8,000	
		0.0070		7,000	1.0
40	40	0.0060	3	6,000	0.9
		0.0050		5,000	0.8
	30				0.7
30		0.0040	2	4,000	0.6
	20				
		0.0030	1.5	3,000	0.5
20					0.4
		0.0020	1	2,000	0.3
10	10		0.8		0.2
	8				
0	6	0.0010	0.6	1,000	
		0.0006		800	
	4	0.0008	0.4	600	0.1
-10					0.08
		0.0004		400	0.06
-20	2		0.2		0.04
		0.0002		200	
-30	1		0.1		
	0.8		0.08		0.02
-40	0.6	0.0001	0.06	100	
		0.00008		80	
	0.4	0.00006	0.04	60	0.01
-50					0.008
		0.00004		40	0.006
-60	0.2		0.02		0.004
		0.00002		20	
	0.1		0.01		
-70	0.08	0.00001	0.008		0.002
	0.06	0.000008	0.006	10	
				8	
-80	0.04	0.000006	0.004	6	0.001
		0.000004		4	0.0008
					0.0006
-90	0.02		0.002		0.0004
		0.000002		2	
-100	0.01		0.001		0.0002

* PARTS OF MOISTURE PER MILLION PARTS DRY AIR BY VOLUME

Fig. 9 . . . . Moisture Content of Air at One Atmosphere Pressure

Fig. 9 shows the moisture content of air at one atmosphere pressure.

SORPTION DEHUMIDIFIERS FOR ELEVATED PRESSURES

The same sorption principles which pertain to low-pressure (approximately atmospheric) dehumidification apply to drying of high-pressure air, process gases, etc. The sorbents described previously can be used with equal effectiveness.

Equipment design may vary considerably in detail, but most basic adsorption units utilize dual bed construction for continuous operation and either internal or external heat source with air or process gas as reactivation purge for liberating previously adsorbed moisture. A single adsorbent bed may be used for intermittent drying requirements. Absorption units are generally constructed in the same manner as atmospheric pressure units, except that they are enclosed in vessels suitable for the operating pressure.

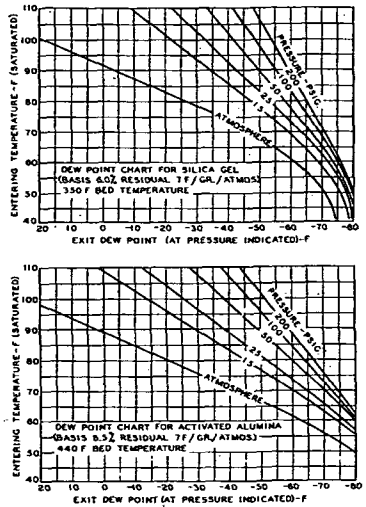


Fig. 10 . . . . Typical Performance Data for Solid Sorption Dehumidifiers at Elevated Pressures

Either compressing of the gas to be dehumidified or precooling by water, brine, or refrigeration will reduce the total moisture load to be handled by the sorbent permitting use of smaller drying units and permitting lower dew-point performance. The performance of the complete system consisting of precooler and adsorbent dryer must be investigated at various temperatures to determine the most economic operating level.

Applications include drying compressed air for processing, testing, and instrument use. Drying of air and gases have made many processes possible or effective in the chemical, petroleum, and steel industries.

Typical performance data for solid sorption dehumidifiers for pressures up to 200 psig are shown in Fig. 10.

A dew-point conversion chart is shown in Fig. 11. This chart is of value to determine changes in air dew points due to compression or expansion.

USES OF SORPTION DEHUMIDIFIERS

Sorption dehumidification equipment is used where it is desirable to control the removal of moisture from the air independent of the dry-bulb temperature of the air. Such a procedure is required in many industrial processes where the relative humidity must be controlled during the manufacture and preservation of the product.

In comfort air-conditioning dehumidification, with cooling, may serve advantageously under certain design conditions such as high moisture load in comparison to sensible heat load, high ratio of fresh air to recirculated air, and internal load variation.

Liquid absorbent dehumidification, in combination with refrigeration, is applied in industrial areas requiring either dew-point or dry-bulb temperatures below the freezing point of water, eliminating the formation of frost or ice on refrigeration coils. This particular function is illustrated by its use in wind tunnels to produce dew points as low as —60 F. It is also employed in low temperature process tunnels and chill rooms.

Other applications include supplying dry air to prevent condensation of moisture on cold surfaces, dehumidification of warehouses or storage rooms for military supplies as well as certain commercial products, and maintaining a dry atmosphere in cargo holds on ships. A more recent development is found in the dehumidification of underground caves.

MOISTURE LOAD CALCULATIONS

The application of dehumidification equipment to maintain spaces at low dew point requires careful calculation of the internal moisture gain in the room from all sources. The internal moisture gain consists of latent load from the occupants, process load from any products which give up moisture, exposed water surfaces in the space, open gas flame, water-vapor migration through cracks around the doors, windows, and other openings in the room (infiltration), and moisture transmitted through the building surfaces.

The latent load from people, exposed water surfaces, open gas flame, and moisture transmitted through building surfaces are found in the appropriate GUIDE chapters. The internal latent load due to water-vapor transfer through cracks, conveyor slots, and open doors has not been standardized and is generally calculated by engineers on the basis of judgment and experience.

Calculation of the dehumidification required to maintain lower than normal moisture content in a given room begins with determination of the rate of moisture gain in the room from all sources. It is common practice, when maintaining a low humidity ratio, to recirculate a large percentage of the air in the room through the dehumidifier, and to add only enough outdoor air to meet the needs of the problem. The humidity ratio of the mixture of outdoor and recirculated air and the dehumidifier performance data can be used to calculate the humidity ratio of the air leaving the dehumidifier. The difference between the humidity ratio of the air in the room and that of the dehumidified air entering the room represents the effective dehumidification per pound of air. The rate of internal moisture gain in grains per minute, divided by the effective dehumidification in grains per pound of air, equals the air quantity required in pounds per minute. The following typical example using arbitrary values shows a general method of determining the dehumidifying requirements. Sensible heat determination considerations are discussed in other chapters, and are purposely omitted here.

Example 1: A sorption dehumidifier is to be used to maintain indoor conditions of 73 F and 20 percent relative humidity, i.e., 24.1 grains per pound of dry air, 30 F dew point, in a room 20 ft x 30 ft x 10 ft high, having a total wall, ceiling, and floor surface area of 2200 sq ft. Outdoor design conditions are 72 F dew point (118.4 grains per pound).

Internal sources of moisture are: 4 occupants; an open natural gas burner using 15 cu ft of natural gas per hour; an open top water tank, having an area of 2 sq ft exposed surface, in which water is maintained at 87 F, with air movement over the water surface being 100 fpm. Determine the quantity and condition of the dehumidified air to be supplied to the room.

Solution: The internal moisture gain consists of items 1 to 5.

		Grains per Minute
1. From occupants:	$4 \times 1800/60 =$	120

1800 grains per person per hour is obtained from Fig. 7, Chapter 6, by interpolation between curves *C* and *D*.

2. From burned gas:	$15 \times 650/60 =$	162

1 cu ft natural gas produces approximately 650 grains of moisture.

3. From exposed water surface:	$2 \times 20 =$	40

Evaporation from water surface is assumed to be 20 grains per (minute) (square foot) at 87 F water with air movement of 100 fpm.

4. From infiltration:	$\frac{6000}{60 \times 13.56} \times (118.4 - 24.1) =$	696

One air change, 6000 cu ft, assumed per hour (see Chapter 11).

5. Moisture transmitted through room surface:

$$\frac{2200}{60} \times 3 \times (0.783 - 0.176) = 67$$

Permeability assumed to be 3 grains per (square foot) (hour) (inch Hg vapor-pressure difference on two sides of wall).

Total moisture gain from internal sources 1085

Let q be the air delivered to the room, pounds per minute. Let it be assumed for this problem that 85 percent of the air is recirculated and 15 percent is outdoor air. Enough air must be supplied to replace leakage from the system or to satisfy normal ventilating requirements for the occupants of the room as given in Chapter 6, whichever is greater. The amount is estimated from experience or obtained by test.

The humidity ratio of the mixture of recirculated and outdoor air entering the dehumidifier is then:

$$\frac{0.85q(24.1) + 0.15q(118.4)}{0.85q + 0.15q}$$

= 38.3 grains per pound entering dehumidifier.

From the manufacturer's performance data for the dehumidifier to be used it is determined that with 38.3 grains per pound of entering air the leaving condition is 6.5 grains per pound.

Effective dehumidification in the room is 24.1 — 6.5 or 17.6 grains per pound of supply air.

Then $q = \frac{1085 \text{ grains per minute}}{17.6 \text{ grains per pound}} = 61.6$ lb air per minute *minimum* that must be supplied to the room to maintain 30 F dew point.

Note that this value represents the minimum requirement for the arbitrary conditions set forth and that in practice, safety margins should be added to the outdoor air percentage value and to the calculated internal moisture gain.

VAPOR TRANSFER TO DEHUMIDIFIED SPACE

The walls enclosing a dehumidified space are subjected to a vapor-pressure differential. The pressure of the vapor outside the walls tends to force moisture through the walls into the dehumidified zone of relatively low vapor pressure. As this process can be an unnecessary load on the dehumidifying equipment, provisions should be made for keeping the vapor transfer to a minimum. Also, if the space is cooled below the ambient dew point, there is a possibility that condensation may occur within the walls, unless vapor

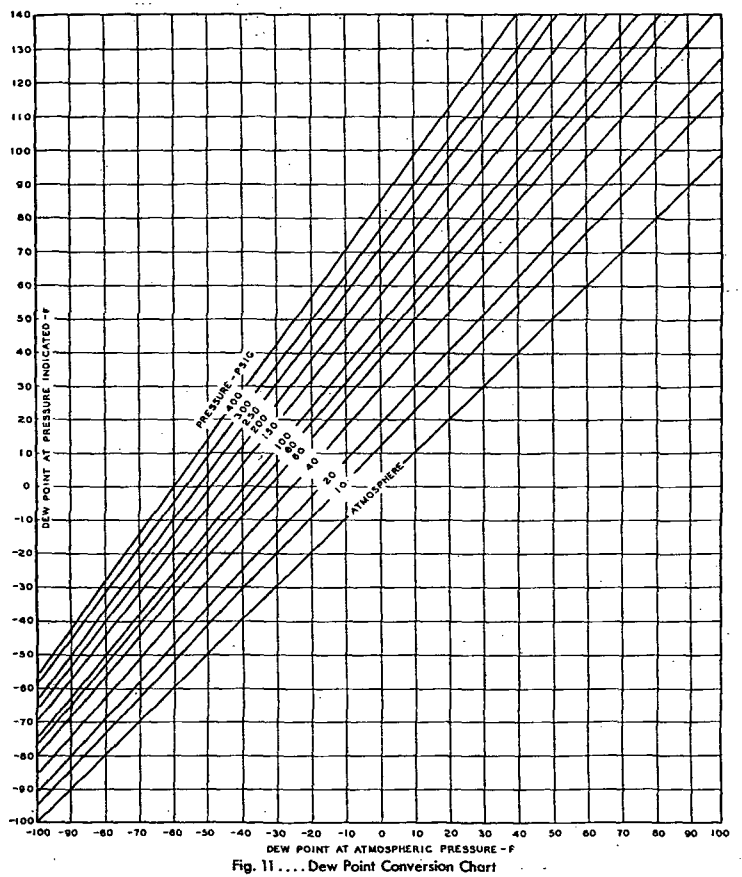


Fig. 11 .... Dew Point Conversion Chart

transfer is controlled. For these reasons a vapor barrier should be located within the wall construction as near to the high vapor-pressure side as feasible. To be effective, a barrier must be continuous and should be so located within the structure that it will be protected from rupture. (See Chapter 10 Moisture In Building Construction.)

BIBLIOGRAPHY

R. C. Amero, J. W. Moore, and R. G. Capell: Design and use of adsorptive units (*Chemical and Engineering Progress*, Vol. 43, July 1947, p. 349). A full bibliography is included in this article.

W. L. Ross and E. R. McLaughlin: An analysis method for predicting behavior of solid adsorbents in solid sorption dehumidifiers (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 321).

Symposium Bulletin on Dehumidification (five papers presented at Symposium held at 63rd Annual Meeting ASHAE February 26, 1957).

G. C. F. Acker and T. H. Urdahl: *Ship Dehumidification Systems* (International Institute of Refrigeration, Nantes, France, June 20, 1957). Several references included.

CHAPTER 43

AUTOMATIC CONTROL

Fundamentals: Types of Control Systems and Action, System Components, Controllers and Controlled Devices, Auxiliary Control Equipment; Design Coordination: Equipment Selection and Layout; Location of Controllers; Control of Steam, Water, or Air Flow; Zone Control; Control Applications: Central Fan, Temperature Control in Hot Water Systems, Zoned Steam and Water Systems, Terminal Equipment, Residential Heating and Air Conditioning

PRESENT-DAY standards of *comfort* combined with capacity and rapid response of modern heating and cooling equipment make automatic controls an essential part of heating, ventilating, and air-conditioning systems. These automatic controls respond to variables such as temperature, relative humidity, and pressure. They operate individually or in sequence to maintain the desired conditions throughout the system and in the occupied space. A factor that has made the subject of automatic control somewhat confusing has been the matter of terminology. Many different expressions or words have sometimes been used to convey a single idea or concept. This chapter, therefore, attempts to use and define the terms that are most common and suitable so that they will be readily understood. The terms and definitions used are also selected so as to conform, as nearly as possible, to automatic control terminology used by engineers in other fields of control.

PART I—FUNDAMENTALS OF AUTOMATIC CONTROL

A *control system*, Fig. 1, consists essentially of (1) a controller, (2) a controlled device, and (3) a source of energy.

A *controller* is a device which measures a variable condition such as temperature, humidity, pressure, and liquid level, and produces a suitable action or impulse for transmission to the controlled devices. Thermostats, humidistats, and pressure controllers are examples.

A *controlled device* reacts to the impulse received from a controller and varies the flow of the control agent. It may be a valve, damper, electric relay, or a motor driving a pump, fan, etc.

The *control agent* is the medium manipulated by the controlled device. It may be air or gas flowing through a damper; gas, steam, water, etc., flowing through a valve; or an electric current.

The *controlled variable* is the condition such as temperature, humidity, or pressure, being controlled.

Most control systems, of which Fig. 1 is typical, form a closed loop. That is, the controller measures and responds to changes in the controlled variable and actuates the controlled device to bring about an opposite change which is again measured by the controller. This system of transmitting information about the results of an action or operation back to its origin is known as *feedback*, and makes true automatic control possible.

An open-loop system is sometimes employed in control circuits, but it does not provide complete control. An outdoor thermostat arranged to control the flow of heat to a building in proportion to the load caused by changes in outdoor temperature is an example. This system depends for its operation on a prearranged relationship between outdoor temperature and heat input to the building, and room temperature has no effect on the controller.

As there is no feedback, the control corrects only for those disturbances of room temperature caused by changes in outdoor temperature.

TYPES OF CONTROL SYSTEMS

Control systems are divided into five main groups according to the primary source of energy:

1. A *self-contained system* combines the controller and controlled device in one unit and employs the power of the measuring system to effect the necessary corrective action. The measuring system derives its energy from the process under control without amplification by any auxiliary source of energy, and may be of the sealed-bellows or remote-bulb type as described under Types of Measuring Elements. Temperature changes at the bellows or the remote bulb result in pressure or volume changes of the enclosed media which are transmitted directly to the operating device of the valve or damper.

2. A *pneumatic system* utilizes compressed air, usually at a pressure of 15 to 25 psig, as a source of energy. This is supplied to the controller which in turn regulates the pressure supplied to the controlled device.

3. A *hydraulic system* utilizes a suitable liquid under pressure as the source of energy. The pressure often is considerably higher than in a pneumatic system, but in other respects the systems are similar. Hydraulic systems find their chief use in applications where large forces are required for operation of the controlled devices.

4. An *electric system* utilizes electric energy, either low or line voltage as the energy source. The electric energy supplied to the controlled device is regulated by the controller either directly or through relays.

5. An *electronic system* also utilizes electric energy, but employs an electronic amplifier to increase the minute voltage variations of the measuring element to values required for operation of standard electrically controlled devices. Measuring elements usually are of the resistance type, but thermocouples also are employed. Combination electronic-pneumatic systems utilize compressed air for operation of the controlled device by converting the output of the electronic amplifier into

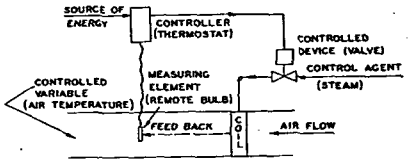


Fig. 1 .... Essentials of a Control System

suitable air-pressure changes by means of an electronic-pneumatic transducer.

TYPES OF CONTROL ACTION

Control action may be divided into five types:

1. *Two-position action*, Fig. 2, is the type in which the controlled device can be positioned only to a maximum or minimum position, or can be either *on* or *off*. A thermostat which opens and closes a valve, or starts and stops a burner, is an example.

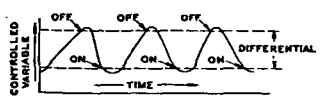


Fig. 2 .... Two-Position Control

2. *Timed two-position action* is a common variation of two-position control action in which the time of *on* periods are prematurely shortened. This type of action usually is employed only in room thermostats, and is accomplished by a heater element in the thermostat which is energized during the *on* periods. The percentage of time *on* is varied in proportion to the system load.

Controller differential applying to two-position control action, is the difference between the setting at which the controller operates to one position and the setting at which it operates to the other position. As an example, if a pressure controller starts a pump at 12 psig and stops it at 15 psig, the differential is 3 psi. It is sometimes desirable to have a controller in which the differential can be changed by manual adjustment. This is designated as an adjustable differential.

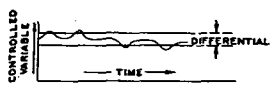


Fig. 3 .... Floating Control Showing Variations in Controlled Variable as Load Changes

3. *Floating action*, Fig. 3, is the type, as with two-position action, where the controller can perform only two operations: moving the controlled device usually at a constant rate either towards its open or its closed position. Generally there is a neutral zone between the two positions which allows the controlled device to stop at any position whenever the controlled variable is within the differential of the controller. When the controlled variable gets outside the differential of the controller, the controller moves the controlled device in the proper direction. An example of floating control is a firebox draft controller positioning a damper in the breeching of a boiler.



Fig. 4 .... Proportional Control Showing Variations in Controlled Variable as Load Changes

4. *Proportional action*, Fig. 4, is the type where the controlled device is positioned proportionately in response to slight changes in the controlled variable. It does not run through its complete stroke, as is the case with two-position control, nor does it continue to move until the change in the controlled variable resulting from that movement is felt by the controller, as is the case with floating control, but it immediately assumes a position in proportion to the system requirement.

An example of proportional control is a thermostat in a fan discharge duct actuating an automatic valve in the steam supply to a coil to regulate the air temperature leaving the coil. This control would be similar to that shown in Fig. 1.

Throttling range is the total amount of change in the controlled variable required for the controller to move the controlled device through its complete stroke, from one extreme to the other. It is often adjustable to meet job requirements.

Set point is the value at which the controller is set and represents the desired value of the controlled variable.

Control point is the actual value of the controlled variable at which the instrument is controlling. It will vary within the throttling range of the controller according to the demand on the system and other variables.

Offset is the difference between the set point and the actual control point under stable conditions. This is sometimes also called *drift*, *deviation*, or *droop*.

5. *Proportional plus automatic reset action*, Fig. 5, combines floating action with proportional action to achieve both the stability of proportional control, with a relatively wide throttling range, and the invariable control point of floating control. The system functions the same as proportional action, the automatic reset serving to shift the control point back to the set point whenever any offset occurs.

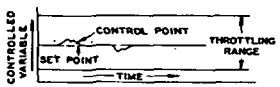


Fig. 5 .... Proportional Plus Automatic Reset Control Showing How Control Point is Returned to Set Point after a Load Change

Reset rate is the number of times per minute the proportional action is duplicated by the reset action, and usually is expressed as *repeats per minute*. The reset rate in most controllers is adjustable and must be carefully matched to the system characteristics to avoid unstable operation. Because automatic reset is necessarily slow-acting, it should be used only when load changes are of reasonably long duration and when the maximum offset resulting from proportional control alone is oustide of acceptable limits.

Figs. 4 and 5 show the results of rather abrupt demand changes, which usually result in two or three cycles before stability is restored. More gradual demand changes will result in a smoother curve.

AUTOMATIC CONTROL SYSTEM COMPONENTS

In addition to controllers and controlled devices, many control systems include auxiliary apparatus such as switches and relays of various kinds, clocks or timers, thermometers, gages, pilot lights, and other indicators for observing the operation of the system. Only those more commonly used will be described here.

Controllers

Automatic controllers have both a measuring element and a controlling element.

The *measuring element* measures changes in the controlled variable and produces a proportional effect on the controlling element. This effect usually is a change in position, force, or electrical resistance.

The *controlling element* converts the effect produced by the measuring element into an effect suitable for operation of the controlled device. (In self-contained controllers the measuring element produces a force which is directly applied to the controlled device. In pneumatic or electronic controllers the controlling element regulates the application of energy such as compressed air or electricity.)

Types of Measuring Elements

Temperature measuring elements usually consist of (1) a bimetal strip, (2) a rod and tube of dissimilar metals, (3) a sealed bellows, with or without a remote bulb, or (4) an electrical resistance.

1. A *bimetal element* is composed of two thin strips of dissimilar metals fused together. Because the two metals have different coefficients of thermal expansion, the element bends as the temperature varies and produces a change in position. Depending on the space available and the movement required, it may be in the form of a straight strip, U-shaped, or wound into a spiral. This element finds its most common application in room thermostats, but is also used in insertion and immersion thermostats.

2. A *rod-and-tube element* consists of a high-expansion metal tube inside of which is a low-expansion rod with one end attached to the rear of the tube. The tube changes length with changes in temperature, causing the free end of the rod to move. The rod-and-tube element is commonly employed on certain types of insertion and immersion thermostats.

3. A *sealed-bellows element* is either vapor-filled, gas-filled, or liquid-filled, after being evacuated of air. Changes of temperature cause changes in pressure or volume of the gas or liquid, resulting in a change in force or a movement. This element is often employed in room thermostats.

A *remote-bulb element* is a sealed bellows or diaphragm to which a bulb or capsule is attached by means of a capillary tube, the entire system being filled with vapor, gas, or liquid. Changes of temperature at the bulb result in changes of pressure or volume which are communicated to the bellows or diaphragm through the capillary tube. The remote-bulb element is useful where the temperature measuring point is remote from the desired thermostat location. It usually is provided with fittings suitable for insertion into a duct or into a pipe or tank.

4. A *resistance element* is made of wire whose electrical resistance changes with temperature changes. It is used with electronic controllers. It is available in forms suitable for measuring room temperature or for insertion into a duct or into a pipe or tank.

Humidity measuring elements are made of (1) hygroscopic organic materials or (2) an electrical resistance.

1. An *organic element* usually is made of human hair, wood, paper, or animal membrane. Changes in relative humidity cause the element to expand and contract.

2. A *resistance element*, for use with electronic controllers, frequently consists of a thin coating of hygroscopic salt on an insulating form. The resistance of the salt varies with the relative humidity.

Pressure measuring elements can be divided into one of two general classes depending upon pressure range.

1. For pressures or vacuums measured in pounds per square inch or inches of mercury, the element usually is a bellows, diaphragm, or Bourdon tube. One side of the element may be open to atmosphere in which case the element responds to pressures above or below atmospheric. A differential-pressure element has connections to both sides so that it will respond to the difference between two pressures.

2. For low ranges of pressure or vacuum which are usually measured in inches of water, such as the static pressure in an air duct, the measuring element may be an inverted bell immersed in oil, a large slack diaphragm, or a large flexible metal bellows. The element usually is of the differential type, and when employed in conjunction with orifices, Pitot tubes, and similar accessories, may be used to measure flow, velocity, or liquid level as well as static pressure.

Measuring elements for other purposes, such as flame detection or for measuring smoke density, specific gravity, CO_2, CO, etc., often are necessary for the complete control of a heating, ventilating, or air-conditioning system.

Types of Controlling Elements

Controlling elements differ because some regulate the application of pneumatic energy while others regulate the application of electrical energy. They also differ according to the type of control action produced.

An *electric controlling element* is used in several of the the common types of control action.

For *two-position control* it may simply make and break an electrical contact, to start a burner, motor, etc., or to actuate a spring-return valve or damper operator. Another type makes one contact while breaking another and is used to control unidirectional valve or damper operators. Either type may be used for timed two-position action by the addition of a small resistance heater.

For *floating control* two contacts are employed, but there is a neutral zone between the two contacts where neither is made. This type is commonly used with reversible valve or damper operators.

For *proportional control* a contact is moved across a potentiometer to vary the voltage applied to a relay. The relay controls the operation of a reversible valve or damper operator and is rebalanced as the controlled device is positioned. In another arrangement the controlling element has two contacts and a neutral zone as described for floating control and is rebalanced by a small solenoid which is energized by a variable voltage feedback from the reversible valve or damper operator which it positions.

An *electronic controlling element* consists of an electronic amplifier and an electric relay or relays. The amplifier receives the minute electrical signals from the measuring element and amplifies them sufficiently to actuate the relays. The electronic controlling element may be two-position, floating, or proportional, depending upon the amplifier and relay arrangement.

An *electronic-pneumatic controlling element* consists of an electronic amplifier and an electropneumatic relay. It may be two-position, or proportional, depending upon the amplifier and relay arrangement.

A *pneumatic controlling element* is most often of the proportional action type but may also be used for the other types of control action. It may be classified as nonrelay type and relay type.

A *nonrelay type* of pneumatic controlling element is the simplest form and employs a restricted air supply and a nozzle or control port. The measuring element varies the opening of the nozzle and the resulting variable pressure is applied to the controlled device. This type is limited to applications requiring small volumes of air.

In a *relay-type* pneumatic controlling element the relay contains both a supply and an exhaust valve usually arranged so that the exhaust valve must close before the supply valve opens. Both valves are closed except when the control pressure is required to change. The variable force for operating the valves may be supplied directly by the measuring element or it may be supplied by a diaphragm actuated by air pressure from a very small nozzle and restrictor arrangement, this air pressure being varied by the measuring element. In either case the control pressure, operating on a diaphragm within the relay, produces a force which balances the force produced by the measuring element.

Pneumatic controllers may be either direct or reverse acting.

A *direct-acting controller* increases the control air pressure as the controlled variable increases. For example, a direct-acting thermostat increases the air pressure as the temperature increases.

A *reverse-acting controller* decreases the control air pressure as the controlled variable increases. A reverse-acting

thermostat decreases the air pressure as the temperature increases.

Indicating and Recording Features

Controllers may be of the indicating or recording type.

The *nonindicating controller* is most common in heating, ventilating, and air-conditioning work, and includes all those in which the measuring element does not provide a visual indication of the *value of the controlled variable*. If an indication is desired, a separate thermometer, relative humidity indicator, pressure gage, etc., is required. With room thermostats, for example, a separate thermometer often is attached to the cover.

An *indicating controller* has a pointer added to the measuring element or attached to it by a linkage so that the value of the controlled variable is indicated on a suitable scale.

A *recording controller* is similar to an indicating controller except that the indicating pointer is replaced by a recording pen which provides a permanent record on a chart driven by a clock motor.

Proportional band is a term used in connection with indicating and recording controllers and has the same meaning as throttling range. It is usually expressed in percent of the scale or chart range of the controller.

Types of Controllers

Controllers are mainly (1) thermostats, (2) humidistats, and (3) pressure types.

1. Many thermostats are of the simple, single-purpose type, but there are *many* others for various special purposes. Various types are indicated in paragraphs (*a*) to (*k*) which follow.

a. The *room* type is designed for mounting on a wall and responds to room temperature.

b. The *insertion* type is designed for mounting on a duct with its measuring element extending into the duct.

c. The *immersion* type is designed for mounting on a pipe or tank. It has a fluid-tight connection to allow the measuring element to extend into the fluid. A separable socket or well is often used with immersion thermostats to avoid the need for draining the system when removal of the thermostat is required. Since the separable socket will reduce the response rate of the thermostat, the socket should be no heavier than necessary and should fit snugly around the element.

d. The *remote-bulb* type is designed for applications where the point of temperature measurement is at some distance from the desired thermostat location. It is often used for centralized panel mounting of the controller. The remote-bulb element may be of either the insertion or immersion type.

e. The *surface type is* designed for mounting on and measuring the temperature of a pipe or similar surface.

f. The *day-night*, or *dual* room thermostat controls at a reduced temperature at night. It may be indexed (changed from day to night operation) *individually or in groups from* a remote point by a manual or time switch. Some electric types have an individual clock and switch built into the thermostat.

g. The *pneumatic day-night* thermostat uses a two-pressure air-supply system, the two pressures often being 13 and 17 psig or 15 and 20 psig. Changing the pressure at a central point from one value to the other actuates switching devices in the thermostat and indexes them from day to night or vice versa. Supply-air mains often are divided into two or more circuits so that switching can be accomplished in various areas of the building at different times, in accordance with use. For example, a school building may have *separate circuits* for classrooms, offices and administrative areas, auditorium, and gymnasium and locker rooms.

h. The *heating-cooling* or *summer-winter* thermostat can have its action reversed and, if desired, have its set point changed by the indexing means. It is used to actuate controlled devices, such as valves or dampers, that regulate a heating source at one time and a cooling source at another time. It is often indexed in groups manually by a switch or automatically by a thermostat measuring the temperature of the control agent, the outdoor temperature, or other suitable variable.

i. The *pneumatic heating-cooling* thermostat uses a two-pressure air-supply system as described for day-night thermostats.

j. A *two-stage* thermostat is arranged to operate *in two* successive steps.

k. A *submaster* thermostat has its set point raised or lowered over a predetermined range in accordance with variations in output from a master controller. The master controller can be a thermostat, manual switch, pressure controller, or similar device. For example, a master thermostat measuring outdoor air temperature can be used to readjust a *submaster* thermostat controlling the water temperature in a heating system. Master-submaster combinations are sometimes designated as *single-cascade action*. When such action is accomplished by a single thermostat having more than one measuring element, it is known as *compensated control*.

2. *Humidistats* are of either the room or insertion type. Submaster humidistats of either type are *sometimes* used in conjunction with an outdoor master thermostat to reduce the *humidity in cold* weather and prevent condensation on windows.

A *wet-bulb* thermostat is often used for humidity control in conjunction with proper control of the dry-bulb temperature. A wick or other means for *keeping* the bulb wet, and rapid air motion to assure a true *wet-bulb measurement* are essential.

3. A *pressure* or *static-pressure* controller is made for mounting directly on a pipe or remotely on a panel or wall. A small pipe or tube may be used for transmitting the pressure signals to the instrument.

Controlled Devices

Controlled devices may be (1) automatic valves, (2) valve operators, (3) automatic dampers, and (4) damper operators. Other controlled devices include electric heaters, and motors on such equipment as fans, pumps, burners, refrigeration compressors, and similar apparatus.

1. An *automatic valve* is designed to control the flow of steam, water, gas, and other fluids and may be considered as a variable orifice which is positioned by an electric or pneumatic operator *in response to impulses from the controller*. It may be equipped with a throttling plug or v-port of special design to provide the desired flow characteristics.

Renewable composition discs are common. They are made of materials best suited to the media handled by the valve, and its temperature and pressure. For high pressures or for superheated steam, metal discs are often employed. Internal parts of valves such as the seat ring, throttling plug or v-port skirt, disc holder, and stem sometimes are made of *stainless* steel or other hard and corrosion-resistant metals for use in severe service. Automatic valves should be properly selected and sized for the specific applications. See the section on Design Coordination in this chapter for *further details*.

The functions of various types of automatic valves are as follows:

a. A *single-seated valve* is designed for tight shut-off. Appropriate disc materials for various pressure ranges and media are used.

b. A *pilot piston valve* utilizes the pressure of the control agent as an aid in operating the valve. It usually is single-seated and is used where large forces are required for valve operation.

c. A *double-seated* or *balanced valve* is designed so that *the media pressure acting against the valve disc is essentially* balanced thereby reducing the force required of the operator. It is widely used where the fluid pressure is too high to permit a single-seated valve to close. It cannot be used where *tight shut-off is required*.

d. A *three-way mixing valve* has two inlet connections and one outlet connection and a double-faced disc operating *between two seats*. It is used to mix, as required, two fluids entering the inlet connections and leaving through the common outlet.

e. A *three-way diverting valve* has one inlet connection and two outlet connections and two separate discs and seats. It is used to divert the flow to either of the outlets or to proportion the flow to both outlets.

f. A *butterfly valve* consists of a heavy ring enclosing a *disc which rotates* on an *axis* at or *near* its center, and in principle is similar to a round single-blade damper. The disc usually seats against a ring machined within the body.

2. Valve operators are of three general types:

a. A *solenoid* consists of a magnetic coil operating a movable plunger. It is used for two-position operation. The most common use of a solenoid operated valve is to control the flow of gas, water, air, and refrigerants and generally it is limited to relatively small sizes.

b. An *electric motor* operates the valve stem through a gear train and linkage. Electric motor operators are classified in three distinct types: *unidirectional*, *spring return*, or *reversible*.

Unidirectional, for two-position operation. The valve opens during one-half revolution of the output shaft and closes during the other one-half revolution. Once started, it continues until the half revolution is completed, regardless of subsequent action by the controller. Limit switches built into the operator stop the motor at the end of each stroke. If the controller has been satisfied during this interval the operator will continue to the other position.

Spring-return, for two-position operation. Electric energy drives the valve to one position and holds it. When the circuit is broken, the spring returns the valve to its normal position.

Reversible, for floating and proportional operation. The motor can run in either direction and can stop in any position. It is sometimes equipped with a return spring. For proportional control applications, a potentiometer for rebalancing the control circuit is also driven by the motor.

c. A *pneumatic operator* consists of a spring-opposed flexible diaphragm or bellows attached to the valve stem in such a way that an increase in air pressure moves the valve stem and at the same time compresses the spring. Springs of various ranges in terms of air pressure can be employed to provide sequence operation of two or more devices by proper selection or adjustment of the springs. *Springless pneumatic operators*, using two opposed diaphragms or two sides of a single diaphragm, also are used but generally are limited to special applications involving large valves or high pressures. Pneumatic operators are designed primarily for proportional control. Two-position control is accomplished by use of a two-position controller, or a two-position pneumatic relay to apply either full air pressure or no pressure to the valve operator.

Pneumatic valves and valves with *spring-return* electric operators can be classified as normally open or normally closed.

A *normally open* valve will assume an open position when all operating force is removed.

A *normally closed* valve will assume a closed position when all operating force is removed.

3. *Automatic dampers* are designed to control the flow of air or gases and function like valves in this respect. Steel blades and frames are most common, but other materials are used for special applications such as those involving corrosive fumes. Large dampers often are made in two or more sections for strength and convenience in handling, the sections being interconnected so as to operate as a unit. The damper types commonly used are:

a. Single-blade, which is generally restricted to small sizes because of the difficulty of securing proper operation with high-velocity air.

b. Multiblade or *louver* damper which has two or more blades linked together. It may be arranged for *parallel operation*, in which all blades rotate in the same direction or *opposed operation*, in which adjacent blades rotate in opposite directions and provide more linear air-flow characteristics than obtained with parallel operation.

c. Mixing dampers, composed of two sections interlinked so that one section opens as the other closes.

4. *Damper operators* of the electric type may be unidirectional, spring-return, or reversible, similar to electric-motor valve operators. Pneumatic damper operators are similar to pneumatic valve operators except that they usually have a longer stroke or the stroke is increased by means of a multiplying lever. Damper operators are mounted on the damper frame and are connected with linkage directly to a damper blade or are mounted outside the duct and connected to a crank arm attached to a shaft extension of one of the blades. On large dampers two or more operators may be required, and *usually are applied at separate* points on the damper. Normally open or normally closed operation is obtained according to the method of mounting the operator and connecting the linkage.

Auxiliary Control Equipment

In addition to the conventional controllers and controlled devices described, many control systems require auxiliary devices to perform various functions. Auxiliary controls for electric systems include:

1. *Transformers* to provide current at the required voltage.

2. *Electric relays* for control of electric heaters or to start and stop oil burners, refrigeration compressors, fans, pumps, or other apparatus for which the electrical load is too large to be handled directly by the controller. Other uses include time-delay, circuit-interlocking safety applications, and similar functions.

3. *Potentiometers* for manual positioning of proportional control devices or for remote set-point adjustment of electronic controllers.

4. *Manual switches* for manually performing a variety of operations. These may be of *two-position or multiple-position* type or may be of single or multiple-pole type.

5. *Auxiliary switches* on valve and damper operators for providing a selected sequence of operations.

Auxiliary control equipment for *pneumatic systems* includes:

1. *Air compressors and accessories*, to provide a source of air at the required pressure.

2. *Electropneumatic relays*, which are electrically actuated air valves for operating pneumatic equipment in accordance with variations in *electrical output*.

3. *Pneumatic-electric relays*, which are actuated by air pressure and make or break an electrical circuit.

4. *Pneumatic relays*, which are actuated by the pressure from a controller to perform numerous functions. They may be divided into two groups:

a. Two-position relays, which permit a controller actuating a proportional device to actuate also one or more two-position devices. They also are used in various automatic switching operations.

b. Proportional relays, which are used to reverse the action of a proportional controller, select the higher or lower of two pressures, average two or more pressures, respond to the difference between two pressures, add or subtract pressures, amplify or retard pressure changes, and perform other *similar functions*.

5. *Positioning relays*, which are devices for assuring accurate positioning of a valve or damper operator in response to changes in pressure from a controller. They are affected by the position of the operator and the pressure from the controller, and whenever the two are out of balance will change the pressure applied to the operator until balance is restored.

6. *Switching relays*, which are pneumatically operated air valves for diverting air from one circuit to another or for opening and closing air circuits.

7. *Pneumatic switches*, which are manually operated devices for diverting air from one circuit to another or for opening and closing air circuits. They may be of two-position or multiple-position type.

8. *Gradual switches*, which are proportional devices for manually varying the air pressure in a circuit.

Auxiliary control devices common to both electric and pneumatic systems include:

1. *Sequence controllers* for operating a number of electric

switches in sequence by means of a proportional electric or pneumatic operator. They are commonly used for controlling a number of steps of refrigeration capacity and may be arranged to prevent simultaneous starting of compressors and to alternate the sequence to equalize wear. They may also be used for sequence operation of electric heaters and other equipment in response to the demands of a proportional controller.

2. *Clocks or timers* for turning apparatus on and off at predetermined times, for switching control systems from day to night operation, and for other time sequence functions.

PART II—DESIGN COORDINATION

A heating or cooling system is designed to meet maximum conditions but functions at partial capacity much of the time. The ability to properly control the system at all times is one of the most important factors having to do with the design of the system and for this reason the proper use and application of controls should be considered at the time the heating or cooling system is being designed. Wherever possible the system should be designed to facilitate control operation. This section includes factors which should be considered in the design of a new heating or cooling system, or the modernization of an existing one, to enable the control system to produce the best possible results.

The design of the system and its controls should take into consideration the nature and type of the building in which it is to be used and the results it is desired to accomplish. Quality of the control equipment should be consistent with the accuracy of control required. Where results more precise than those for normal comfort applications are required, such as in certain manufacturing areas, constant temperature rooms, etc., both the air-conditioning system and the control system should be given special consideration so that the required results can be produced. Controls can be made to produce almost any degree of accuracy required, but it is useless to provide such controls unless the air-conditioning system is capable of properly responding to the demands of the controllers. For example, it is virtually impossible to maintain close control of temperature and humidity by starting and stopping a refrigeration compressor or opening and closing a refrigerant valve. It is necessary instead to employ refrigeration equipment that will permit proportional control by one of the methods discussed in Chapter 38, or to use a chilled-water system with a cooling coil or air washer that will permit proportional control. Neither is it good practice or economical to select equipment capable of producing far more precise control than the application requires, or to complicate the system to obtain special sequences or cycles of operation when not necessary. It is well to remember that the system must be adjusted and maintained in operation for many years, and that the simplest system which will produce the necessary results is usually the best.

SIZE OF CONTROLLED AREA

No individually controlled area should be excessively large because the difficulties of obtaining good distribution and of finding a representative location for the space controls become greater as the area increases. Each individually controlled area must have similar load characteristics throughout and otherwise should conform to the recommendations given in the section on Zone Control. For uniform conditions throughout the area equitable distribution must be provided by competent engineering design, careful sizing of equipment, and proper balancing of the system. The control can measure conditions only at the point where it is located, and it cannot compensate for variable conditions throughout the area caused by improper distribution or inadequate design. Areas or rooms having dissimilar load characteristics, or having different conditions to be maintained, should be individually controlled. The smaller the controlled area the better will be the control obtained and the more nearly will the system approach the optimum in performance and flexibility.

Even when individual room control is employed it often is well to consider some form of overall zone control of the whole system in accordance with the major factor influencing system load. Thus the water temperature in a hot water heating system, the steam temperature or pressure in a steam heating system, or the delivered air temperature in a central fan system may be varied inversely as the outdoor temperature varies. This obtains a fair share of the necessary control on the system as a whole, relieves the individual space controls of a part of their burden, and makes more accurate space control possible. Furthermore, modifying the basic rate of heating or cooling input to the system in accordance with system load reduces losses in the distribution system. The cost of the additional controls is often justified by the reduction in operating costs.

Where such overall system controls are employed, accurate zoning, while desirable, is not necessary. The system must always be able to satisfy the area or room having the greatest demand. The individual controls take care of the variations in demand within the area served by the system. The more accurately the system is zoned the larger will be the share of the control accomplished by the overall system controls, the smaller will be the system distribution losses, and the more accurately will the individual space controls be able to maintain the space conditions.

EQUIPMENT SELECTION AND LAYOUT

Heating and cooling coils should be carefully selected to avoid oversizing. A coil that is too large is not only more difficult to control but will increase air temperature stratification. When temperature measuring elements are to be placed between coils, ample space between the coils must be provided. The measuring element should be located as near as possible to the coil it is controlling. For example, an idle water coil interposed between a thermostat and the coil it is controlling introduces enough thermal inertia to make it difficult to produce stable control.

The elimination of stratification in an air-handling system is essential not only in producing good control but in obtaining equitable distribution of heating or cooling. Outdoor-air and return-air dampers should be arranged in such a way as to help blend the outdoor air and return air. Face dampers should be placed far enough from coils to allow for equalization of air flow over the coil surface. The controller for face and bypass dampers must be located far enough downstream so that the air has become thoroughly mixed before reaching the controller. Other sources of stratification are coils which heat or cool unevenly and coils arranged in parallel and controlled in sequence. Whenever stratification cannot be eliminated by proper design or arrangement of equipment, special baffles, mixing chambers, or other mechanical devices should be used for this purpose. If stratification still exists at a point where the measuring element of a thermostat is to be located, the use of a long averaging element, or one made up of a number of small elements, may make a suitable temperature measurement possible. This does not solve the other problems relating

to stratification which can cause discomfort or, in extreme cases, freezing of equipment.

Good air distribution and properly spaced diffusers and grilles will greatly assist in maintaining even temperatures throughout an area. A space temperature controller cannot prevent drafts. This can be accomplished only by proper attention to such factors as balancing the system, by introducing air in the correct manner, and by providing a suitable low-limit control on the delivered-air temperature. A system that employs constant air volume with variable air temperature results in more uniform distribution than one which requires variations in air volume to achieve temperature control. Varying the volume of air affects both ventilation and distribution, two factors which usually have little relation to thermal requirements. Return-air grilles usually can and should be located so as to aid both the distribution and control of the system.

A further aid to both distribution and control is the provision of adequate facilities for exhausting air from the building. The larger the percentage of outdoor air introduced the more important becomes the provision for adequate exhaust. To prevent drafts due to infiltration, it is quite common to provide for positive exhaust of less air than is brought in. On systems where the quantity of outdoor air is large but constant, this may be accomplished by use of a properly sized exhaust fan. Systems having large internal heat gains, or large solar gains, may use a variable amount of outdoor air up to 100 percent when cooling is required during cool weather. In such cases the outdoor-air quantity may be controlled from space conditions and the quantity of exhaust air may be controlled by a static-pressure controller set to maintain the space under a slight positive pressure.

LOCATION OF SPACE CONTROLLERS

Space controllers such as room thermostats must be placed where they will measure the variables they are to control and where the condition is representative of the whole area served by the controller. If the return-air duct is used to represent space conditions, the controller should be located as near the space as possible to eliminate the effect of any heat gain or loss in the duct.

Room thermostats should usually be placed in occupied areas on inside walls or columns. They should never be placed in spaces such as corridors, lobbies, and foyers unless they are used for the control of these areas only.

In a typical home which is not zoned a satisfactory location for a room thermostat can usually be found on an inside wall of the living room or dining room. It should always be located toward the center of a relatively open room situated on the coolest rather than the warmest side of the building. It should not be mounted on an outside wall or other cold surface where it is exposed to cold drafts from an outside door. Neither should it be mounted where it will be affected by direct rays of the sun, by heat from a nearby warm surface such as a chimney, pipes or ducts in the wall, or radiators, or by direct air currents from a register. The location should provide ample air circulation unimpeded by furniture or other obstructions, and should afford protection from mechanical injury.

CONTROL OF FLOW

The successful operation of practically any air-conditioning system is dependent on the proper control of flow in one form or another. The flow being controlled may be steam, water, air, or some combination thereof. In each case a controlled device is used to regulate the flow at the command of a controller. The proper design and sizing of these controlled devices, valves for steam or water, and dampers for air, are essential to obtain the desired results from the air-conditioning system.

The performance of a valve or damper is expressed by the control industry in terms of its flow characteristics as it operates through its stroke, based on a *constant pressure drop*. Three common characteristics are shown in Fig. 6 and may be defined as follows:

Quick Opening: Maximum flow is approached rapidly as the device begins to open.

Linear: Opening and flow are related in direct proportion.

Equal Percentage: Each equal increment of opening increases the flow by an equal percentage over the previous value.

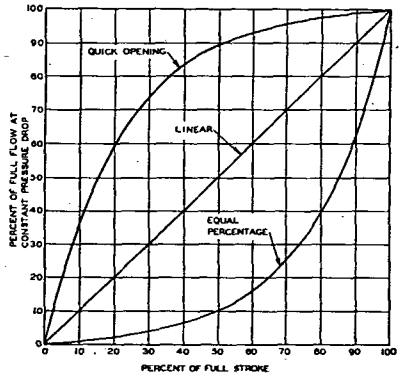


Fig. 6 .... Typical Flow Characteristics

Since the pressure drop across a valve or damper seldom remains constant as its opening changes, actual performance usually deviates from the published characteristic curve. The magnitude of the deviation is determined by the overall system design. For example, in a system arranged so that control valves or dampers can shut off all flow, the pressure drop across a controlled device increases from a minimum at design conditions to the total system pressure drop at no flow. Fig. 7 shows the extent of the resulting deviations for a valve or damper designed with a linear characteristic, when the selection is based on various percentages of the total system pressure drop. In order to approximate the designed characteristic of the valve or damper, the design pressure drop should be a reasonably large percentage of the total system pressure drop or the system should be designed and controlled in such a manner that this pressure drop remains relatively constant.

Higher pressure drops for controlled devices are obtained by using smaller sizes in conjunction with a possible increase in size of other equipment in the system. Since

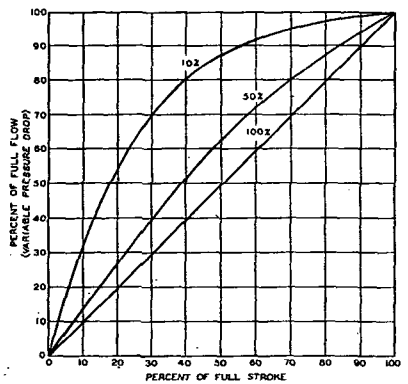


Fig. 7.... Typical Performance Curves for Linear Devices Selected at Various Percentages of Total System Pressure Drop

sizing techniques are different for steam, water, and air, each is covered separately.

THE CONTROL OF STEAM FLOW

The size of an automatic steam valve is based on the required capacity at maximum load, under the conditions which will exist at that time. Equation 52, Chapter 4, for mass flow through an orifice or nozzle can be rewritten as

$$w = CA\sqrt{P_1 - P_2} \qquad (1)$$

where

- w = the flow of steam in pounds per hour.
- A = the area of the port opening in square inches.
- P_1 = the steam supply pressure, pounds per square inch, absolute.
- P_2 = the downstream pressure, pounds per square inch, absolute.
- C = a constant for a given steam supply pressure and a given valve body.

Valves for two-position operation usually are selected of a size adequate to supply the required maximum capacity with a reasonably low pressure drop ($P_1 - P_2$). It is necessary only to be sure that the valve will close against the system pressure and will have ample capacity when fully open.

Proportional valves for steam service must be carefully sized if stable control is to be achieved. Since, in general, the tendency is to oversize such valves it should be remembered that no controller can produce its best results if handicapped by a valve which is too large. The function of the valve is to vary the value of area A in Equation 1 and it is desirable that the flow vary in proportion to the value of A. The correct maximum required capacity must first be determined. There should be no factor of safety in this determination, and it should be based on actual heating requirements rather than on the condensing capacity of the coil, since the coil may be larger than necessary. It is also necessary to base the valve size on the actual supply pressure P_1 which will exist at the valve, since this supply pressure affects both the density of the steam and the pressure drop which will be selected.

Selecting the proper pressure drop ($P_1 - P_2$) across the valve when it is wide open and is supplying the maximum steam requirement is the most important consideration in determining the valve size. For purposes of discussion it will be assumed that P_1 is constant and that the valve is supplying steam to a heating coil which has air entering at a constant temperature and velocity. Any pressure drop between the valve and the coil will be neglected so that P_2 represents the pressure in the coil. The condensing capacity of the coil varies with changing values of P_2. When the valve is fully open, P_2 assumes a value such that the capacity of the valve and the condensing rate of the coil are in balance. If this value of P_2 is relatively large, then, as the valve is partially closed, P_2 will decrease until the capacity of the valve and the condensing rate of the coil are again in balance. Reducing the value of A in Equation 1 results in an increase in the value of $\sqrt{P_1 - P_2}$ and partially offsets the effect of closing the valve. This effect is greatest when P_2 is nearly equal to P_1 and diminishes as P_2 decreases.

The minimum value which P_2 can assume is established by the return line pressure (or vacuum) of the system. However, P_2 may reach a value which results in the critical velocity through the valve before it has been reduced to its minimum value. If so, any reduction of P_2 beyond this value has no further effect on the valve capacity. The critical velocity exists when P_2 is approximately 58 percent of P_1. The undesirable effect caused by variations in P_2 can be eliminated by selecting a valve size such that P_2 is at or near its minimum value, or the value corresponding to the critical velocity, whichever is higher, when the valve is wide open.

On low-pressure steam applications it is good practice to base the valve size on a pressure differential equal to 75 or 100 percent of the gage supply pressure. When the supply pressure is high enough so P_2 at the critical velocity (58 percent of P_1) is above atmospheric pressure, the valve size can be based on a pressure differential of 42 percent of P_1 absolute. All valves of the same size do not have the same capacity for a given pressure differential. Having established the supply pressure and the pressure differential, the valve manufacturer's tables or charts should be consulted in selecting the proper size of valve for the required capacity.

It is important that the supply pressure to an automatic steam valve be constant because it is difficult for a thermostat to produce stable results if the pressure is constantly changing. This situation arises, for example, when a steam boiler is operated by a two-position pressure controller. A pressure-reducing valve, which will supply a constant pressure to the steam mains while permitting the boiler pressure to fluctuate between the limits of the pressure controller, is recommended. Lacking this, the differential of the pressure controller should be set as narrow as is consistent with efficient boiler and burner operation. On vacuum return systems the vacuum pump also should operate on a narrow differential.

The use of two automatic control valves in parallel will often provide more stable control than a single larger valve, especially on large coils or steam-to-water heat exchangers. The valves should be sized so that one valve handles about one-third of the total capacity and the other about two-thirds. They are controlled in sequence so that the small valve operates during periods of light load and the large valve begins to open only after the load exceeds the capacity of the small valve.

The steam piping from the valve to the coil inlet, and from the outlet to the trap should be the full size of the coil tapping or size specified by the manufacturer of the coil and should conform to good standard practice as illustrated in Chapter 26. It is very important that condensate be promptly drained from the coil. When the coil is at reduced pressure, as it will be much of the time, there will not be sufficient difference in pressure between the coil and the return line to obtain any lifting of the condensate.

THE CONTROL OF WATER FLOW

The capacity of a proportional valve for water service is not influenced by the supply water pressure but only by the pressure differential. This is so because water is incompressible and because the inlet pressure is not an important factor in the thermal capacity of a water system. If a certain water circuit has a total resistance of 30 psi at maximum flow, of which 1 psi is through the valve, the resistance of the rest of the circuit is 29 psi. As the valve closes and flow is stopped the resistance across the valve becomes the entire 30 psi. Thus the pressure drop across the valve increased by a ratio of 30:1 as the valve travelled from full open to full closed position. This increase in pressure drop as the valve is closed offsets partially the effect of closing the valve.

It is obviously impracticable to design a water system such that there will be no increase in pressure drop across the valve as it closes. However, it is important to design the system, including the valve, in such a manner that the ratio of this increase in pressure drop as the valve closes is kept as low as possible. Thus, if the design pressure drop across the valve is 5 psi instead of 1 psi, the ratio of increase is only 6:1 instead of 30:1 and the performance of the valve will be improved accordingly. This discussion assumes a circuit in which the entire flow is controlled by a single valve. The same reasoning will apply to the more complicated systems involving a multiplicity of parallel circuits, each having a valve. In general it can be said that the pressure drop across a water valve at full flow should be a reasonable percentage of the resistance of the circuit in which the valve controls the flow. Due consideration should be given to any resultant increase in pumping head and operating costs.

Consideration must also be given to the fact that the Btu release from the hot water coil does not vary in direct proportion to changes in flow, i.e., as flow is reduced the water temperature drop through the coil increases so that there is very little change in the total heat given off. Conversely, this is true for cold water coils. Considerable improvement in control valve operation is obtained where systems are designed for large temperature drop and valves are selected at minimum practical size. Special designs of throttling plugs in valves may further improve control.

Water circuits involving variable flow are subject to variable pressures in accordance with the pressure vs. flow characteristic of the pump. On such circuits it is desirable to provide some means of system pressure control. A bypass from supply to return is often used to accomplish this. The bypass connection either has a spring-loaded relief valve, subject to the system pressure difference, or an automatic control valve operated by a differential-pressure controller.

Another very important consideration when using hot water as the heating medium, is that the heat output of a coil is not linearly related to flow. As the flow is reduced,

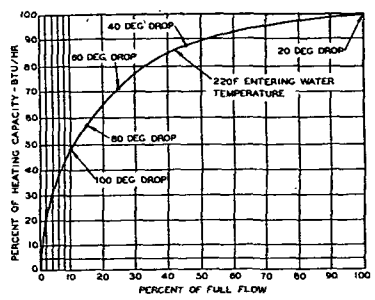


Fig. 8.... Heat Emission vs. Flow Characteristics of Typical Hot Water Heating Coil

a greater amount of heat is transferred from each gallon of water and this tends to counteract the reduction in flow. Fig. 8 shows the heat emission versus flow characteristics for a typical coil with 220 F entering water and a 20-deg design temperature drop and shows how the temperature drop increases as the flow is reduced.

To improve the coil characteristics for hot water, it is possible to design for higher water temperature drops. Selecting a temperature drop of 60 deg would require a slightly larger coil but would improve control performance and substantially lower the water requirements. This reduction in gpm also results in smaller pipe sizes, smaller control valves, and reduced pumping costs.

A properly sized control valve with equal percentage characteristics should be used to control the flow of hot water to coils for modulating service.

CONTROL OF AIR FLOW

It is common practice to size a damper to fit the duct into which it is installed. This usually results in a very low design pressure drop across the damper. It also keeps the fan horsepower requirements as low as possible. In some cases this practice results in very poor control performance. This is particularly true of dampers used to control the entire flow into a large duct, such as a static pressure damper.

For a constant pressure drop, the flow characteristic of a damper varies with the number of blades, the manner in which the blades rotate with respect to each other, and the length to width ratio of blades.[1] Dampers commonly are of the single-blade or multiblade type. The latter may have blades with either parallel or opposed operation.

When the application is such that the pressure drop across the damper increases substantially as it closes, it is preferable to use opposed blade dampers. As explained in a preceding section, Control of Flow, the changing pressure will cause a deviation from the design flow characteristic. This will change the performance from nearly equal percentage to nearly linear as shown in Fig. 9.

Even though Fig. 9 indicates good performance from an opposed blade damper under the conditions tested, it does not eliminate the need for using smaller dampers with higher design pressure drops in some applications. An extreme example is a static-pressure damper in a high-velocity system. Such a damper should not be placed in the low-

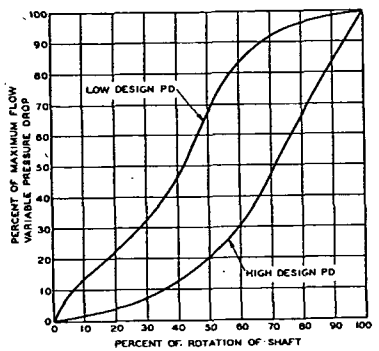


Fig. 9 .... Effect of Pressure Drop Selection on Opposed-Blade Damper Performance

velocity coil section where it would have a design pressure drop of less than 0.1 in. water, if it is expected to control with 2 to 3 in. pressure drop during periods of low flow. It should be placed in a high-velocity section where the design pressure drop will be 0.5 in. water or more, careful consideration being given to avoid excessive turbulence on either entering or leaving side of the damper.

ZONE CONTROL

Zone control for any heating, ventilating, or air-conditioning system is employed where it is desired to control, by one set of controls, the heating or cooling effect in a number of rooms or areas, having similar orientation or occupancy. For zone control to be successful the requirements must be approximately consistent throughout the extent of the zone. Whether zoning a building for heating or cooling, the following factors should be considered in determining the number and arrangement of the zones:

1. *Exposure.* Solar effect, prevailing winds, and the shelter afforded by surrounding structures affect the heat gain or loss.

2. *Occupancy.* Indoor temperature requirements for the various activities carried on in different portions of the building and the hours of occupancy of the various spaces impose definite control problems.

3. *Building Structure.* The physical characteristics of the building will influence the subdivisions of the system into zones since satisfactory temperature conditions throughout a single zone of given extent may not be obtainable equally in buildings of dissimilar types of construction. Other factors are the height of the building and its horizontal extent and form.

4. *Floors.* Even though several floors have the same wall exposure, it is advisable to have separate zones on the lowest and highest floors due to variations of basement or ground floor requirements and the effect of roof exposure. It is also desirable to have a separate zone or zones for the intermediate floors.

The initial investment will often influence the decision as to the final number of zones to be employed. In larger buildings accepted practice is to have at least one zone for each exposure. Each exposure may require subdivision vertically into two or more zones for the higher structures. Also, the presence of two or more wings having the same general exposure may suggest the desirability of more restrictive zoning. When the street floor or any other portion of the building is used for public occupancy or activities which differ from those carried on in the remainder of the building, it is desirable to provide separate thermostats and controls for each individual area.

Zone controls alone may not provide satisfactory temperature conditions in all rooms or areas within the zone because occupancy, lighting load, space arrangements, and similar factors cannot always be predicted with accuracy. A combination of zone controls with individual room control in critical areas may be necessary to achieve complete satisfaction.

INDIVIDUAL ROOM CONTROL

The ideal temperature control system for any building is one that promotes maintenance of the desired temperature in every room at all times regardless of location and occupancy. Individual room temperature control is desirable for securing proper thermal environment in schools, hospitals, and offices.

Control of the temperature in each room, or possibly of adjacent rooms having the same orientation (as in an apartment), overcomes many of the problems encountered when attempting to regulate the temperature of a building as a whole or of large areas or zones. Each individual thermostat controls the heat input to its particular space regardless of occupancy or exposure to sun and wind. The advantage of individual room control is in fuel economy and comfort for the occupants. Each room should have a thermostat that controls valves or dampers on all the controllable sources of heating and cooling. One such uncontrolled source may prevent proper operation of the control system.

PART III—CONTROL APPLICATIONS

Applications of controls to systems or apparatus of a general nature will be found in this section. Applications pertaining to specific systems or apparatus will be found in the chapters covering these subjects. The application of controls may conveniently be considered for the three divisions of a system indicated in Fig. 10: (1) *primary source apparatus*, (2) *distribution system*, and (3) *terminal equipment*.

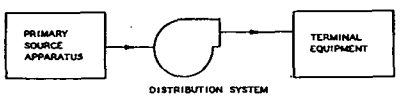


Fig. 10 .... Three General Divisions of Control Applications

PRIMARY SOURCE APPARATUS

The control of *primary source apparatus* is described in the following chapters: Chapter 18, Warm Air Heating Systems, Chapter 34, Automatic Fuel-burning Equipment, Chapter 39, The Heat Pump, and Chapter 38, Refrigeration. The second and third divisions are considered in this section.

DISTRIBUTION SYSTEMS

The control of distribution systems is discussed separately for central fan systems, hot water systems, zoned steam systems, and zoned hot water systems, due to the difference in these systems.

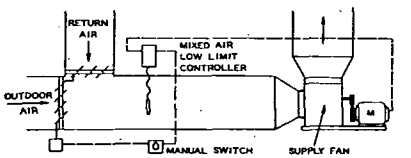


Fig. 11 .... Manual Adjustment of Outdoor-Air Quantity With Low-Limit Control of Mixture Temperature

CENTRAL FAN SYSTEMS

Refer to Chapter 19 for additional material on control for a typical central fan system.

Since there are numerous control combinations that can be employed with central fan systems, the following sections will deal with only a few of the typical combinations for controlling outdoor- and return-air dampers, preheat coils, heating coils, cooling coils, and humidity control. The complete control system is obtained by combining the selected method for each item into an integrated set of controls. Proper consideration must be given to the interrelation of the several parts and the sequence of operation, etc.

Outdoor-Air Damper Control

Outdoor air for ventilation is usually controlled by a damper arrangement providing a homogeneous mixture of outdoor air and recirculated air. Outdoor air to meet minimum ventilation requirements should be available whenever the fan is running and is usually provided in one of the following ways:

a. By a minimum outdoor-air damper which is fully open when the fan is started. A maximum outdoor-air damper may also be provided if additional outdoor air for natural cooling is desired.

b. By an outdoor-air damper which opens to a minimum position when the fan is started. Additional outdoor air is provided by further opening of the damper under automatic control.

Recirculated-air and exhaust-air dampers are synchronized to operate with the outdoor-air dampers. Mechanical interconnections are preferable but separate operators on each of the dampers can be used when mechanical interconnections are impractical. In most arrangements provisions are made to prevent admission of more than the minimum outdoor air required for ventilation both during heating operation and when the outdoor-air temperature is too high to provide some natural cooling. Provisions should also be made to close the outdoor-air and exhaust-air dampers and open the return-air damper whenever the fan is stopped.

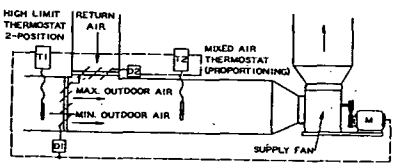


Fig. 12 .... All-Season Ventilation Control With Fixed Minimum Outdoor Air Plus Additional Outdoor Air When Practicable

Fig. 11 shows a system having manual adjustment for maximum outdoor air and having an optional, but very desirable, mixed-air controller. This controller overrides the manual switch whenever the mixed-air temperature drops below its set point. This provides for a minimum mixed-air temperature for safety of downstream equipment. Whenever the fan motor is stopped the outdoor-air and exhaust-air dampers close while the return-air damper opens.

Fig. 12 shows an all-season ventilation control system with a fixed minimum quantity of outdoor air and automatic control of maximum outdoor- and return-air dampers. This arrangement is sometimes referred to as an economizer control of outdoor air. With outdoor-air temperatures below the set point of the summer controller, the mixed-air controller positions the outdoor- and return-air dampers. When outdoor-air temperature rises to the set point of the outdoor high-limit controller, usually about 70 F, the maximum outdoor-air damper is closed and the return-air damper opened. The minimum quantity of outdoor air is assured by the separate minimum outdoor-air damper. Both outdoor-air dampers are closed and the return-air damper opened whenever the fan motor is stopped. By adding a thermostat in the conditioned space or in the return-air duct the opening of the maximum outdoor-air damper can be limited to only the desired minimum quantity of outdoor air until the space heating requirements are satisfied. This feature is desirable for quick warm-up during winter heating.

Preheater Coil Control

The function of a preheater is to temper the outdoor air to prevent the possibility of freezing beyond the coil. Care must be taken to avoid freezing in the coil itself. The proper selection, installation, and sizing of steam traps are very important to assure rapid elimination of condensate to prevent freezing. Other important considerations are vacuum breakers to insure condensate removal, provisions for rapid air elimination from coil, and the use of vertical rather than horizontal tubes wherever possible.

Fig. 13 shows a simple application of an outdoor thermostat usually set for 35 F controlling the preheat coil valve. Careful sizing of the coil is very important with this type of control. The temperature rise through the coil should be limited to a maximum of 30 deg when the entering air is 35 F. A higher temperature rise will very likely cause overheating of the space. If more than a 30-deg temperature rise is needed then multiple coils should be used and their control valves operated in sequence at selected outdoor temperatures.

Fig. 14 offers a method of control which reduces the

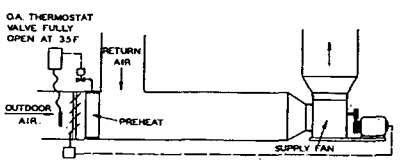


Fig. 13 .... Control of Preheat Coil From Outdoor Air

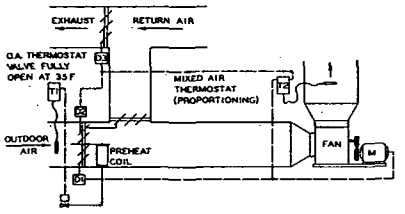


Fig. 14 .... Control of Preheat Coil From Outdoor Air With Damper Control From Mixed Air

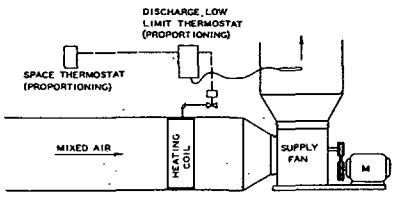


Fig. 16 .... Control of Heating Coil From Space and Discharge Air Using Valve

overheating problem outlined for Fig. 13. Here a mixed-air thermostat operates an outdoor-air bypass damper in conjunction with the exhaust- and return-air dampers. The use of bypassed outdoor air makes it possible to control the mixed-air temperature without endangering the preheat coil. A minimum quantity of outdoor air is assured by use of a two-position face damper on the coil.

Steam-distributing type coils can be controlled by a proportional-type thermostat set for 55 F or higher and located downstream from the coil to operate a valve in the steam supply line. Where a temperature rise of more than 30 deg is required, two or more coils in series should be used, each having a separate valve. This arrangement is shown in Fig. 15. The valves should be arranged to operate in sequence, with the valve on the upstream coil *opening first*.

Hot water preheat coils should be controlled by face and bypass dampers operated by a thermostat downstream from the coil, with continuous water circulation when the outdoor air is below freezing. A pressure switch to stop the fan in case of pump failure is often employed. Further freeze protection can be accomplished by a thermostat in the coil outlet arranged to stop the fan if the water temperature drops to 35 F. In all applications where face and bypass dampers are used, precautionary measures must be taken to prevent stratification of the mixture beyond the coil. One method is to locate the bypass damper above the face damper so that the cooler bypassed air will tend to drop while the warmer air through the face damper tends to rise thereby mixing the air by gravity.

Heating Coil Control

Heating coils when used for tempering air for ventilation can be controlled by a proportional type of insertion thermostat, preferably located in the fan discharge where the air is usually least stratified. This thermostat operates the heating coil valve or valves, or face and bypass dampers, to maintain a constant discharge air temperature. When face and bypass dampers are used it is good practice to have a valve on the coil which closes as the face damper closes to prevent overheating due to radiation from the coil, damper leakage, or escape of heat from the coil.

Fig. 16 shows a typical control arrangement for a heating coil when the fan system is used for heating as well as for ventilation. Here a room or a return-air thermostat controls the heating coil valve until the space temperature is satisfied. A discharge duct thermostat then assumes control of the valve to maintain the desired minimum air discharge temperature. With this system no portion of the air entering the coil should be less than 32 F.

Fig. 17 shows an arrangement using face and bypass dampers on the heating coil and two-position control of the valve. It should be noted that without a low-limit controller the air entering temperature must be the desired minimum discharge air temperature.

Fig. 18 offers a typical water coil control application. The room or the return-air duct thermostat operates the three-way mixing valve until the space temperature is satisfied. The discharge duct thermostat then takes over the valve control to maintain the desired minimum air discharge temperature.

Cooling and Dehumidification Coil Controls

Dehumidification by condensation is so closely related to sensible cooling that it is convenient to discuss them together. Three typical applications of controls are discussed.

Direct-Expansion Systems. A thermostat measuring the space temperature may be used to open and close a solenoid refrigerant valve on the coil inlet; control the starting

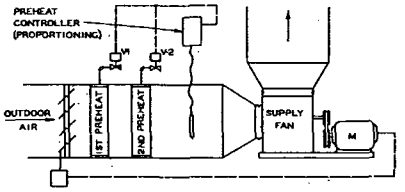


Fig. 15 .... Proportional Control of Two Preheat Coils in Sequence

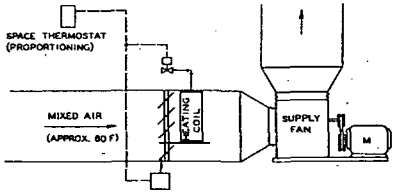


Fig. 17 .... Control of Heating Coil From Space Temperature Using Valve and Face and Bypass Dampers

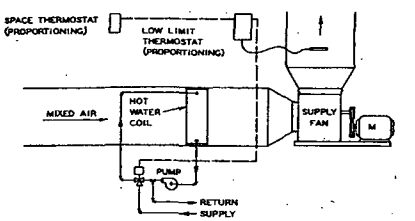


Fig. 18 .... Control of Hot Water Coil From Space and Discharge Air Using Mixing Valve.

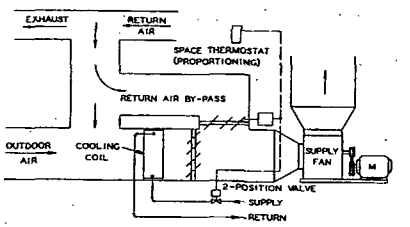


Fig. 20 .... Control of Chilled Water Coil From Space Temperature Using Return Bypass Damper and Valve

and stopping of one or more refrigerating compressors, or otherwise control the compressor capacity; position face and bypass dampers, or a return-air bypass damper, in combination with the opening and closing of the refrigerant coil valve or the starting and stopping of the compressor; or control a back pressure regulating valve with proportional action. The positive opening and closing of the refrigerant valve or the starting and stopping of the refrigeration compressor will result in wide variations in discharge temperature and dehumidification. When such variations are objectionable, it is desirable to employ proportional damper controls or proportional control of refrigerant flow in conjunction with refrigeration equipment capacity controls as discussed in Chapter 38.

Chilled-Water and Brine Coils. A thermostat measuring the space temperature may be used to control a valve on the inlet or outlet of the cooling coil; control a three-way mixing valve; position face and bypass dampers at the coil; or position a return-air bypass damper. An apparatus dew-point thermostat may also be used to control the cooling coil valve. When cooling capacity is controlled only by a dry-bulb thermostat the amount of dehumidification will vary with the sensible cooling demand. When independent dehumidification is required it is necessary to employ some form of reheat.

Fig. 19 shows a simple arrangement where chilled water is supplied to the coil at a constant temperature and the water volume through the coil is varied. The space or return-air thermostat modulates the three-way valve to *increase or decrease the volume of water through the coil.* Although this arrangement provides constant flow through the chiller, it should be noted that throttling of chilled water to the coil causes higher coil temperatures with light sensible cooling loads which may result in unsatisfactory humidity conditions.

Fig. 20 shows a control system using two-position control of a cold water coil valve and modulating control of a return-air bypass damper. Here the cooled air through the coil is mixed with bypassed return air to satisfy the space thermostat demands. When the return-air bypass damper is fully opened the water valve can be closed. It is often desirable to omit the valve control and instead maintain a constant flow of chilled water through the coil. This provides some dehumidification with light sensible cooling loads.

Air Washers. Since the air leaving an efficient washer is practically saturated, the temperature of the spray water will *determine the dew-point temperature of the air leaving* the washer. With this condition it is possible to maintain very accurate dew-point temperatures and consequently accurate relative humidities under a given ratio of sensible-to-latent heat gain in the conditioned space, provided the space dry-bulb temperature is accurately controlled. In general, the water pump operates continuously and a thermostat measuring the dew-point temperature is used to position a three-way valve to mix refrigerated water with bypassed washer water. To compensate for variations in latent load, a humidistat measuring the space relative humidity may be employed as a master controller to readjust the set point of a dew-point submaster controller.

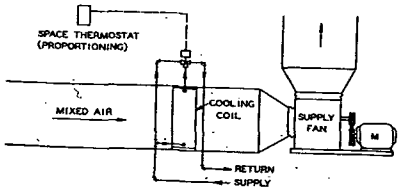


Fig. 19 .... Control of Chilled, Water Coil From Space Temperature Using Mixing Valve

Humidity Controls

Fig. 21 shows a simple method of humidity control employing a room or a return-air duct humidistat to control a valve supplying steam to an evaporator pan type humidifier. Additional control provides for closing the valve

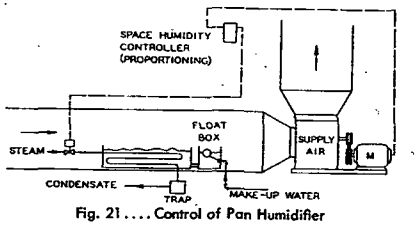


Fig. 21 .... Control of Pan Humidifier

when the fan is stopped. This prevents evaporation of water which might later condense in the ductwork and cause damage. Since untreated city water is evaporated there are none of the objectionable odors usually caused by chemically treated boiler water. Such odors often are noticeable with steam jet humidifiers.

Control of steam jet humidifiers is the same as for pan humidifiers.

Control of atomizing water spray humidifiers also is the same as for pan humidifiers except that two-position valve action must be employed. This action is necessary in order to obtain efficient atomization.

Fig. 22 shows an example of winter dew-point control of an air washer humidifier. A space or return-air humidistat readjusts the set point of the dew-point controller to compensate for latent load changes. If the latent load of the space is constant then the humidistat can be omitted in which case the dew-point controller would control at a constant set point. The dew-point controller increases the temperature first by closing the maximum outdoor-air damper and then by gradually opening the water heater valve.

Static-Pressure Control

Since fans are selected to produce the desired results for a certain system characteristic, any change in the system characteristic will change the output of the fan. Such changes occur, for example, on single-duct systems in which the volume of air delivered is varied and in double-duct systems where quantities of air are shifted from one duct to the other to obtain final space control. System characteristics may also change when the relative quantities of outdoor air and recirculated air are varied, when exhaust fans are run intermittently, and for other reasons. Such changes result in a changing static pressure in the system, a changing noise level, and changing velocities at air outlets which upset the air-flow patterns.

Static-pressure control of some form is usually required on such systems, and is accomplished by a static-pressure controller measuring the pressure at a selected point in the system with respect to an atmospheric reference point or some other point inside the building. It can be used to actuate a damper operator on fan inlet vanes, on a multiblade damper on the suction or discharge side of the fan, or on a damper in a bypass connection between the fan discharge and the fan suction.

Fig. 23 shows a simple method of static-pressure control. Here a differential-pressure controller measures the static pressure in the supply duct in relation to some reference space such as outdoors. With increase in duct static pressure due to throttling of space volume control dampers the controller throttles the static-pressure damper proportionately. For controlling system static pressure, louver dampers may be installed on either the suction or discharge side of the fan or fan inlet vanes may be employed. To control static pressures in various sections of a distribution system, louver dampers are installed in the inlets to the various sections, with separate static-pressure controllers for each section. Measuring tubes must be located within the system or section at a point where constant pressures are most desirable.

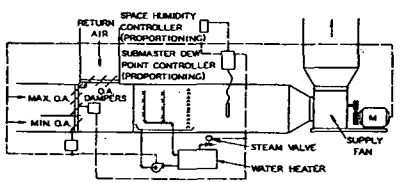


Fig. 22 .... Control of Air Washer Humidifier With Submaster Dew-Point Controller

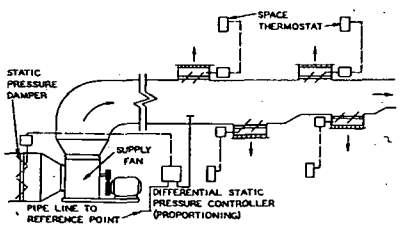


Fig. 23 .... Static-Pressure Control in Variable Volume System

Control of a Typical Central Fan System

Fig. 24 shows an example of control arrangement using some of the system components previously described and referred to in Part II of this chapter under Design Coordination. The controls can be pneumatic, electric, or electronic.

When the fan is started relay *E-1*, actuated by the fan-motor starter, opens outdoor-air damper *D-1*, places humidistat *H* in service, and allows insertion thermostat *T-1* to operate preheat coil valve *V-1*, maximum outdoor-air damper *D-2*, return-air damper *D-3*, and exhaust damper *D-4*. When the fan stops, *E-1* is de-energized thereby closing dampers *D-1*, *D-2*, and *D-4* while opening *D-3*. Humidifier valve *V-4* is closed and preheat valve *V-1* is opened.

The sequence of operation described will in general also apply to high-pressure fan systems except that it is considered desirable to open the outdoor-air damper before starting the fan. With this arrangement, a manual switch operates damper *D-1*. An auxiliary switch mounted on the damper starts the fan when the damper is *fully* open. It also stops the fan when the damper *starts* to close.

Insertion thermostat *T-1* maintains a temperature of 55 F at the preheat coil discharge by operating preheat coil valve *V-1*. This serves as a low limit for the fan discharge temperature during heating operation. When *V-1* is closed, thermostat *T-1* then gradually opens maximum outdoor-air damper *D-2* and closes return-air damper *D-3* proportionately. At 55 F outdoor temperature *D-2* is wide open and *D-3* is closed. Exhaust-air damper *D-4* opens and closes in synchronization with *D-2*. At about 70 F outdoors, when the outdoor air is no longer useful for natural cooling, *T-2* closes *D-2* and *D-4* and opens *D-3*.

Reheater coil valve *V-3* is controlled by submaster thermostat *T-3* in the fan discharge. This thermostat is automatically readjusted over an appropriate range of temperature by thermostat *T-4* so that the discharge temperature matches the space requirements. When the space temperature tends to rise above a predetermined value, *T-3* has been readjusted downward far enough to keep reheat valve *V-3* closed. *T-4* then controls cooling coil valve *V-2* directly to maintain the desired space temperature.

Humidistat *H* operates humidifier valve *V-4* to maintain the desired humidity in the conditioned space. It is usually adjusted to maintain 30 to 40 percent relative humidity, and therefore keeps the humidifier off during cooling operation when the humidity is higher.

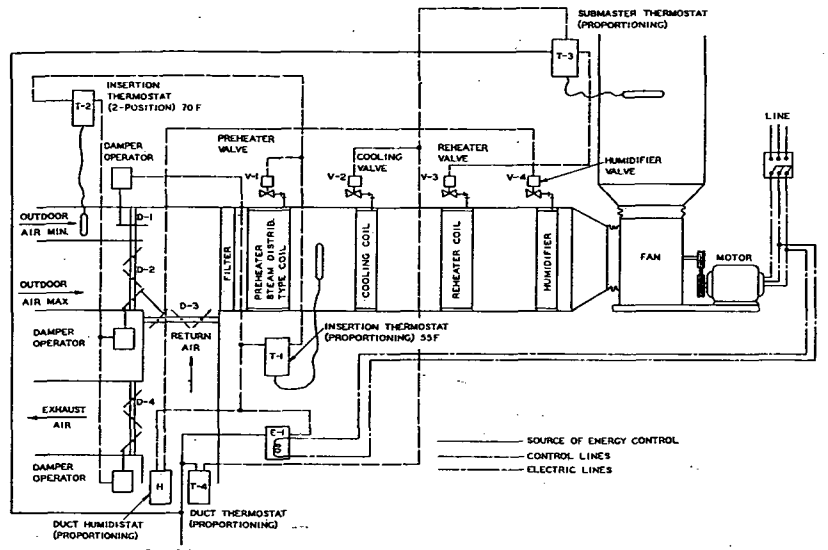


Fig. 24 .... Control Diagram for a Typical Central Fan Air-conditioning System

TEMPERATURE CONTROL IN HOT WATER SYSTEMS

Hot water heating systems permit use of relatively simple automatic temperature control for the purpose of reducing operating costs and providing comfortable temperatures. On installations not requiring different supply-water temperatures simultaneously and where the boiler is not used to heat domestic hot water, outdoor reset control of the supply-water temperature may be obtained by direct control of the burner from a thermostat that is located in the boiler water and is reset in accordance with outdoor temperatures. A seperate high limit control should be provided.

In systems using converters for a hot water supply, a controlling thermostat actuated by the leaving water temperature modulates the heat source to the converter. The set point of this supply-water thermostat can be readjusted in accordance with outdoor temperature.

In some systems a boiler or converter supplies water at a constant temperature, and three-way mixing valves are used to supply water at different temperatures for various uses or to multiple zones. In these cases, a three-way mixing valve under control of a thermostat in the valve discharge blends boiler water with return water to provide supply water at the desired temperature. The control setting of this thermostat in the valve discharge can be reset in accordance with outdoor temperatures.

On systems using outdoor-reset supply water and lowered night or week-end temperatures, provision should be made for warmup on restoration to day or normal operation. This can be accomplished by a program control, a warmup space thermostat, or some similar means of providing higher than normal water temperatures during the warmup period.

A measure of control can also be accomplished by use of a room thermostat to operate the burner. This type of control is applicable only to residences or very small commercial installations, where there is no great difference in heating requirement throughout the building. Where a difference in heating requirements throughout the building does exist, it is advisable to control the supply-water temperature with provision for automatic control or manual adjustment of heating unit output in individual spaces.

Economical zone control can be obtained by separate return or supply piping for zones, with a zone thermostat to operate zone pumps or automatic valves. The piping for each zone should be isolated to avoid a common return line, because in many cases the return water from one zone is warmer than required for the supply water to another zone.

The control of air-handling units in a hot water heating system requires careful consideration. Where the piping required is within economical limits, it is advisable to use a three-way mixing valve and a pump in the coil circuit (see Fig. 19, Chapter 28) to achieve control. This arrangement provides a constant flow to the coil while the temperature of the water to the coil is varied by changing the mixture of supply water and return water. If impracticable to use separate three-way valves and pumps for each coil, it is necessary when varying the flow to a hot water coil to use control valves with plugs or discs designed for close control of flow. They should also be sized accurately for the flow required by the coil at a pressure drop as large as the pump and the piping system will permit. Face and bypass dampers to vary the flow of air over a coil with constant water flow can be used in the control of air-handling units.

Precautions should be taken to prevent the freezing of water in a hot water heating system whenever design outdoor temperatures are lower than approximately 20 F, particularly if the system is to be operated on an intermittent basis as in schools where normal operation is not required on Saturdays or Sundays or during vacation periods. In selecting a means of freeze protection, consideration should be given to the type of system as well as outdoor temperature. For instance, an air-handling unit to heat outdoor air must have the coil protected for 32 F, whereas supply mains and radiation usually are not subjected to freezing temperatures until the outdoor temperature drops considerably below 32 F for extended periods. Intake dampers on air-handling units that heat outdoor air should be interlocked with the fan motor so as to close whenever the fan is not running. Provision should be made to close the outdoor-air intake and/or stop fan operation if the water circulating in the coil drops below a safe temperature level. The possibility of water freezing in the mains can be minimized by using an outdoor thermostat to provide continuous circulation of the water whenever the outdoor temperature drops below approximately 30 F. Another arrangement uses an interval timer and outdoor thermostat to provide intermittent cycling of the circulating pump whenever the outdoor temperature drops below some predetermined temperature. A thermostat in the return water at the boiler can be used to start the burner or other heat source if the temperature of the return water drops to about 40 F. See Chapter 43 for general information on controls.

ZONE CONTROL OF STEAM AND WATER SYSTEMS

This section is concerned with zone controls for steam or hot water heating systems. Refer to Part II of this chapter for a general discussion of zoning.

Most zone control systems are designed to supply heat to a zone at a rate equal to the heat loss. This rate of heat input may be established by room thermostats, by controllers responsive to outdoor conditions, or by a combination of both. The outdoor controller responds to temperature and may respond to solar radiation and wind velocity and direction. Features which may be added to the basic zone control system include:

1. Means for automatically or manually varying the heat input to the zone to compensate for variations in internal heat gain. The automatic arrangement requires a thermostat at a representative location within the zone. The manual arrangement usually consists of a switch located where convenient.

2. Means for maintaining a lowered economy temperature or shutting off the heat completely at night or at other times when the zone is unoccupied. This may be accomplished manually or by means of a clock.

3. An arrangement providing rapid warm-up following a period at the nonoccupancy temperature. During this warm-up period full heat may be admitted to the system or the heat input may be controlled at a higher-than-normal rate. The warm-up may be accomplished by a manual switch or may be part of the clock program. A warm-up thermostat located within the zone may be arranged to terminate the warm-up period independently of the switch or clock when the zone approaches the desired occupancy temperature.

4. A manual switch to provide full heat or no heat independently of other controls and to permit manual selection of the various clock functions.

5. Means for shutting off the heat completely when the outdoor temperature reaches a point where heat no longer is needed.

6. A high-limit thermostat in the zone to shut off the heat completely when the zone becomes overheated.

Steam: Zone Distribution

In *continuous-flow steam systems* the quantity of steam supplied to the zone is varied in accordance with the demands of the control system. To obtain equalized distribution of the steam, metering orifices are generally required on the inlets to all radiators or convectors. Supplementary controls for low heat output are often desirable. The following two arrangements are common:

1. Varying the difference in pressure between the supply and return which results in partially filling the heating units in accordance with demand. This method can be used on atmospheric or vacuum return systems.

2. Varying the pressure in the system while maintaining a constant differential pressure between the supply and return. This results in the heating units being filled with steam at a temperature which is proportional to demand. A vacuum pump maintaining a vacuum slightly below the supply pressure is required, and vacuums as high as 20 in. Hg during mild heating weather are common.

In *intermittent systems* steam at full pressure is supplied intermittently to the zone, the length of the *on* and *off* periods being varied in accordance with the demands of the control system. Two common arrangements are:

1. A timing device controlled by an outdoor thermostatic element. This can be used to vary the on-off periods of boiler operation or the opening and closing of a steam valve as a function of outdoor temperature.

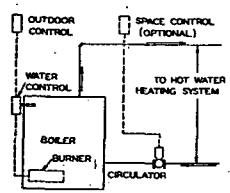
2. A control responding to an outdoor element and an element attached to a radiator or convector. This control varies the length and frequency of the on-off intervals in such a way that the radiator or convector temperature is varied in accordance with outdoor temperature.

For proper operation of zone control equipment the heating system must be carefully designed and installed, and maintained in good operating condition. Vents, traps, vacuum pumps, and valves must be given a careful inspection and repaired or replaced when required. Piping must be of adequate size, graded, and dripped. Return piping must be vented and any pockets or lifts removed.

Water: Zone Distribution

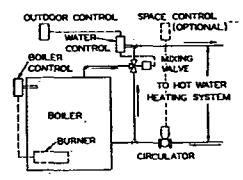
A common method of obtaining zone control of hot water systems consists of varying the temperature of water supplied to the zone inversely as the outdoor temperature varies. This may be accomplished by a submaster thermostat in the zone supply line readjusted by an outdoor master thermostat, or by a single controller having one

measuring element in the supply line and another compensating element outdoors. On single-zone installations the controller may actuate the burner directly as shown in Fig. 25. If the boiler is also used for domestic hot water the controls may be arranged as in Fig. 26. In this case the boiler is controlled at a fixed temperature and the zone controller is arranged to operate a valve to proportion hot water from the boiler with cooler return water to obtain the desired zone water temperature. On multiple-zone installations the boiler usually is controlled at a fixed temperature, and each zone is controlled by a separate valve and outdoor compensated controller. Each zone should be equipped with a separate pump because in many cases the



(Limit and primary controls not shown)

Fig. 25 .... Control of Hot Water Heating Without Domestic Hot Water Service



(Limit and primary controls not shown)

Fig. 26 .... Control of Hot Water Heating With Provisions for Domestic Water Heating

return water from some zones is warmer than the required supply water for other zones. By using individual zone pumps the returns can be separated and thus prevent overheating that would otherwise occur.

To compensate for variations in internal load, the action of the zone control may be modified by a room thermostat in each zone which either starts and stops the zone pump, modifies the relationship between outdoor temperature and water temperature, or operates an automatic valve in the zone supply or return line. Each zone may be further subdivided by providing a separate thermostat and valve for each subdivision.

Another zone control method for hot water systems consists of maintaining a constant water temperature and varying the volume of water supplied to the zone by means of a room thermostat which either starts and stops the zone pump or operates a two-position valve in the zone supply or return line.

Whenever flow control valves are employed in water circuits it is necessary to employ a relief valve or pressure-regulating valve in a bypass connection around the pump.

TERMINAL EQUIPMENT

Control applications to the following types of terminal equipment are described in their respective chapters:

Radiant Panels—Warm Air Ceiling Panels, Chapter 18
—Hot Water Panels, Chapter 30
—Electric, Chapter 30 and 17
Snow Melting, Chapter 49
Unit Heaters, Chapter 15
Unit Ventilators, Chapter 15
Unit Air Coolers, Chapter 16
Induction Units, Chapter 19
Dual-Duct Mixing Units, Chapter 19
Room Fan-Coil Units, Chapter 19
Reheat Coils for Individual Spaces, Chapter 19

Classroom Heating and Cooling Units

Controls for winter heating and ventilating are essentially the same as for classroom unit ventilators as described in Chapter 15, Unit Ventilators and Unit Heaters, except that face and bypass dampers are usually added. The supply hot water temperature should be varied inversely with outdoor temperature for best results. The space thermostat operates the face and bypass dampers, coil valve, and the outdoor- and return-air dampers.

When the system water is changed from hot to chilled water the controls are reversed. For summer cooling the outdoor-air damper opens to a fixed position to admit the required percentage of outdoor air when the fan motor is started. The space thermostat is now reverse-acting and operates only the face and bypass dampers at the cooling coil while the valve remains open.

Radiators and Convectors

When individual room control is used each radiator and convector is equipped with an automatic control valve. Depending upon the size of the room one thermostat may control one or several valves in unison.

Volume Damper or Single-Duct Units

When individual room control is used each volume damper is equipped with an operator. Often several air outlets are supplied through one damper. The damper operation is usually controlled by a room-type thermostat. The volume damper is usually set to provide for minimum ventilation requirements regardless of thermostat demand. When the air-supply system is used for heating as well as cooling it is necessary to use a heating-cooling thermostat.

RESIDENTIAL HEATING AND AIR CONDITIONING

The control equipment function in the residence may vary from the regulation of a coal-fired heating plant to the completely automatic control of a year-around heating and cooling system. Regardless of the type of heating or air-conditioning system used, the control system should be selected carefully to insure safety and comfort of the occupants as well as economy of operation.

The simplest type of domestic control is one in which the room thermostat starts and stops the burner, as described in the preceding section. It may be used on gravity or forced-warm air, hot water or steam systems. In forced

warm air or forced hot water systems the fan or circulator usually is controlled by a thermostat in the furnace or boiler and runs whenever the air or water temperature is above a minimum value, as discussed in Chapter 26. If straight two-position control is used for this application, the *on* and *off* periods are relatively long and there is a tendency towards cold drafts on the floor during off periods. If timed two-position control is used, the burner may be cycled (started and stopped) often enough to provide results very comparable to proportioning control. The heat input to the home is proportioned continuously to the heat loss from the home. The fan of a forced warm air system or the circulator of a forced hot water system may be run almost continuously, thus minimizing the possibility of cold drafts.

Proportional control is used on residential applications when provisions are made for it in the design of the system. Face and bypass dampers on warm air systems, throttling valves on steam systems and three-way mixing valves on hot water systems are typical examples of controlled devices which permit proportional control. With these arrangements the burner usually is controlled by a thermostat in the furnace or boiler. Proportional control will permit nearly continuous circulation of warm air or hot water, thus reducing cold drafts.

Automatic night set-back through the use of a day-night type room thermostat is a refinement frequently added to residential control systems. Though opinions vary regarding the amount of fuel that can be saved by automatic night set-back, tests made both in the warm air heating research residence and the I = B = R research home at the University of Illinois indicate that, on thermostatically controlled systems, a possible fuel saving of from 7 to 10 percent may be obtained by reducing the house temperature 6 to 10 F from about 10:00 p.m. to 5:30 a.m.

It is often desirable to divide the house into two or more zones for greater accuracy of control and comfort. Each zone may then be maintained individually at the desired temperature level. The division into zones should be based upon exposure or occupancy; the most common division is usually found to be (1) the living section such as living room, dining room, den; (2) the sleeping section; (3) the service section such as kitchen, pantry, servants' quarters; and (4) recreational areas.

Further discussion of zone control principles has been given in Part II of this chapter.

Year-round residential air-conditioning systems which provide for heating in winter and cooling in summer should be given the same consideration in selecting the control system as required for commercial air-conditioning systems. See Chapter 19. The basic principles are the same and the final results must provide for the comfort of the occupants. A reduction in investment for automatic controls may result in wasteful operation of the system and discomfort for the occupants.

Special thermostats have been designed to take care of the year-round residential air-conditioning systems. These thermostats offer manual, semi-automatic, and automatic change-over from heating to cooling as selected by the unit manufacturer. Generally speaking, the heating unit is controlled by timed two-position action and the cooling unit by straight two-position action. Some form of humidity control may be added.

When an *add-on* unit is installed to obtain a year-round air-conditioning system by change of a straight heating system, it may be controlled separately or integrated with the heating control system. However it is accomplished, the control system should be simple enough for the user to understand how to operate it effectively.

REFERENCE

[1] E. J. Brown and J. R. Fellows: Pressure losses and flow characteristics of multiple-leaf dampers (ASHAE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, August 1957, p. 119).

BIBLIOGRAPHY

Automatic Control Terminology (American Society of Mechanical Engineers, 1954).

R. H. MacMillan: *An Introduction to the Theory of Control in Mechanical Engineering* (Cambridge University Press, New York, 1951).

E. W. F. Feller: *Instrument and Control Manual for Operating Engineers* (McGraw-Hill Book Co., New York, 1947).

J. E. Haines: *Automatic Control of Heating and Air Conditioning* (McGraw-Hill Book Co., New York, 1953, 1st ed.).

Save Fuel for Victory (University of Illinois, *Engineering Experiment Station Circular Series* No. 47, p. 31).

CHAPTER 44

INSTRUMENTS AND MEASUREMENTS

Temperature Measurement, Pressure Measurement, Air-Flow Measurement, Air Change Measurement, Measurement of Air Currents, Humidity Measurement, Measurement of Mean Radiant Temperature, Heat Transfer Through Building Materials, Combustion Analysis, Smoke Density Measurements, Sound and Vibration Measurements, Determination of Air Contaminants

HEATING and air-conditioning engineers and technicians require instruments for both laboratory and field use. Somewhat more precision is attainable and essential in the laboratory, where research and development are undertaken, than in the field, where acceptance and adjustment tests are conducted. The purpose of this chapter is to describe briefly some of the more important instruments, and to outline their uses and characteristics.

TEMPERATURE MEASUREMENT[1, 2]

Thermometers

Any device capable of indicating temperature is a thermometer, but in common usage the term *thermometer* without qualification has come to signify the ordinary liquid-in-glass temperature indicating device. Mercury-filled thermometers have a useful range from −40 F, the freezing point of mercury, to about 1000 F, at or near which the glass usually softens. Lower temperatures can be measured with alcohol-filled thermometers for which the range is about −94 F to +248 F. The better thermometers have their scales, either Fahrenheit or Centigrade, etched with acid into the glass which forms their stems.

Thermometers are calibrated during manufacture for at least two temperatures, frequently the freezing and boiling points of water, and the space between the calibration points is then divided into the desired scale divisions. The probable error for etched stem thermometers is plus or minus one scale division. For precise work, recalibration is frequently necessary to determine the proper corrections to be applied to the scale readings.

Liquid-in-glass thermometers are calibrated by the manufacturer for either complete or partial stem immersion. If a thermometer which has been calibrated for complete immersion is used at partial immersion, that is, with a portion of the liquid column at a temperature different from that of the bath, an *emergent stem correction* should be made. This correction may be calculated by Equation 1:

$$\text{Stem correction} = Kn\,(t_b - t_s) \qquad (1)$$

where

K = the differential expansion coefficient of the mercury or other liquid in the glass. K may be taken as 0.00016 for Centigrade mercurial thermometers and 0.00009 for Fahrenheit mercurial thermometers. For values of K for other liquids and for specific glasses, see Reference 1.

n = the number of degrees of the liquid column emergent from the bath.

t_b = the temperature of the bath.

t_s = the average temperature of the emergent liquid column of n degrees.

A thermometer can be appreciably affected by radiation from surrounding surfaces. If the temperature to be measured is approximately the same as that of the surrounding surfaces, radiation effects may usually be ignored. If the temperature being measured differs considerably from that of the surroundings, steps should be taken to minimize the effect of radiation.[3] This is usually accomplished either by shielding or by aspiration. Shielding is accomplished by placing highly reflective surfaces between the thermometer bulb and the surrounding surfaces. However, this must be done in such a way that air movement around the bulb is not appreciably restricted.[4] Improper shielding may increase rather than decrease errors. Aspiration is accomplished by drawing a high-velocity stream of air or gas over the bulb of the thermometer.

Industrial-type thermometers are available for permanent installation in pipes or ducts. These instruments are fitted with metal guards to prevent breakage, and are useful for many purposes. However, the considerable heat capacity and conductance of the guards or shields may introduce errors.

To prevent errors in temperature measurements, ample time should be allowed for the thermometer to attain temperature equilibrium with the surrounding fluid. In reading a thermometer, the eye should be at the same level as the top of the liquid column to avoid parallax.

Thermocouples

When two wires made of dissimilar metals are joined by soldering, welding, or merely by twisting, a thermocouple or thermo-junction is formed. An electromotive force, which depends upon the temperature of the junction, is found to exist between the wires. When the wires are joined at two points a thermocouple circuit is formed. If one junction is kept at a temperature different from the other, an electric current flows through the circuit due to the difference in emf developed by the two junctions. This phenomenon is employed for temperature measurements in thermocouple systems, one junction being ordinarily kept at a constant temperature, as in an ice bath, while the other junction is placed at a point at which it is desired to observe the temperature. In practice it is desirable to utilize emf to indicate temperature because, at small or zero current flow, the resistance of the circuit is unimportant. A high resistance millivolt meter is useful in some cases but the potentiometer yields better results. In the potentiometer the electromotive force generated by the thermo-

couples is balanced against an electromotive force from the battery so that observations are made with no flow of current through the thermocouple circuit. A conventional arrangement is illustrated in Fig. 1. The thermocouple leads *A-B* are so connected that their polarity opposes that of battery *C*. *If the position of E on the graduated slide wire rheostat DF* is adjusted until galvanometer *G* shows no current flowing, resistance *DE* will indicate directly the voltage generated by the thermocouple. In order to calibrate the instrument, switch *S* is thrown over to the standard cell circuit while rheostat *R* is adjusted so that the galvanometer shows zero current. Battery *C* then exerts the known voltage of the standard cell at *DH*.

The act of adjusting rheostat *D-F* (Fig. 1) for zero current flow is known as balancing the potentiometer. Automatic self-

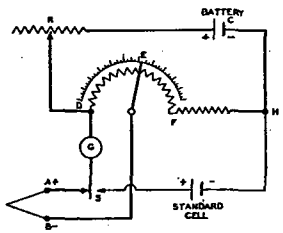


Fig. 1.... Basic Circuit and Connections for Thermocouple and Potentiometer

balancing instruments of both the indicating and recording types are on the market. They usually contain an automatically compensating cold junction to avoid the use of an ice bath, and special thermocouple wire is furnished with them from the factory.

The choice of materials for thermocouple wire is determined by the temperature to be measured, the protection from corrosion afforded the couple, and the precision and service life required. In general, copper vs. constantan is suitable for temperatures up to 700 F, iron vs. constantan up to 1500 F, and chromel vs. alumel up to 2200 F. Higher temperatures require the use of noble metal thermocouples (platinum vs. platinum-rhodium). In addition to the higher initial cost of a noble metal couple, it does not develop as high an emf as the base metal couples do. Impurities make large differences in the performance of thermocouple wires and for this reason calibration of samples from each spool of wire is essential for precise work. Data on wire can usually be obtained from the manufacturer.

With a suitable potentiometer, small wires serve as well for thermocouples as large ones, and the fineness of the wires is limited only by consideration of mechanical strength and convenience in handling. Small couples respond more promptly to changes in temperature and are less affected by radiation than large ones. Heavy gage couples, however, are necessary for high temperature work where corrosion of the wire is a problem. For use in heated air or gases, thermocouples are often shielded,[8] as are thermometers, and aspirated thermocouples are sometimes used. An arrangement has been described for avoiding error due to radiation. It involves the use of several thermocouples of different wire sizes, the true temperature being estimated by extrapolation of readings to zero diameter.[8]

By the use of thermocouples, temperatures at remote points may be indicated or recorded on conveniently located instruments, and temperatures may be obtained within thin materials, narrow spaces, or otherwise inaccessible locations.

Thermocouples in series, with every alternate junction maintained at a common temperature, will produce an emf which, when divided by the number of couples, gives the average emf corresponding to the true average temperature. This series arrangement of thermocouples, often called a thermopile, can have extreme sensitivity and is useful in detecting very small changes in temperature.

Thermocouples in parallel, having the similar metals of a number of couples connected together and run to a common cold junction, will cause an indication on a potentiometer which is the true average emf only if the electrical resistances of the parallel circuits are the same.[8, 9]

The thermocouple is particularly useful in determining a surface temperature. It may be attached to a metal surface in any of several ways. For permanent installations, soldering, brazing, or peening may be desirable. A small hole is drilled for the peening operation; the thermocouple is inserted and the metal is peened to retain it. The fact that the thermocouple is in electric contact with the surface is unimportant in usual circuits. For temporary arrangements, couples may be attached by means of surgical or cellophane tape. For boiler or furnace surfaces, furnace cement serves very well. To minimize the possibility of error due to heat conduction along the wires, a surface thermocouple should be made of fine wires, and the wires should be held in close contact with the surface for an inch or so from the junction. Electrical insulation between the wires must be perfect except at the junction.

Resistance Thermometers

Resistance thermometers depend for their operation upon the change of electric resistance of metal with change in temperature. The resistance generally increases with rising temperature. Their use largely parallels that of thermocouples, although readings tend to be unstable above 950 F. Two-lead temperature elements are not recommended, since they do not permit correction for lead resistance. Three leads to each resistor are necessary to obtain consistent readings.

A typical circuit used by several manufacturers is shown in Fig. 2. In this design a differential galvanometer is used, in

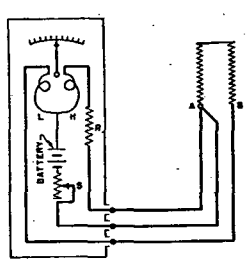


Fig. 2.... Typical Resistance Thermometer Circuit and Connections

which coils *L* and *H* exert opposing forces on the indicating needle. Coil *L* is in series with the thermometer resistance *AB*, and coil *H* is in series with the constant resistance *R*. As the temperature falls, the resistance of *AB* decreases allowing more current to flow through coil *L* than through coil *H*. This causes an increase in the force exerted by coil *L*, pulling the needle down to a lower reading. Likewise, as the temperature rises the resistance of *AB* increases, causing less current to flow through coil *L* than through coil *H*. This forces the indicating needle to a higher reading. Rheostat *S* must be adjusted occasionally to maintain a constant flow of current.

As compared to the thermocouple, the resistance thermometer does not require a cold junction, and it can be simply scaled for more accurate measurements; but, generally because of its construction it is more costly and is apt to have considerable lag. It gives best results when used to measure steady or slowly changing temperature. For accurate results the entire thermometer coil must be exposed to the temperature to be measured.

Pyrometers

The pyrometer is the usual instrument for measuring high temperatures such as those of incandescent bodies or furnace interiors. There are two types. In the radiation pyrometer the radiant energy from an observed surface falls on a thermopile, and the emf generated by the pile, measured by a galvanometer or potentiometer, is an index of the surface temperature. With the optical pyrometer a narrow spectral band, usually red, emitted by the surface, is matched visually with the filament of a special electric lamp. The emf necessary to cause the filament to match the surface in brightness is the index of the temperature of the surface. Pyrometers are calibrated by means of various metals with known melting or freezing points. Portable as well as laboratory models are manufactured.

Color Indicating Crayons

Crayons are available, the marks of which change color or melt at specified temperatures. Such crayons have been sold in boxes covering the range from about 100 F to about 800 F in 100 deg steps, with a precision of some 10 deg. They are a rough but convenient means of determining temperatures, and of locating isothermal lines on surfaces below red heat.

PRESSURE MEASUREMENT

Pressure Gages

The Bourdon is the most common type of pressure gage, and its appearance probably is familiar to anyone having an acquaintance with power plants or laboratories. The essential element of such a gage is the Bourdon tube, a metal tube of oval cross-section curved along its length to *form an almost complete circle. One end is closed* and the other is connected to the vessel in which the pressure is to be measured. With an increase of pressure, the tube tends to straighten, and vice versa. The resulting motion of the closed end is communicated by suitable linkages to a needle moving over a graduated dial. If the range is above about 20 psi, such gages are usually calibrated by means of a *dead weight tester*, whereby known pressures are produced in a fluid by imposing known weights on a *piston of known area. Suction gages and pressure gages* with ranges below about 20 psi are ordinarily calibrated against mercury manometers. Gages are commonly set to read accurately at or near the pressure of probable use. Gages of several different types or qualities are available on the market.[8]

Manometers

The manometer is a simple and useful means for measuring partial vacuum and low pressure. It is, moreover, a primary instrument; it does not require calibration, and it is often used as a standard for the calibration of other instruments. It is so universally used that both the inch of water and the inch of mercury have become accepted units of pressure measurement. In its simplest form, the manometer consists of a U-shaped glass tube partially filled with a liquid. A difference in height of the two fluid columns denotes a difference in pressure in the two legs, which is proportional to the difference in height.

For converting manometer readings into other pressure units, certain proposed standard factors are applicable for precise work. These are based on a standard gravitational acceleration of 32.1740 ft per (sec) (sec) and are as follows:

1 Standard Atmosphere = 14.696 lb per sq in. = 29.921 in. mercury at 32 F = 33.96 ft water column at 68 F

For most ordinary purposes, the following figures are of ample accuracy:

1 Atmosphere = 14.7 lb per sq in. = 29.9 in. mercury = 34.0 ft (408 in.) water column

Manometer tubes should be chemically clean. The bore is not important, except insofar as it affects the meniscus through wetting or surface tension. Bores of at least 3/16 in. for rough, and ½ in. for more precise, measurements are recommended. Liquids other than water are sometimes used for low-pressure measurement and, when this is done, the readings must be corrected for the density of the fluid.

For measuring pressure differences of a few inches of water, or less, U-gages are often set at an angle for scale amplification. In many gages of this type, commonly termed draft gages or inclined manometers, only one tube of small bore is used and the other leg is replaced by a reservoir. The scale is calibrated to read in inches of water, and it is necessary to use a fluid having the same gravity as that for which the gage was originally calibrated, or to apply a correction if another fluid is used. Such gages may be checked one against another. For more accurate calibration the gage may be checked against a micromanometer or a calibrating device known as a hook gage.[10] The accuracy of a draft gage is dependent on the slope of a tube, and consequently the base of the gage must be leveled carefully. It is not desirable to use a slope of less than 1 in 10. Where pressures are read under extreme conditions of temperature, and calibration is possible only at normal temperature, it is necessary to correct for the change in density of the liquid in the manometer.[11] For measuring low-pressure differences to within 0.001 in. of water, very sensitive micromanometers are available, such as the Illinois or Wahlen,[12] *and the Emswiler.*[13] *Various instrument manufacturers* are also prepared to furnish accurate micromanometers.

Barometer

The simplest and earliest type of barometer consists of a glass tube somewhat more than 30 in. long filled with

mercury, and inverted in a cup partially filled with mercury. The height of the mercury column in the tube above the mercury surface in the cup is a measure of the existing atmospheric pressure, except for the slight pressure of the mercury vapor in the space above the mercury in the tube. This can ordinarily be ignored.

Elaborate mercury barometers, fitted with vernier scales, are available. For precise work, corrections must be made for the thermal expansion of the mercury and of the scales.[14] The instruments are usually calibrated for 32 F mercury and 62 F scale temperature, and the correction C to be subtracted from the observed barometer's height is obtained by means of Equation 2.

$$C = \frac{h(t - 28.630)}{(1.1123t - 10978)} \qquad (2)$$

where

C = correction to be subtracted, inches of mercury.
h = observed height, inches of mercury.
t = observed temperature of the barometer, Fahrenheit degrees.

Standard atmospheric pressure at sea level is 29.921 in. Hg, and since normal atmospheric pressure decreases about 0.01 in. Hg for each 10 ft increase in elevation, it is important to make a correction if the elevation of the barometer is not that of the test apparatus. In many cases the barometric reading may be obtained from a nearby Weather Bureau Station, in which case inquiry should be made as to whether the value is for *station* or *sea level* pressure.

Atmospheric pressure may also be measured by an *aneroid barometer* which is easily portable. In this type, variations in atmospheric pressure deflect the thin surface of a sealed diaphragm capsule. Most commercially available aneroid barometers are not as accurate as the mercurial type, and the best require occasional recalibration. Open-scale aneroid barometers are more expensive than common mercurial barometers.

Most of the pressure gages used in engineering work indicate gage pressures, that is, the difference between the pressure being measured and the atmospheric pressure. Such pressures are called gage pressures. Absolute pressure may be obtained by adding barometric pressure and gage pressure algebraically.

AIR-FLOW MEASUREMENT

The theory of various means for measuring the flow of fluids is discussed in Chapter 4 Fluid Flow. Heating and air-conditioning engineers are called upon to measure the flow of air more often than that of other gases, and usually the air is measured at or near atmospheric pressure. Under this condition, the air can be treated substantially as an incompressible fluid which implies that simplified formulas can be used with sufficient accuracy for the solution of many problems.[15]

The Pitot Tube

The Pitot tube, used in conjunction with a suitable manometer, provides a simple method of determining the air velocity in a duct. The construction of a Standard Pitot Tube,[11] and the method of connecting it to a draft gage is shown in Fig. 3. The equation for determining the air velocity from the measured velocity pressure is as follows:

$$V = 1096.5 \sqrt{\frac{h_w}{\rho}} \qquad (3)$$

where

V = velocity, feet per minute.
h_w = velocity pressure (Pitot tube manometer reading), inches of water.
ρ = density of air, pounds per cubic foot.

Since the velocity in a duct is seldom uniform across any section, and since a Pitot tube reading indicates a velocity at only one location, a *traverse* is usually made to determine the average velocity so that the flow can be computed. In general, the velocity is lowest near the edges

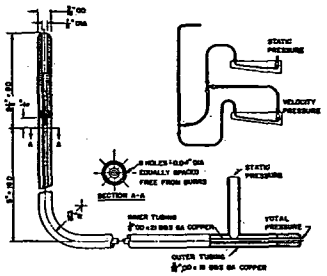


Fig. 3 .... Standard Pitot Tube

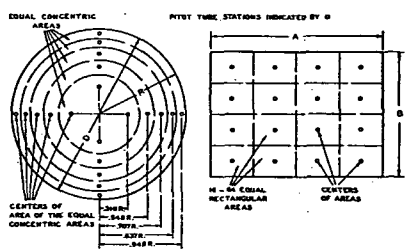


Fig. 4 .... Pitot Tube Traverse for Round and Rectangular Ducts

or corners, and greatest at or near the center. Suggested Pitot tube locations for traversing round and rectangular ducts are shown in Fig. 4. In round ducts not less than 20 readings should be taken along two diameters at centers of equal areas as shown. In rectangular ducts the readings should be taken in the center of equal areas over the cross-section of the duct. The number of spaces should not be less than 16, and need not be more than 64. When less than 64 are taken, the number of equal spaces should be such

that the centers of the areas are not more than 6 in. apart. In determining the average velocity in the duct from the readings given, the calculated individual velocities or the square roots of the velocity heads must be averaged. It is incorrect to use the average velocity head for this purpose.

Pulsating or disturbed flow will give erroneous results and therefore, if possible, the Pitot tube should be located at least 7½ diameters downstream from a disturbance such as that caused by a turn. Straightening vanes[11] located 1½ duct diameters ahead of the Pitot tube will serve to improve the precision of the measurements.

The type of manometer to be used with a Pitot tube depends upon the magnitude of the velocity pressure being measured and the accuracy desired. At velocities greater than 1500 feet per minute, a draft gage of appropriate range is usually satisfactory. If the Pitot tube is being used to measure low air velocities, a precision manometer of some type is essential.

Many forms of Pitot tubes, other than the one described, have been used and calibrated.[16] A double-ended tube,[17] one end pointing downstream, and one upstream, is sometimes used for low velocities, but it should be carefully calibrated for accurate results. A special form of this tube design consists of two straight ⅛-in. tubes soldered together, closed at the end, and with a 0.04 in. hole in each tube opposite the line of contact. This tube is useful in exploring velocities in exhaust inlets, such as hoods placed around grinding wheels. To meet special conditions, different sized Pitot tubes which are geometrically similar to the standard tube can be used.

Orifices and Nozzles

Application of the Pitot tube is often inconvenient when velocities are low, because the resultant velocity pressures become so small that extraordinary means are necessary for measuring them. In addition, velocity surveys of the whole cross-sectional area of a duct are inexpedient when numerous test runs are in prospect. Chiefly for these reasons orifices are used in many test and research applications.

Plate or square-edge orifices[18, 19] are simple to construct and convenient to use. They may be mounted between flanges, and a set of orifices of different sizes may be made readily interchangeable to measure a wide range of air-flow rates. The proportions of standard orifices and nozzles are given in Chapter 4.

Several standard arrangements of pressure taps for use with orifices are described in Chapter 4. Where velocities are low or where flow is free of large eddies and parallel to the walls of the duct, a drilled hole cleared of burrs and at a right angle to the stream is satisfactory.[20] For higher velocities it is common practice to provide four or more holes around the periphery of the duct and connect them into a manifold.

To determine air flow by means of an orifice or nozzle, the pressure drop across the device is measured. The flow rate may then be calculated by means of the equation,

$$Q = 1096.5\, K A_2 \sqrt{\frac{h}{\rho}} \qquad (4)$$

where

Q = air flow, cubic feet per minute.
A_2 = area of the orifice or nozzle, square feet.
h = pressure drop across orifice or nozzle, inches of water.
ρ = density of air, pounds per cubic foot.
K = orifice or nozzle coefficient.

The discharge coefficient K for an orifice is approximately 0.60, and for a nozzle is approximately 0.95. Correct values of K for various diameter ratios, pressure tap locations, and Reynolds numbers are given in Chapter 4. In some instances nozzles are used in multiple so that the capacity of the testing equipment can be changed by shutting off the flow through one or more nozzles. An apparatus designed for testing the air flow and capacity of air-conditioning equipment is described by Wile[21] in an article in which pertinent information on nozzle discharge coefficients, Reynolds numbers, and the resistance of perforated plates is also presented. Such apparatus in some laboratories is commonly referred to as a *code tester*.

The Venturi meter is like the nozzle, except for the addition of a downstream transition section that reduces the pressure drop through the measuring apparatus.

In some cases air flow through a heater coil or heating unit may be estimated by computation from the heat given up by the coil, and the temperature rise (measured by thermocouples) of the air passing through. It is essential to have a uniform flow over the entire inlet and outlet of the heater at the planes of temperature measurement, and temperatures must be taken at enough points to provide a good average.

Propeller or Revolving Vane Anemometer

The propeller or revolving vane anemometer consists of a light revolving wind-driven wheel connected through a gear train to a set of recording dials that read the linear feet of air passing in a measured length of time. It is made in various sizes, 3 in., 4 in., and 6 in. being most common. Each instrument requires individual calibration. At low velocities the friction drag of the mechanism is considerable. In order to compensate for this, a gear train that overspeeds is commonly used. For this reason the correction is often additive at the lower range and subtractive at the upper range, with the least correction in the middle range of velocities. Most of these are not sensitive enough for use below 200 fpm.

Deflecting Vane Anemometer

The deflecting vane anemometer consists of a pivoted vane enclosed in a case. Air exerts a pressure on the vane as it passes through the instrument from an upstream to a downstream opening. The movement of the vane is resisted by a hair spring and a damping magnet. The instrument gives instantaneous readings of directional velocities on an indicating scale. With fluctuating velocities, it is necessary to average visually the swings of the needle to obtain average velocities. This instrument is very useful for studying motion of the air in a room,[22] and in locating objectionable drafts. Various attachments are available, such as the double tube arrangement for determining velocities in ducts, and a device for measuring static pressures. Each instrument, and the attachments for it, must receive individual calibration. For determining average velocity in a duct, it is necessary to *traverse* the duct as is done when using the Pitot tube.

Thermal Anemometers

If a suitable sensing element is heated electrically at a fixed rate and exposed to an air stream, the temperature

difference between the element and the stream becomes a measure of velocity by calibration. In the *hot-wire anemometer*, a very fine heated wire is employed as a resistance-thermometer element whose temperature may be determined accurately.[23, 24, 25] In the *heated-bulb thermometer* type, a heating wire is wound around the bulb of a mercury-in-glass thermometer, and the temperature difference between this thermometer and a similar unheated one serves as an index of air speed.[26] The *heated-thermocouple anemometer* is calibrated to give velocity in terms of the differential *emf* between heated and unheated thermo-junctions exposed to an air stream.[25, 27] Combined measurements of air temperature and velocity are particularly useful for air distribution studies, and automatic recording potentiometers or resistance-thermometer devices facilitate spatial traverses. A correct calibration of thermal anemometers requires consideration of the effects of temperature, humidity, and pressure upon the air properties which influence convective heat transfer.[27] With sensing elements of simple shapes for which convection data are known, thermal anemometers may be designed, both thermally and electrically, for desired characteristics. Directional sensitivity is controllable. Thermal anemometers are convenient and practical for low velocities.

Cup Anemometer

The cup-type anemometer is almost universally used for measuring wind speeds. It consists of three or four hemispherical cups mounted radially from a vertical shaft. Wind from any point of the compass will cause the cups and shaft to rotate. The instrument is usually so constructed that wind speeds may be recorded or indicated electrically at some remote point.

Measurement of Velocities at Inlets and Outlets of Ducts

In the field it is often desirable to make volume measurements at the face of the supply openings. For accuracy, the instrument and its application should be checked on a similar approach and grille in the laboratory before use in the field.

Tests have shown that the propeller-type anemometer can be used successfully on most of the common types of supply grilles.[28, 29] The core area is divided into equal squares, and the anemometer is held against the face of the grille for the same length of time in each. To obtain the air volume in cubic feet per minute, the average corrected velocity in feet per minute thus obtained is multiplied by the average of the gross and net free area of the grille (core) in square feet.

On exhaust openings, the anemometer traverse is made as described previously. The air volume may be determined by multiplying the corrected velocity in feet per minute by the gross core area of the grille in square feet, and by a coefficient for average conditions of 0.85.[29]

When a propeller-type anemometer is held in a stream of varying velocities, it tends to indicate higher than the true average, that is, the speed of the propeller is nearer to the top velocity in its area than it is to the minimum velocity. This is the main reason for the large difference in ratings of unit ventilators by the anemometer method, and by air volume measurements in a duct approach to the inlet.[21]

Deflecting vane anemometers can be also used within their range at the face of supply grilles when properly applied. In principle, it is a case of finding the velocity at many points, and using the average thus found with the correct discharge area at that cross-section. The deflecting vane anemometer equipped with a jet on the end of a rubber tube has been found especially convenient and accurate on supply grilles.[21] On modern air conditioning grilles, the core area is used without a correction coefficient when the jet is held one inch away from the face of the grille. At this distance, the constriction due to the thin bars has disappeared, since the small air jets have reunited and the air stream has not yet spread beyond the core dimensions. With deflecting grilles the exploring jet should be turned to the angle giving a maximum reading. With suitable traversing tips and calibration, this instrument may also be used on exhaust grilles if proper grille factors are applied.[29] Those contemplating such measurements should consult the references cited.

MEASUREMENT OF AIR CURRENTS

The measurement of air current velocities within an enclosed space is usually a tedious undertaking. However, useful data can be obtained with any of several instruments now available, if they are maintained in calibration, and the user understands their operation and limitations.[30] Thermal-type anemometers, when calibrated for low air velocities, are well suited for this service.

The Kata Thermometer

Another instrument useful for studying air currents in free spaces is the Kata thermometer. This is essentially an alcohol thermometer with a very large bulb, and with only two scale divisions etched on its stem. On the *Low Kata* these divisions are 95 and 100 F; on the *High Kata* they are 125 and 130 F. In use, the instrument is heated, usually in a water bath, until the alcohol has risen sufficiently above the upper calibration on the stem. It is then thoroughly dried, and placed at the point at which the air velocity is to be measured. The time in seconds required for the temperature to drop from the upper to the lower scale division is observed. From this cooling time and the calibration factor inscribed on each Kata thermometer, the air velocity can be calculated or determined from a chart. The Kata thermometer is also used in determining the cooling power of the atmosphere, since it loses heat by radiation and convection when dry, and by radiation, convection, and evaporation when the bulb is equipped with a wetted cloth covering.[31] To minimize the effect of radiation, the Kata thermometer is also made with a silvered bulb.

Smoke is a qualitative tool which is very useful in studying air movements. Satisfactory smoke can be obtained from titanium tetrachloride (which, however, is very irritating to nasal membranes) or by mixing potassium chlorate and powdered sugar (a nonirritating smoke) and firing the mixture with a match. This latter process evolves considerable heat, and it should be confined in a pan away from flammable materials. The titanium tetrachloride smoke lends itself to spot determinations, particularly for leakage through casings and ducts, as it can be easily handled in a small pistol-like ejector. The fumes of aqua ammonia and of sulfuric acid, if permitted to mix, form a white precipitate which is useful for some purposes. Two bottles, one containing ammonia water and the other acid, are connected to a common nozzle by means of rubber tubing. Air is forced over the surfaces of the liquids in the bottles by means of a syringe, and the two streams, upon mixing at the nozzle, form a white cloud.

A satisfactory test smoke may also be made by bubbling an air stream through ammonium hydroxide and then through muriatic acid.[32] Smoke tubes, smoke candles, and smoke bombs are available for studying airflow patterns.

Infiltration or Air Change Measurement

The total air leakage into an existing building caused by wind and temperature difference forces can be determined with considerable accuracy by determining the decrease in concentration of a tracer gas. Hydrogen and helium have both been used rather successfully as tracer gases in conjunction with a sensitive thermal conductivity comparator.[33, 34] About one percent of tracer gas by volume is introduced into the room or building, and the tracer gas is thoroughly mixed with the air. The decrease in concentration of the tracer gas is observed at regular time intervals with a thermal conductivity meter as infiltration dilutes the mixture. The following formula is used to calculate the infiltration from the observed concentration measurements:

$$C = C_o e^{-\frac{kt}{v}} \tag{5}$$

where

C_o = initial tracer gas concentration, percent.
C = concentration of tracer gas after t minutes, percent.
v = volume of room or building, cubic feet.
k = infiltration rate, cubic feet per minute.
e = 2.718.

Closets and cupboards should be left open during such infiltration tests so the entire space will have the same rate of change of tracer-gas concentration.

HUMIDITY MEASUREMENT

Psychrometers

Any instrument capable of measuring the humidity or hygrometric state of the air is a hygrometer. A psychrometer is a particular kind of hygrometer which consists of two mercury thermometers, one of which has a cloth wick or *sock* applied to its bulb. For use, the wick is wetted with distilled water and ventilated with air moving at a recommended rate of 900 fpm or more, relative to the instrument.[35] In the simpler and more common type, known as the sling psychrometer, the two thermometers are mounted side by side on a frame fitted with a handle by which the device can be whirled through the air. The motion is arrested for reading the thermometers, and continued until the thermometer readings become steady. Due to evaporation, the wet-bulb thermometer will indicate a lower temperature than the dry-bulb thermometer, and the difference is known as the wet-bulb depression. Charts and tables are available showing the relation between the thermometer readings and the humidity.[35, 36, 41] Data are usually based on a total pressure of one standard atmosphere. For precise work, a correction is necessary for barometric pressure and is usually made by multiplying the observed relative humidity by the ratio of the observed to the standard atmospheric pressure.

For air temperatures below 32 F, the water on the wick may either freeze or supercool, and its state must be known and a proper table or chart used, since the wet-bulb temperature is different for ice and for water. Some operators remove the wick from the wet-bulb for freezing conditions and dip the bulb in water a few times, allowing the water to freeze on the bulb between dips and to form a film of ice. Since wet-bulb depression is slight at low temperatures, precise temperature readings are essential.

In the ventilated or aspirated psychrometer, the thermometers remain stationary, and a small fan or blower or a syringe is used to move the air across the thermometer bulbs. Various designs have been employed in the laboratories, and several commercial models are available.

The Dew-Point Hygrometer

In the usual form of these instruments, means are provided for cooling, and observing the temperature of, a surface which is exposed to air. The temperature at which visible condensation occurs on the surface is considered the dew point of the air. With the dew-point temperature known, the relative humidity and other properties of the air can be determined from tables and charts (See Chapter 3). A bright surface or metallic mirror is usually employed to improve the visibility of the dew deposit, and various means are used to cool the mirror from the back, including evaporation of ether or another refrigerant, or a stream of air passed through dry ice. Dew-point temperatures, in some cases, are observed by means of thermometers in fluids in contact with the back of the mirror, but in modern instruments thermocouples are used, and are soldered or welded to the mirror itself. The dew-point apparatus is not so commonly used as the psychrometer, probably because it is less convenient. It is usable, however, for higher temperatures than the wet- and dry-bulb psychrometer, and should be considered for dew points near or above the boiling point, as in the case of flue gases. Special apparatus for high precision has been constructed, in which the photronic cell and a light source are used for dew or frost detection instead of visual inspection.

Organic Hygrometers

Many organic materials change in dimensions with changes in humidity and this action has been utilized in a number of simple and effective humidity indicators, recorders, and controllers. See also Chapter 43 Automatic Control. Motion caused by the changes in dimension, through a suitable linkage, causes a pointer to move across an indicating dial, moving a pen across a recording chart, or actuating a pneumatic or electric control mechanism.

Common organic materials employed are human hair, animal membrane, animal horn, wood, and paper. Unfortunately, although the field has been well explored, no organic material has been found which can be relied upon to consistently reproduce its action over an extended period of time. Such devices require initial calibration and frequent recalibration or setting, especially if they are exposed to extremes of either high or low humidity. In spite of this they are highly useful because they read directly in terms of humidity, they are simple and inexpensive by comparison with most other types, and the inconvenience of recalibration is often less than that of providing distilled water and servicing the wicks of wet- and dry-bulb psychrometers.

Electrolytic Hygrometers

The dampness, and therefore the electrical resistance of a salt film, varies with the humidity of the atmosphere to

which the film is exposed, and at least two types of instruments based on this fact have been developed. The Dunmore hygrometer was originally designed for use in radiosondes or small balloons, and means were devised whereby the device transmits humidity data back to earth in the form of a radio signal. In the more usual form, this hygrometer consists of a dual winding of small wire on a nonconducting tube. The whole is coated with an electrolytic film, usually containing a salt such as lithium chloride, which forms an electric connection between the windings. Means are provided for determining the electrical resistance of the film, which is an indication of the humidity. In the radio-sonde, variations in the resistance of the film affect the frequency of an oscillating circuit. These hygrometers are usually calibrated by comparison with a wet- and dry-bulb psychrometer. For calibration for some purposes, particularly for use at subzero temperatures, means have been provided for producing atmospheres of known humidity.[42] Advantages of instruments of this type for some scientific and industrial purposes are becoming apparent, and some forms of them are on the market.

The Weaver-type hygrometer is particularly useful for determining the humidity of air or other gas in pipes or closed vessels at various pressures and temperatures. It consists of a threaded plug carrying a central electrode insulated from the plug, except for a gelatinous electrolytic film. The film is exposed to the air or gas from the pipe or vessel, and provision is made for determining its electrical resistance. The hygrometer, or detector, is mounted in a manifold equipped with valves whereby the film can be alternately exposed to the test gas, and to a standard gas having a known absolute humidity. The pressure of the standard gas is varied until contact with it establishes the same resistance in the film as the test gas, and the humidity of the test gas is then determined by computation based on the gas laws.

Chemical Hygrometry

The humidity of an atmosphere can be measured directly by extracting and weighing the water vapor from a known sample. For precise laboratory work, powerful desiccants such as a sulfuric acid and phosphorus pentoxide are used for the extraction process, while for some purposes, calcium chloride, lithium chloride, or silica gel are satisfactory. Freezing the water vapor out of a measured stream of air or gas with solid carbon dioxide, and weighing the resulting ice, is a similar operation. A thermal conductivity method for gas analysis can be used for temperatures above 212 F, or for very low humidities.[43]

MEASUREMENT OF MEAN RADIANT TEMPERATURE

Because if its effect on human comfort, it is sometimes desirable to measure the radiation exchange between a person and his surroundings. The radiant environment of a space is usually expressed in terms of its mean radiant temperature (MRT), which is defined in Chapter 1.

The Vernon globe thermometer[44, 45] is commonly used to measure mean radiant temperature. This instrument consists of a 6-in. diameter hollow copper sphere coated with flat black paint and having a thermocouple or thermometer bulb at its center. The temperature assumed by the globe at equilibrium is the result of a balance between the heat gained or lost by radiation and the loss or gain by convection. In terms of heat-transfer relationships:

$$T_r^4 = T_g^4 + 0.103 \times 10^9 \sqrt{V}\,(t_g - t_a) \qquad (8)$$

where

T_r = mean radiant temperature, Fahrenheit absolute (black-body equivalent).
T_g = globe temperature, Fahrenheit absolute.
V = air velocity, feet per minute.
t_g, t_a = globe ambient air temperature, Fahrenheit.

It will be noted from the above equation that the air temperature and the air velocity around the globe must also be determined. It is the velocity measurement which presents the greatest problem in the use of the globe thermometer.

A *two-sphere radiometer*[46] has recently been developed which may be used for the measurement of MRT. This instrument utilizes two spheres approximately 2 in. in diameter, one of which is gold-plated, and the other black. The two spheres are heated electrically to the same temperature, thus eliminating differences in convection. The difference in energy required by the two spheres to maintain temperature equilibrium is measured, and from this difference, the MRT of the space may be calculated.

Several instruments have been developed to evaluate the combined effect of radiation and convection on human comfort. Among these are the eupatheoscope[47] and the thermal integrator.[48, 49]

HEAT TRANSFER THROUGH BUILDING MATERIALS

Thermal Conductivity

The thermal conductivity (k value) of an insulation, as defined in Chapter 9, is a unit heat-transfer factor. Practically universally accepted for the conductivity determination of flat insulation is the guarded hot plate, which is described in ASTM test method C-177.[50] In its simplest form it consists of an electrically heated plate and two water-cooled plates. Two identical specimens or slabs of a material are required for a test, and one is mounted on each side of the hot plate. A cold plate is then pressed against the outside of each specimen by a clamp screw. The heated plate is divided into two portions: the central or measuring section, and the outer or guard section. During tests the two sections are maintained as nearly as possible at the same temperature. The purpose of the guard section is to minimize errors due to edge effects. The electric energy required to heat the measuring section is carefully observed and converted to Btu per hour. From this heat quantity, the area of the test section, the temperature gradient, and the specimen thickness, the thermal conductivity of the material can be calculated. Hot plate apparatus accommodating specimens on the order of one foot square and an inch or more thick is common. The apparatus at the National Bureau of Standards takes specimens 8-in. square, while plates as large as 3-ft square have been used. The thermal conductivity of cylindrical or pipe insulation (Chapter 27) is determined in a similar manner except that an equivalent thickness must be calculated to account for the cylindrical shape.[51]

Recently there has been considerable development of transient state conductivity apparatus utilizing a slender probe.[52, 53, 54, 55] These instruments are commercially available and have the advantages of rapidity and requiring a small test specimen. While useful as research and development tools, the probe has not been generally accepted as have the guarded hot plate and pipe insulation apparatus.

Wall Conductances

The thermal conductances (C values) of many walls can be satisfactorily estimated from the conductivities of their components and their dimensions, but some walls are complicated by the inclusion of metal, for instance, and tests for conductance are required. The apparatus is required to accommodate large specimens representing actual construction. The guarded hot box apparatus was developed for this purpose. Test specimens for the apparatus at the National Bureau of Standards are 5 ft long and 8 ft high, while others require different sizes, some larger, others smaller.

The guarded hot box is described in the *ASTM Standard Test Code for Built-up Sections* C-236-54T.[56] The apparatus consists essentially of three boxes: a cold box, cooled by a refrigerating machine; a hot box, heated electrically; and a metering box also heated electrically. Each box has an open side to be placed against the specimen. The cold box is clamped against one side of the specimen, and the hot box against the other. The hot box encloses the metering box and is kept at the same temperature to minimize heat exchanges to or from the metering box, except through the specimen. The electric energy necessary to heat the metering box is measured, converted to Btu per hour, and divided by the area and the temperature difference through the wall, from surface to surface, to yield the conductance of the wall. The transmittance or U value of the wall is then computed by means of the surface coefficients from Chapter 9.

A heat-flow meter is sometimes useful for measuring steady heat flow through a wall or other building member.[57, 58, 59] In essence, this meter consists of a plate or slab of material of known thermal resistance having attached thermocouples on both sides. For use, the device is pressed against or cemented to the wall to be tested. At steady state, the temperature difference through the slab, measured with the thermocouples, with the known thermal resistance of the slab, indicates the heat flow through the slab and hence, through the wall covered by it. For best results, such meters are calibrated by means of a guarded hot plate or other suitable apparatus. The chief precaution is to assure that the heat flow is steady at the time of measurement.

COMBUSTION ANALYSIS

There are two approaches to the problem of measuring the capacities of fuel-burning devices, such as boilers and furnaces. The direct or calorimetric test consists in measuring the change in enthalpy or heat content of the fluid, air or water, heated by the device and multiplying by the flow rate in pounds per hour to arrive at the capacity in Btu per hour.[60] The indirect test consists in determining the heat lost in the flue gases and deducting it from the heat evolved by combustion of the fuel.[61] A heat balance consists in the simultaneous application of both tests to the same device. The indirect test almost invariably indicates the greater capacity, and the difference is credited to radiation from the boiler or furnace casing and *unaccounted for loss.*

In the case of some small equipment, the expense of the direct test is not considered justifiable, and the indirect test is relied upon with an arbitrary *radiation and unaccounted for* factor.[62]

Flue-Gas Analysis

The Orsat apparatus is commonly used for analyzing flue gases. In its ordinary form, it consists of three pipettes and a means for isolating a sample of flue gas in a graduate. After measuring, the sample is expelled from the graduate into the first pipette where the carbon dioxide is extracted by potassium hydroxide. The sample is then remeasured and successively passed into the second and third pipettes, where the oxygen and the carbon monoxide are respectively extracted by potassium pyrogallate and cuprous chloride.

For field testing and burner adjustment, simpler portable devices are available for carbon dioxide determination only. From curves, based on typical hydrogen content of several common fuels, efficiencies may be estimated from the carbon dioxide value obtained. More elaborate laboratory equipment is sometimes provided for precise determination of carbon monoxide content by burning the carbon monoxide to carbon dioxide in presence of a catalyst.[63, 64] In large plants, carbon dioxide recorders are used to obtain a continuous indication of the plant's efficiency.[65, 66]

SMOKE DENSITY MEASUREMENTS

Ringelmann charts are widely used for evaluating the density of smoke discharged from chimneys or stacks, and smoke ordinances are based on them in some cities. Each chart is composed of a series of crossed black lines on white paper which, at a distance of about 50 ft, is visually compared with the smoke under observation. Four charts are used with different degrees of blackness as shown in Table 1. The smoke density is specified by Ringelmann numbers from 1 to 4.

Table 1.... Ringelmann Smoke Chart Spacings

Number of Card	Thickness of Lines, mm	Distance in Clear Between Lines, mm
1	1.0	9.0
2	2.3	7.7
3	3.7	6.3
4	5.5	4.5

The photoelectric cell is used in some apparatus developed for smoke density recording in large plants. The same device is included in the testing equipment for domestic oil burners described in *National Bureau of Standards, Commercial Standard* CS75-42.[67] Under Laboratory Tests this publication contains the following section: "Smoke Determination.—After combustion has reached equilibrium, the amount of smoke in the flue gases, when viewed lengthwise through 4 feet of the smoke pipe in accordance with the *Underwriters' Laboratories, Inc., Standard for Domestic Oil Burners* (Subject 296), March 1934 and subsequent revisions, shall not reduce the output of a standard photoelectric cell from 9 microamperes, with a clear smoke pipe, to less than 8 microamperes." The Commercial Standard also requires that during a test after installation, the burner shall operate without visible smoke at the chimney top.

A method of evaluating smoke produced by pot-type oil

burners was developed for the Institute of Cooking and Heating Appliance Manufacturers by R. N. St. John. A glass rod is interposed between a light source and a photosensitive cell, both before and after being exposed to the flue gases from a heating device. The diminution of the light transmitted by the rod, due to the deposit of soot on its surface, causes a reduction in the cell emf which is taken as an index of the concentration of smoke in the flue gases. A description of the method is contained in *National Bureau of Standards Commercial Standard* CS104-46.[68]

DETERMINATION OF AIR CONTAMINANTS

Two measures of air dustiness are in use: particles per unit volume of air, and weight per unit volume of air. A comparative method consists in drawing known samples of air through a known area of filter cloth or paper, and comparing the density of the resulting spots with blackness charts or with spots from other sources.

For counting, particles are captured in a device such as the Smith-Greenburg impinger, the Owens jet dust counter, or in an electrostatic or a thermal precipitation device designed for the purpose.[69] Counting is done with a microscope, and the method yields important results when the nature or constituents of the dust are of interest.

For a weight determination, a known volume of air is drawn through a porous crucible or thimble, and the weight gained by the thimble during the operation is the weight of the dust captured from the air sample.[69] For precise work, the thimble must be dried in a desiccating chamber before weighing each time.

At the National Bureau of Standards a test method was developed for interested Government agencies, under which measured samples of the uncleaned air and of the air cleaned by a device under test are passed through filter papers.[69] The ratio of the flow rates through the two filter papers is adjusted during successive tests, until the resulting dust spots approach equality in density as shown by a photometer. The ratio of the flow rates is then indicative of the effectiveness of the cleaner in arresting dust. A statement of an air cleaner's efficiency by any test method, is meaningless unless the test dust is specified. Other instruments for determining gaseous and particulate air contaminants are described in industrial hygiene literature.[68, 69, 70]

SOUND AND VIBRATION MEASUREMENTS[71, 72, 73, 74, 75]

The measurement, analysis, and ultimate control of noise in air-conditioning systems is becoming increasingly important. The sound-level meter is the basic instrument used in acoustic measurements. While a crystal-type microphone is usually supplied with the meter, dynamic and condenser-type microphones can also be used. The dynamic type is employed when the microphone and sound-level meter are separated by a long cable. The condenser microphone is stable and has a flat response. Crystal microphones are stable and sensitive, but require a temperature correction when connected to the sound-level meter through a long cable.

In the refined art of acoustics, the overall or total sound level is of little value. Loudness and annoyance as interpreted by the ear are a function of frequency as well as sound level. The effective rating and control of noise must, therefore, involve the measurement of frequency distribution. The octave-band analyzer, which divides the noise spectrum into eight frequency bands, is most commonly used in conjunction with the sound-level meter. A number of the criteria that have been developed for environmental noise levels are based on octave bands. Narrow-band filters are also available, and impact noise analyzers can be used to measure transient noise levels such as those produced by drop-forge hammers.

Electronic instruments are available for measuring the frequency, amplitude, and acceleration of a vibrating mass. Vibration meters are usually portable, self-contained, and direct reading.

REFERENCES

[1] For a comprehensive treatment of temperature measurement the reader is referred to *Temperature, Its Measurement and Control in Science and Industry* (symposium of American Institute of Physics, published by Reinhold Publishing Corp., New York).

[2] H. D. Baker, E. A. Ryder, and N. H. Baker: *Temperature Measurement in Engineering* (John Wiley and Sons, New York).

[3] Temperature measurement (*ASME Power Test Code*, Part 3).

[4] G. V. Parmelee and R. G. Huebscher: ASHVE Research Report No. 1292—The shielding of thermocouples from the effects of radiation (ASHVE Transactions, Vol. 52, 1946, p. 183).

[5] P. Nicholls and W. E. Rice: Errors in the measurement of the temperature of flue gases (ASHVE Transactions, Vol. 35, 1929, p. 473).

[6] A. C. Willard, A. P. Kratz, and V. S. Day: *Investigation of Warm Air Furnaces and Heating Systems (Illinois Engineering Experiment Station Bulletin* No. 120).

[7] P. Nicholls: Measuring heat transmission in building structures and a heat transmission meter (ASHVE Transactions, Vol. 30, 1924, p. 94).

[8] W. B. Kirk and G. J. Pacanovsky: Parallel-connected thermocouples for the testing of gas appliances (*Gas*, September 1939, p. 51).

[9] *American Standard for Indicating Pressure and Vacuum Gages, Round Dial Type, with Elastic Pressure Chamber* (American Standards Association, B. 40.1—1939).

[10] *Standard Test Code for Centrifugal and Axial Fans* (ASHVE and NAFM, published as *NAFM Bulletin* No. 110, 1938).

[11] *Fan Engineering* (Buffalo Forge Company, Buffalo, New York, 5th ed., p. 165).

[12] *Illinois Micromanometer* (University of Illinois, *Engineering Experiment Station Bulletin* No. 120, p. 91).

[13] J. E. Emswiler and W. C. Randall: The weathertightness of rolled steel windows (ASHVE Transactions, Vol. 34, 1928, p. 527).

[14] *Psychrometric Tables for Vapor Pressure, Relative Humidity and Temperatures of the Dew-Point* (U. S. Department of Agriculture, Weather Bureau).

[15] For technical data refer to *Fluid Meter Reports* (American Society of Mechanical Engineers, Parts 1—1937, 2—1931, and 3—1933).

[16] *National Advisory Committee for Aeronautics Technical Notes* No. 546 (November 1935).

[17] F. R. Ingram, E. Diez-Canseco, and L. Silverman: The characteristics of double Pitot tubes (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, November 1942, p. 708).

[18] H. S. Bean, E. Buckingham, and P. S. Murphy: Discharge coefficients of square edged orifices for measuring the flow of air (*Bureau of Standards Journal of Research*, Vol. 2, 1929, p. 561).

[19] Flow measurement by nozzles and orifice plates (*ASME Power Test Codes*, 1949, Part 5 of Chapter 4).

[20] Pressure measurement (*ASME Power Test Code*, 1936, Part 2 of Chapter 2).

[21] D. D. Wile: Air flow measurement in the laboratory (*Refrigerating Engineering*, June 1947, p. 515).

[22] G. L. Tuve, G. B. Priester, and D. K. Wright, Jr.: ASHVE Research Report No. 1204—Entrainment and jet-pump action of air streams (ASHVE Transactions, Vol. 48, 1942, p. 241).

[23] A. P. Kratz, A. E. Hershey, and R. B. Engdahl: ASHVE Research Report No. 1165—Development of instruments for the study of air distribution in rooms (ASHVE Transactions, Vol. 46, 1940, p. 351).

[24] D. D. Wile: Development of testing apparatus for thermostats (ASHVE Transactions, Vol. 42, 1936, p. 349).

[25] J. F. Kemp: Stable hot-wire anemometer for low speeds (ASHAE Journal Section, *Heating, Piping and Air Conditioning*, October 1957, p. 137).

[26] C. P. Yaglou: The heated thermometer anemometer (*Journal of Industrial Hygiene and Toxicology*, Vol. 20, October 1938).

[27] H. B. Nottage: A simple heated-thermocouple anemometer (ASHVE Transactions, Vol. 56, 1950, p. 431).

[28] L. E. Davies: ASHVE Research Reports Nos. 857, 911, and 966—Measurement of the flow of air through registers and grilles (ASHVE Transactions, Vol. 36, 1930, p. 201; Vol. 37, 1931; p. 619; and Vol. 39, 1933, p. 373).

[29] G. L. Tuve and D. K. Wright, Jr.: ASHVE Research Report No. 1162—Air flow measurements at intake and discharge openings and grilles (ASHVE Transactions, Vol. 46, 1940, p. 313).

[30] A. M. Greene, Jr. and M. H. Dean: ASHVE Research Report No. 1092—The flow of air through exhaust grilles (ASHVE Transactions, Vol. 44, 1938, p. 387).

[31] G. L. Larson, D. W. Nelson, and R. W. Kubasta: ASHVE Research Report No. 936—Investigation of air outlets in class room ventilation (ASHVE Transactions, Vol. 38, 1932, p. 463).

[32] D. W. Nelson and D. J. Stewart: ASHVE Research Report No. 1076—Air distribution from side wall outlets (ASHVE Transactions, Vol. 44, 1938, p. 77).

[33] G. L. Tuve: Measuring air flow (*Heating, Piping and Air Conditioning*, December 1941).

[34] G. L. Tuve, D. K. Wright, Jr., and L. J. Siegel: ASHVE Research Report No. 1140—The use of air velocity meters (ASHVE Transactions, Vol. 45, 1939, p. 645).

[35] O. W. Armspach and Margaret Ingels: Temperature, humidity and air motion effects in ventilation (ASHVE Transactions, Vol. 28, 1922, p. 103).

[36] H. B. Nottage, J. G. Slaby, and W. P. Gojsza: A smoke-filament technique for experimental research in room air distribution (ASHVE Transactions, Vol. 58, 1952, p. 399).

[37] J. B. Dick: Measurement of ventilation using tracer gas technique (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, May 1950, p. 131).

[38] C. W. Coblents and P. R. Achenbach: ASHAE Research Report No. 1616—Design and performance of a portable infiltration meter (ASHAE Transactions, Vol. 63, 1957, p. 477).

[39] W. H. Carrier: The temperature of evaporation (ASHVE Transactions, Vol. 24, 1918, p. 25).

[40] H. B. Nottage: ASHVE Research Report No. 1401—A proposed psychrometric chart (ASHVE Transactions, Vol. 56, 1950, p. 411).

[41] W. H. Carrier and C. O. Mackey: A review of existing psychrometric data in relation to practical engineering problems (*ASME Transactions*, January 1937, p. 33). Discussion (*ASME Transactions*, August 1937, p. 528).

[42] Arnold Wexler: *Divided Flow Low Temperature Humidity Test Apparatus (National Bureau of Standards Research Paper* No. 1894).

[43] H. A. Daynes: *Gas Analysis by Measurement of Thermal Conductivity* (Cambridge Press, New York, 1933).

[44] H. M. Vernon: The measurement, in relation to human comfort, of the radiation produced by various heating systems (*Institution Heating Ventilating Engineers Proceedings*, Vol. 31, p. 160).

[45] T. Bedford and C. G. Warner: The globe thermometer in studies of heating and ventilating (*Institution Heating Ventilating Engineers Journal*, Vol. 2, 1935, p. 544).

[46] D. J. Sutton and P. E. McNall, Jr.: A two-sphere radiometer (ASHVE Transactions, Vol. 60, 1954, p. 297).

[47] A. C. Willard, A. P. Krats, and M. K. Fahnestock: The application of the eupatheoscope for measuring the performance of direct radiation and convection in terms of equivalent temperature (ASHVE Transactions, Vol. 39, 1933, p. 303).

[48] C. P. Yaglou: Physical procedures in air analysis—Instruments and methods for recording thermal factors affecting human comfort (*American Journal Public Health Supplement*, Vol. 26, March 1936, p. 76). C. P. Yaglou, A. P. Krats, and C.-E. A. Winslow: Instruments and methods for recording thermal factors affecting human comfort, I—Report of subcommittee on physical procedures in air analysis (*American Journal Public Health Supplement*, Vol. 27, March 1937, p. 84).

[49] C.-E. A. Winslow and Leonard Greenburg: The thermo-integrator—a new instrument for the observation of thermal interchanges (ASHVE Transactions, Vol. 41, 1935, p. 149).

[50] *Standard Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate* (adopted July 1942 by ASHVE, *ASTM Designation* C 177-45).

[51] *Tentative Method of Test for Thermal Conductivity of Pipe Insulation (ASTM Designation* C 335-54T).

[52] D. D'Eustachio and R. E. Schreiner: A study of transient heat method for measuring thermal conductivity (ASHVE Transactions, Vol. 58, 1952, p. 331).

[53] F. C. Hooper and F. R. Lepper: Transient heat flow apparatus for the determination of thermal conductivity (ASHVE Transactions, Vol. 56, 1950, p. 309).

[54] F. C. Hooper and S. C. Chang: Development of thermal conductivity probe (ASHVE Transactions, Vol. 59, 1953, p. 463).

[55] C. P. Lents: A transient heat flow method of determining thermal conductivity: Application to insulating materials (*Canadian Journal of Technology*, Vol. 30, June 1952, p. 153).

[56] *Tentative Method of Test for Thermal Conductance and Transmittance of Built-up Sections by Means of Guarded Hot Box* (American Society for Testing Materials, ASTM-C-236-49T).

[57] P. Nicholls: ASHVE Research Report No. 685—Measuring heat transmission in building structures and a heat transmission meter (ASHVE Transactions, Vol. 30, 1924, p. 65).

[58] R. G. Huebscher, L. F. Schutrum, and G. V. Parmelee: A low-inertia low-resistance heat flow meter (ASHVE Transactions, Vol. 58, 1952, p. 275).

[59] J. T. Gier and R. V. Dunkle: Using the heat flow meter to study heat transfer (*Refrigerating Engineering*, Vol. 62, October 1954, p. 63).

[60] *I = B = R Testing and Rating Codes for Low Pressure Heating Boilers* (Institute of Boiler and Radiator Manufacturers, 1947).

[61] *Commercial Standard for Warm Air Furnaces Equipped with Vaporizing Pot-type Oil Burners* (National Bureau of Standards, C.S. 104-26).

[62] Martin Shepherd: Rapid determination of small amounts of carbon monoxide (*Industrial and Engineering Chemistry, Analytical Edition*, 19, 77, 1947).

[63] Martin Shepherd: Determination of small amounts of carbon monoxide in air by various reference methods (*National Bureau of Standards Journal of Research* 38, 351, 1947, R. P. 1777).

[64] *Commercial Standard for Mechanical Draft Oil Burners Designed for Domestic Installations* (National Bureau of Standards, C.S. 75-42).

[65] Philip Drinker and Theodore Hatch: *Industrial Dusts* (McGraw-Hill Book Co., New York).

[66] S. R. Lewis: Testing and rating of air cleaning devices used for general ventilation work (ASHVE TRANSACTIONS, Vol. 39, 1933, p. 277).

[67] R. S. Dill: A test method for air filters (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 379).

[68] M. B. Jacobs: *Analytical Chemistry of Industrial Poisons, Hazards and Solvents* (Interscience Publishers Inc., New York, 1941).

[69] J. J. Bloomfield and J. M. Dalla Valle: *The Determination and Control of Industrial Dust* (*U. S. Public Health Bulletin* No. 217, 1935).

[70] F. A. Patty: Sampling and Analysis of Atmospheric Contaminants (*Industrial Hygiene and Toxicology*, Vol. 1, Interscience Publishers Inc., New York, 1948).

[71] E. E. Gross, Jr.: Noise measuring and sound-control (*Refrigerating Engineering*, Vol. 66, May 1957, p. 49).

[72] A. P. Peterson and L. L. Beranek: *Handbook of Noise Control* (General Radio Co., Cambridge, Massachusetts, 1954).

[73] L. L. Beranek: *Acoustics* (McGraw-Hill Book Co., New York).

[74] H. F. Olson: *Elements of Acoustical Engineering* (D. Van Nostrand Co., New York).

[75] *American Tentative Standards for Sound Level Meters for Measurement of Noise and Other Sounds* (American Standards Association, Z24.3-1944).

CHAPTER 45

MOTORS AND MOTOR CONTROLS

Fundamentals of Motor Selection; Alternating-Current Motors, Types and Control Equipment; Motor Rating; Functions of Motor Control Equipment; Glossary of Motor Terms, Enclosures, Speed Classification and Mounting

THE electric motor, available in many different types suitable for various services, is now the most widely used form of motive power. The equipment for starting, controlling, and protecting these motors varies with the type and with the functions it is desired to attain. Motors are divided into two general classifications, *alternating-current* or *direct-current*, depending on the power source to be used.

FUNDAMENTALS OF MOTOR SELECTION

The following characteristics of the *power supply* should be determined: (1) whether current is alternating or direct, (2) voltage, (3) alternating-current phase, (4) alternating-current frequency, (5) voltage regulation, (6) continuity of power.

1. *A-C vs. D-C Systems*. For most applications, a-c supply is satisfactory since suitable performance can usually be obtained with a-c motors and control. Where special characteristics, such as an extra wide speed range and severe accelerating or reversing duty are involved, conversion by means of motor generator sets, by rectifiers, or in special cases by converters, may be justified.

2. *Voltage*. Standard conditions of voltage and frequency are the values listed on the name plate of the motor. Reasonable horsepower design limits are given in Table 1. Power lines are often given voltage ratings known as *nominal system voltages* which are numerically slightly different from the standardized motor voltages.

3. *Phases*. Three-phase power supply is most desirable, but only single-phase is offered for most residential and rural districts.

4. *Frequency*. Sixty-cycle systems predominate in the United States. In foreign countries, 50-cycle systems are common and nominal system voltages are frequently different.

5. *Voltage Regulation*. The voltage regulation of the power supply should be known in order to select motors which will deliver sufficient torque even with the probable drop in voltage, to start and carry the load. All induction motor torques and synchronous-motor starting and pull-in torques vary as the square of the voltage.

6. *Continuity of Power*. Dips in voltage from switching or other line disturbances may necessitate time-delay undervoltage protection, and, in case of synchronous-motors, high torque designs and resynchronizing control. Sustained low voltage may necessitate higher torque motors.

The following characteristics of the *driven machine* should be determined: (1) mechanical arrangement including position of motor and shaft, portability desired, drive connection, and space limitations; (2) speed range desired; (3) horsepower requirement; (4) torque; (5) inertia; and (6) frequency of starting.

1. *Mechanical Arrangement*. Arrangement of the driven machine usually determines whether a *horizontal* or *vertical* motor is needed. Horizontal motors are more generally available and less expensive; most grease-lubricated ball-bearing motors will operate in either position. Fractional-horsepower waste-packed sleeve-bearing motors are satisfactory for short periods of vertical operation where no thrust is involved.

If shaft is *tilted* for momentary operation, special construction of bearing housings will be required for oil-ring-lubricated sleeve-bearing motors, to avoid loss of lubricant. In case of long periods of tilted operation, bearings suitable for end thrust may be necessary. Ball-bearing motors with grease lubrication are suitable for tilted operation.

Most motors are suitable for mounting with base above a horizontal shaft or to one side of the shaft, provided the end shields are rearranged. If, during operation, the angle of the motor (with regard to the horizontal shaft) changes more than 10 or 12 deg, a ball-bearing motor will usually be required. Sleeve-bearing motors are also applicable within the angle given if modified oil gages are provided.

On portable machines, motors of greater compactness and less weight than standard may be required, and special bearing construction may be needed, except for ball-bearing motors. Direct connection should always be considered where machine speed coincides with available motor speed.

Maintenance, efficiency, power factor, space, and initial cost, will determine the choice between *direct connection* and other methods, such as *belt*, *chain*, or *gear* drive. When direct connection is possible (where parts of the driven machine, such as shaft or bearings, are common with the motor structure) a *built-in* construction may be advantageous.

Belt Drive. Diameters and widths of pulleys or sheaves and center distances are factors in determining motor-bearing pressures and shaft deflection. Flat belts should not run at greater speeds than about 5000 fpm. Application of flat belting to vertical-shaft motors is difficult.

Chain Drive. The chain manufacturer should be consulted so that the best drive on a basis of quietness and economy of operation may be selected.

Gear Drive. Compactness and arrangement of drive often indicate *gear motors*, which are obtainable in a variety of mechanical constructions with speed ratios of 3 to 1 upwards, and are generally limited to about 75 hp maximum. Where the pinion of ordinary spur gearing is mounted on the motor shaft, two-bearing motors are limited in horsepower ratings. Maximum pitch-line speed with steel pinions is about 1400 fpm.

The selection of the motor part of a *gear motor* is the same as for a conventional motor.

Space limitations may affect the choice of motor and require (*a*) built-in construction; (*b*) a gear motor; (*c*) forced

Table 1 . . . . Reasonable Horsepower Design Limits for Standard Motor Voltages

Power Supply	Standard Motor Voltage	Suggested Minimum Horsepower	Suggested Maximum Horsepower
Alternating 1-phase	115	None	1
	230	None	15
Alternating 3-phase	110	None	15
	220	None	200
	440–550	1	1000
	2300	50	6000
	4000	100	7500
	4600	250	8000
	6600	400	None

ventilation using an external blower; or (*d*) a small frame with Class B or H insulation permitting higher temperature rise.

2. *Speed Range.* Where more than one speed or a range of speeds is required, one of the motor types listed in Table 2 may be applicable, depending upon the power supply and the speed range required.

3. *Horsepower Requirement.* The horsepower required by the driven machine determines the motor rating. Where the load varies with time, a horsepower vs. time curve will permit determination of the peak horsepower required. The calculation of the root-mean-square (*rms*) horsepower indicates the proper motor rating from a heating standpoint. In case of extremely large variations in load, or where shutdown, accelerating, or decelerating periods constitute a large portion of the cycle, the rms horsepower may not give a true indication of the equivalent continuous load. The motor manufacturer should, therefore, be consulted.

Where the load is maintained at a constant value for an extended period (varying from 15 min to 2 hr, depending on the size), the horsepower rating required will usually not be less than this constant value, regardless of other parts of the cycle.

If the driven machine is to operate at more than one speed, the horsepower required at each speed must be determined.

4. *Torque.* The torque required to operate the driven machine at every moment between initial breakaway and final shutdown is important in determining the type of motor. A torque-speed curve is desirable and sometimes essential.

The *starting torque* or breakaway torque required by the driven machine may be as low as 10 percent, as in the case of medium-sized centrifugal pumps, or as high as 225 to 250 percent of full-load torque, as in the case of a loaded reciprocating two-cylinder compressor. The breakaway torque may vary greatly at different times because of frequency of start, temperature changes, type and amount of lubricant, etc. The motor torque available at the shaft must be well above the torque required by the driven machine, taking into consideration these variables as well as the possibility of low voltage and the type of starter used.

The torque required after breakaway for *acceleration to full speed* varies with different driven machines, remaining at a rather high value throughout acceleration for such machines as loaded compressors and plunger pumps. The torque delivered by the motor must at all points, up to full speed, be in excess of the torque required by the driven machine. The greater this excess torque, the faster will be the acceleration. The approximate time required for acceleration from rest to full speed is:

$$\text{Time in seconds} = (\text{rpm}) \times WR^2 \div (T \times 308) \qquad (1)$$

where

(rpm) = full-load speed in revolutions per minute.
T = average torque available for acceleration, foot-pound.
WR^2 = inertia of rotating parts, pound-foot square.

If the time to accelerate on full voltage is greater than about 20 sec, special motors or starters may be required to avoid overheating.

5. *Inertia of Driven Machine.* The inertia or flywheel effect WR^2 of the rotating parts of the driven machine affects the accelerating time and, therefore, the heating of motors and control, particularly where reversing duty or frequent starting is involved.

Where synchronous motors are applied, the WR^2 must be known, since the pull-in torque required of this motor varies approximately as the square root of the total WR^2 of motor and load.

The WR^2 of a rotating member of the driven machine which operates at a speed different from that of the motor may be converted to an equivalent value at the motor shaft by multiplying by

$$[(\text{rpm of rotating member}) \div (\text{rpm of motor})]^2 \qquad (2)$$

6. *Frequency of Starting.* The frequency of starting the driven machine affects the motor and control by increasing their heating, particularly where accelerating time is prolonged by high WR^2 and high load torques. In general, driven machines starting more than 4 to 6 times per hour may require special motors and control.

Table 2 . . . . Speed Ranges for Various Types of Motors

Power Supply	Type	Speed Range
Single-Phase a-c	* Brush-shifting repulsion motor	3:1
	* Capacitor motor with tapped winding	2:1
	Multi-speed capacitor motor	2 or 3 fixed speeds
Poly-phase a-c	Multi-speed squirrel-cage	2, 3, or 4 fixed speeds
	* Wound-rotor motor	2:1
	* 2-speed wound-rotor motor	4:1
	Brush-shifting shunt motor	20:1
	* Brush-shifting series motor	3:1
	Squirrel-cage motors with variable frequency supply	Very wide range
	Motor-Generator Set—D-C Drive Motor	Very wide range
	Rectifiers—D-C Drive Motor	Very wide range
d-c	Shunt-wound standard *constant-speed* motor with field control	2:1 in some cases
	* D-C motor with armature control	Wide
	Adjustable-speed motor	From 3:1 to 6:1
	Shunt motor with adjustable-voltage supply	Very wide

* Speed regulation relatively wide. Unsuitable for some loads.

ALTERNATING-CURRENT MOTORS

Alternating-current motors are divided into two main classifications: *polyphase* and *single-phase* (see Table 3), according to the type of power supply. They are further subdivided by type of motor winding.

When polyphase power is available it is usually found more economical to apply polyphase motors in preference to single-phase motors. A typical 5 hp, 1200 rpm capacitor-start induction-run single-phase motor, for instance, will cost approximately twice as much as the corresponding three-phase *Design B* squirrel-cage motor. In addition, the polyphase motor has the advantage of higher efficiency.

Polyphase Motors

The three types of polyphase motors are: *squirrel-cage induction* motors, *wound-rotor induction* motors, and *synchronous* motors.

Squirrel-cage motors are specified by NEMA standards providing a variety of speed and torque characteristics. *Design A* motors provide normal starting torque at starting current in excess of *Design B* motors, and are suitable for constant speed application to equipment such as fans and blowers. *Design B* motors provide normal starting torque with NEMA starting current values shown in Table 4, which are acceptable by many power companies for full voltage starting. They are used for the same type of application as *Design A*. *Design C* motors provide high starting torque with starting current same as *Design B*, and are used on compressors started without unloaders, and on reciprocating pumps. *Design D* motors have high slip* and are used with flywheels for widely pulsating loads on equipment such as reciprocating compressors and pumps where other motors would draw high peak currents. *Design F* motors have low starting current and low torques. They are used on power systems of limited capacity. Careful application must be made since the low breakdown torque provides only limited safety margin for overload or low voltage conditions. A drop in voltage, for instance, could cause the motor to stall.

Figs. 1 and 2 illustrate the characteristics of squirrel-cage motors. The motor operates under load from near synchronous speed at light load to about 95 percent synchronous speed at full load. Overloads decrease the speed further until the maximum torque point is reached. If the load torque exceeds this point the motor will stall. The motor torque varies as the square of the voltage. If the voltage drops 10 percent, the motor torque will be only 81 percent of rated value. Both power factor and efficiency are improved if the motor is operating as near rated load as possible. In addition, as shown in Fig. 2, power factor and efficiency are better for higher speed motors.

Hermetically-enclosed polyphase a-c motors are now quite generally used with large centrifugal compressors. The motors are built into the compressors and are cooled by circulation of refrigerant through the motor windings or by circulation of condenser cooling water through the motor housing. In either case, the cooling is excellent and the motor can deliver more power than if it were air-cooled.

Wound-rotor motors are used for applications requiring high starting torque at low starting current, because a wound-rotor motor with its controller and resistance can develop full load torque when starting with about full load current. For comparison, a squirrel-cage motor would require from 3 to 5 times as much current to develop full load torque at starting. The wound-rotor motor is also used for varying speed service to drive fans, blowers, and other continuous duty apparatus.

The addition of resistance to the secondary winding of the wound-rotor motor changes the speed torque characteristics. The motor speed, with the resistance added, is dependent on load, and consequently, the motor has very poor speed regulation when secondary resistance is added to reduce the speed

*Refer to Glossary at end of chapter.

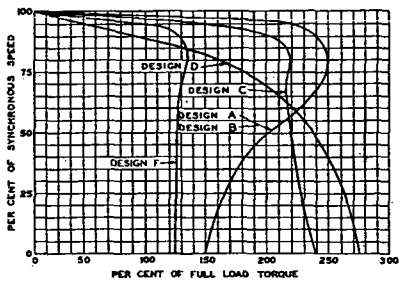


Fig. 1 . . . . Speed Torque Characteristics of Squirrel-Cage Motors

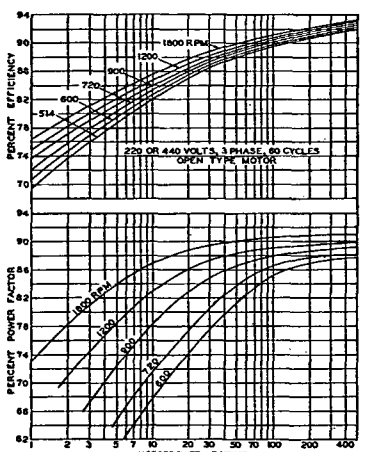


Fig. 2 . . . . Efficiencies and Power Factors for Squirrel-Cage Induction Motors

Table 3 .... Classification of Motors

	Type	Speed Characteristics	Full Voltage		HP Range	Application See Footnotes (a) to (e)
			Starting Torque	Starting Current		
	Constant Speed Drives					
Polyphase a-c	Squirrel-cage general purpose Design A	Constant	Normal 1-2.5 times[f]	High 6-8 times	All	(a) Fans and (c) centrifugal pumps and centrifugal compressors
	Squirrel-cage Design B	Constant	Normal 1-2.5 times[f]	Normal 5-6 times	Medium Small	(a) Fans and centrifugal pumps and centrifugal compressors
	Squirrel-cage Design C	Constant	High 2-2.5 times[f]	Normal 5-6 times	Medium Small	(b) Reciprocating pumps (e) and compressors started loaded
	Squirrel-cage Design F	Constant	Low 1.25	Low 4 times	Medium Large	Fans, centrifugal pumps, and compressors
	Wound rotor	Constant or variable	High 1-2.5 times (with secondary control)	Low 1-3 times	All	(a) Hoists (b) reciprocating pumps and compressors (c) and frequent (e) or hard start
	Synchronous high speed	Exactly constant	Normal 0.75-1.75 times	Normal 5-7 times	Medium Large	(a) Fans and centrifugal pumps and centrifugal compressors
	Synchronous low speed	Exactly constant	Low 0.3-0.4 times	Low 3-4 times	Medium Large	(a) Reciprocating compressors starting unloaded
Single-phase a-c	Two value capacitor	Constant	High	Normal	Small	(b) Pumps and compressors
	Permanent split capacitor	Constant	Low	Normal	Fractional	(a) Fans, Blowers
	Capacitor start	Constant	Moderate	Normal	Small Fractional	(a) Fans and pumps
	Repulsion Induction	Constant	High	Normal	Medium Small	(a) Fans (b) pumps and compressors
	Split phase	Constant and adjustable	Normal	Normal	Fractional	(a) Fans (b) pumps and compressors (d) fans—direct

to values below 50 percent. Fig. 3 illustrates the characteristics of wound-rotor motors.

Synchronous motors are used for continuous duty applications at constant speed where efficiency and power factor are important. Another advantage of these motors is that of lower initial cost in large sizes and for low speeds when compared with squirrel-cage type motors.

The outstanding advantage of the synchronous motor is that its power factor can be changed to compensate for the low power factor of other drives in the same location. Lagging power factor is an inherent characteristic of all induction apparatus, such as induction motors and neon signs. Unless synchronous motors or capacitors are installed, the plant power factor may be comparatively low. This does not

Table 3 .... Classificatiom of Motors (Continued)

	Type	Speed Characteristics	Full Voltage		HP Range	Application See Footnotes (a) to (e)
			Starting Torque	Starting Current		
	Adjustable Speed Drives					
Polyphase a-c	Squirrel-cage high slip. Transformer adjustment	Variable	Normal	Normal	Medium Small	(a) Fans
	Squirrel-cage separate winding or regrouped poles	Constant multi-speed	Normal or high	Normal or low	All	(a) Fans (b) pumps and (c) compressors
	Wound rotor	Variable	High (with secondary control)	Low	All	(a) Fans (b) centrifugal pumps and compressors
Single-phase a-c	Repulsion	Variable	High	Normal	Low and Fractional	(a) Fans, centrifugal pumps (b) compressors
	Capacitor low torque tapped winding	Variable two speed	Low	Normal	Fractional	(d) Fans, direct
	Capacitor low torque transformer adjustment	Variable	Low	Low	Fractional	(d) Fans
	Split phase regrouped poles	Constant	Normal	Normal	Fractional	(d) Fans

a. Drives having medium or low starting torque and inertia WR^2 such as fans and centrifugal pumps or reciprocating pumps and compressors started unloaded.
b. Drives having high starting torques, such as reciprocating pumps and compressors started loaded.
c. Similar to (a) except where frequent or hard starting (large WR^2) requires a higher starting and accelerating torque. d. Fans direct connected. e. Stoker drives.
f. Torque depends on hp rating and speed. See *NEMA Standard* MG1-4.10 on Motors and Generators.

Table 4 .... Locked-rotor Current of Three-phase, 60-cycle Motors at 220 Volts [a, b]

HP	Design B, C, and D Amperes	Design F Amperes	HP	Design B, C, and D Amperes	Design F Amperes
1 or less	24[c]	..	30	435	270
1½	35	..	40	580	360
2	45	..	50	725	450
3	60	..	60	870	540
5	90	..	75	1085	675
7½	120	..	100	1450	900
10	150	..	125	1815	1125
15	220	..	150	2170	1350
20	290	..	200	2900	1800
25	365	..			

[a] The locked-rotor current of three-phase, 60-cycle, constant-speed, induction motors, measured with rated voltage and frequency impressed and with rotor locked, shall not exceed the tabulated values.
[b] Locked-rotor current at other voltages shall be inversely proportional to the voltage.
[c] For 1 hp or less the value is given per hp.

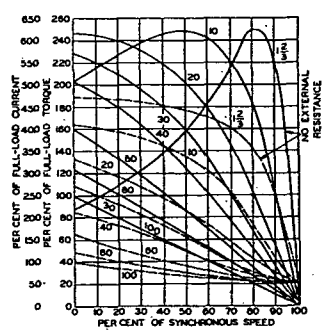


Fig. 3 .... Speed-Torque Speed-Current Characteristics of Wound-rotor Motors

necessarily mean that corrective equipment must always be installed, but in most cases it is desirable to determine what advantages may be gained by improving the power factor. With purchased power, if the rates include a clause embodying a penalty for low power factor, or a bonus for high power factor, the saving in power costs may often make a very good return on the investment required for the corrective equipment.

Synchronous motors are used to drive fans, blowers, pumps, compressors, and other applications. Compressor applications having a high peak torque require the use of flywheels to smooth out power peaks, and should always be referred to the electrical manufacturer for recommendations.

Synchronous motors are provided with built-in damper windings on the rotor and operate during the starting period similarly to squirrel-cage motors. After the motor is nearly up to speed, field excitation is applied and the motor draws into step at synchronous speed. After excitation is applied, the motor runs at exactly constant speed and will remain at this speed until a load approaching the pull-out load is reached, whereupon the motor pulls out of synchronism and stops.

In applying synchronous motors, consideration must be given to the torque the motor can develop on pull-in, that is, at the instant when field excitation is applied. Table 5 shows typical application requirements of synchronous motor drives, listing starting, pull-in, and pull-out torques.

Multi-Speed motors provide flexibility in many types of drives. Synchronous motors can be furnished only with a 2 to 1 ratio in speed, single-winding. Squirrel-cage induction motors may be 2-, 3-, or 4-speed. Two-speed induction motors are usually of single-winding type, having a 2 to 1 speed ratio such as 600 rpm and 1200 rpm, or may be double-winding. Three-speed induction motors are always two-winding, and four-speed motors are usually two-winding with a 2 to 1 speed ratio in each winding. Motors can be provided in constant torque, varying torque, or constant horsepower ratings. The constant horsepower type of motor is considerably larger than the constant torque motor, due to the fact that the same horsepower must be developed at either reduced speed or high speed.

In selecting two-speed motors for fan, pump, blower, or compressor applications, it is usually found that two-winding motors are more expensive than the single-winding type. The control cost for two-speed, two-winding motors, however, is more economical, and therefore the combined price of both motor and control for the two-winding motor is only slightly higher. Because of the improved performance of the two-winding motors, and because of the factor of safety provided by two independent windings, the increased cost is frequently worth the difference.

Single-phase Motors

Single-phase induction motors have auxiliary windings or devices for starting, and are classified by the method used.

Capacitor-start motors develop high starting torque in fractional horsepower ratings, and moderate starting torque in larger ratings. They are used for constant speed drive such as fans, blowers, and centrifugal pumps. During the starting period, a winding with a capacitor in series is connected to the motor circuit and when the motor comes up to speed, a centrifugal switch cuts the capacitor and second winding out of the circuit.

Two-value-capacitor motors develop high starting torque employing a starting capacitor and a running capacitor. The starting capacitor gives high starting ability, but is suited for short time operation only, and is cut out for the running condition by a centrifugal switch. The running capacitor gives high efficiency at full speed. These motors are used on compressors, reciprocating pumps, and similar equipment which may start under heavy load.

Permanent-split-capacitor motors have low starting torque and are ideally suited for small fan drives. Operation is similar to the capacitor-start motor, except that the capacitor is not cut out when running.

Repulsion-start Induction-run motors develop high starting torque. The motors have a wound rotor connected to a commutator for starting. A centrifugal switch shorts the commutator bars when the motor comes up to speed to obtain a winding approximately like the squirrel-cage in its function.

Repulsion-start Induction-run motors are suitable for applications, such as industrial compressors, requiring high breakaway torque, and where commutator and brush noise are not factors.

Split-phase motors have a high-resistance auxiliary winding which is in the circuit during starting, but is disconnected through the action of a centrifugal switch as the motor comes up to speed. Under running conditions it operates as a single-phase induction motor with one winding in the circuit. These units are available for the small horsepower ratings, and when equipped with a high-slip rotor, may be used for adjustable varying speeds through line-voltage control. The motors are ideally suited for fan duty.

Speed-torque characteristics of single-phase motors are shown in Fig. 4.

Full-voltage starting of single-phase motors is general practice, but since most of these motors are connected to secondary distribution systems located in light-load density areas, power companies check carefully the starting currents in order to prevent objectionable voltage dips. This is particularly important for motors started frequently, such as those controlled by pressure or temperature-sensitive devices and applied to refrigerators, stokers, oil burners, and water pumps.

The report of a joint committee of AIEC-EEI-NEMA recommended three application rules taking into account the greater annoyances resulting from frequent motor starting during lighting hours against infrequent starting at any time. Table 6 shows typical applications of these rules which are becoming widely accepted for full-voltage starting. Rules 1 and 2 apply to general use.

Rule 1 limits the locked-rotor current to 20 amp at 115 volts, and 25 amp at 230 volts when automatically controlled (usually frequently started) motors are used. Rule 2 permits twice these values when manually controlled (usually infrequent starting) motors are used. Rule 3 applies to special conditions where larger currents may be allowed above Rules 1 and 2 upon approval of the electric company.

Hermetically-enclosed single-phase a-c motors are widely used in appliances such as domestic and commercial refrigerators, room air conditioners, beverage coolers, freezer chests, and water coolers. In such appliances, the motors are sealed inside the compressor, and the gas and oil circulate freely within and around the motor. Centrifugal starting switches are not used on such motors because any arcing in the gas and oil atmosphere would be detrimental, and also because the units are permanently sealed and cannot be serviced or repaired in the field. In place of a centrifugal switch, an external relay is used to start hermetically-enclosed motors.

Hermetically-enclosed motors for refrigeration compressors will, in the future, have the horsepower rating eliminated. The full-load current is to be the operating current in amperes, when the compressor is delivering rated output. The motors for hermetically-enclosed service are usually of the

Table 5 .... Typical Application Requirements of Synchronous Motor Drives Showing Starting, Pull-In, and Pull-Out Torques

	Application	Method of Connecting Motor to Load	Starting Conditions	Torques			Remarks
				Starting	Pull-In	Pull-Out	
Fans	Exhaust and ventilating	Coupled or belted	Usually loaded	50	60–125	150	WR^2 of fan must be considered
Blowers	Cycloidal positive	Coupled or engine type	Unloaded	40–60	40–60	150	Two-speed motors sometimes used
	Blowing engines reciprocating	Engine-type	Unloaded	40	40–60	150	
	Turbo high speed	Direct connected or step up gear	Unloaded (intake closed)	30	50	150	WR^2 of blower must be considered
Compressors	Air	Engine-type	Unloaded	40	30	150	Flywheel effect important
	Ammonia and ammonia booster	High speed—belted Low speed—engine-type occasionally coupled	Unloaded (by bypass)	40	30	150	Flywheel effect important
	Freon	High speed—belted Low speed—engine	Unloaded (by bypass)	45	50	150	Flywheel effect important
	Gas reciprocating	High speed—belted Low speed—engine	Unloaded (by bypass)	40	30	150	Flywheel effect important

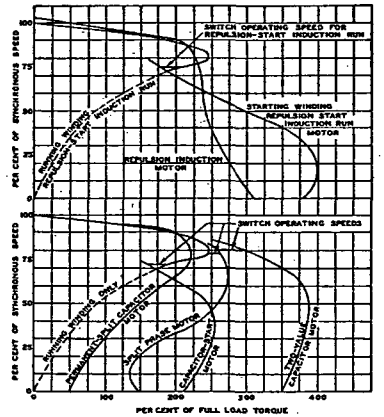


Fig. 4 .... Speed Torque Characteristics of Single-phase Motors

Table 6 .... Recommended Single-Phase Motor Ratings for Full-voltage Starting

HP Rating	Locked-rotor Current[a] at 25 C Amperes		General Use (Rules 1 & 2)		Special Conditions (with Utility Permission) (Rule 3)	
	115 V	230 V	115 V	230 V	115 V	230 V
1/20	20	10	A-M	A-M	A-M	A-M
1/8	20	10	A-M	A-M	A-M	A-M
1/6	20	10	A-M	A-M	A-M	A-M
1/4	23	11.5	A-M	A-M	A-M	A-M
1/3	31	15.5	M	A-M	A-M	A-M
1/2	45	22.5	M	A-M	A-M	A-M
3/4	61	30.5		M	A-M	A-M
1	70	35		M	A-M	A-M
1 1/2		40		M		A-M
2		50		M		A-M
3		70				A-M
5		100				A-M

A refers to automatically controlled devices.
M refers to manually controlled devices.
[a] These values of Locked-rotor current are the same as the NEMA standard for single phase Design M motors.

split-phase type or of the capacitor-start (or two-value capacitor) type.

For starting the split-phase type of motor, two methods are generally acceptable. In one method a thermally-operated time delay is used. When the motor is started, a contact which is normally closed applies power to the starting winding. A thermal element which controls these contacts is in series with the motor and carries line current. Due to the current flow through this element, it is heated until, after a definite period of time, it is warmed sufficiently to open the contacts and remove power from the starting winding. The running current then heats the element enough to keep the contacts open. The setting of the time for the starting contacts to open is determined by tests on the system components, i.e., the relay, the motor, and the compressor, and is based on a prediction of the amount of time delay required to bring the motor up to speed.

The second method of starting split-phase motors utilizes a series-coil or current-operated relay. This device is affected indirectly by motor speed.

In this type of connection, a relay coil carries the line current going to the motor. When the motor is started, the inrush current to the running winding, passing through the relay coil, causes the contacts to close, and applies power to the starting winding. As the motor comes up to speed, the current decreases until at a definite calibrated value of current corresponding to a preselected speed, the magnetic force of the coil diminishes to a point which allows the contacts to open to remove power from the starting winding.

Capacitor-start and *two-value-capacitor hermetically-sealed* motors are usually started with a voltage-type relay. In this method of starting the relay coil is connected in parallel with the starting winding. When power is applied to the line, the relay does not operate because it is calibrated to operate on a higher voltage. As the motor comes up to speed, the voltage across the starting winding and relay coil increases in proportion to the motor speed. At a definite voltage corresponding to a preselected speed, the relay operates and opens its contacts, thereby opening the starting-winding circuit. The relay then keeps these contacts open because there is sufficient voltage induced in the starting winding, when the motor is running, to hold the relay in the open contact position.

CONTROL FOR ALTERNATING-CURRENT MOTORS

Squirrel-cage motors are usually started at line voltage where power company limitations permit. In sizes up to 5 hp at 220 volts, or 7½ hp at 440 volts, polyphase motors may be started by means of manual switches having overload current elements for motor protection. In larger ratings a linestarter is usually provided with either an additional safety switch or circuit breaker for disconnecting and short-circuit protection. Reduced-voltage starting may be used when the current inrush on full voltage exceeds the limitations set by the power company. The magnitude of this inrush current is entirely independent of the load on the motor, and is the same whether the motor is lightly loaded or heavily loaded.

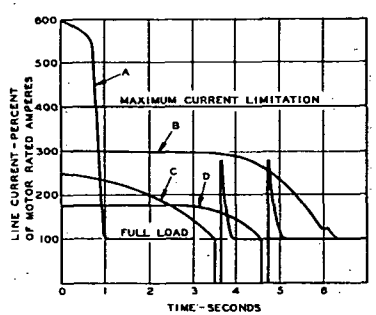
Restrictions on starting current may limit the maximum current inrush or may limit the increments of current inrush which may be drawn on each step. Table 7 shows a comparison of the various methods of reduced-voltage starting. The choice may depend upon the limitation to be met. Fig. 5 shows examples for maximum current limitations. Fig. 6 shows examples for increment limitations. The increment starters would not be satisfactory for meeting maximum current limitations as they exceed the maximum value before the motor reaches full speed. Both Curves *B* and *C* of Fig. 6 would be satisfactory if the increment limitation were 400 percent but only the starter illustrated by Curve *C* would meet a limitation of 200 percent increments.

Table 7 . . . . Comparison of Squirrel-Cage and Synchronous Motor Starting Methods

Type of Starter	Percent of Motor Rated Values		
	Voltage	Starting Torque	Line Current
Full Voltage Starter	100	100	100
Autotransformer			
80 percent tap	80	64	68
65 percent tap	65	42	46
50 percent tap	50	25	30
Resistor, single step	80	64	80
Reactor			
50 percent tap	50	25	50
45 percent tap	45	20	45
37.5 percent tap	37.5	14	37.5
Part Winding	60*	50*	60*
Star-delta	58	33	33

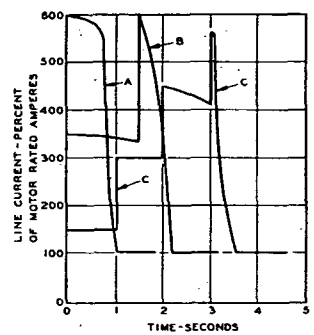
* Varies with portion of winding used.

When increment starters are used, the motor may not develop enough torque to start until the last step. Fig. 6 shows that with the resistor step starter the motor does not start until Step 3 or after 2 seconds on Curve *C*. Part-winding motors may start and then stall at about half speed until the second step which applies full torque.



Curve A—Full Voltage Starter
Curve B—Reactor 50 Percent Tap
Curve C—Autotransformer 65 Percent Tap

Fig. 5 . . . . Starters for Maximum Current Limitations

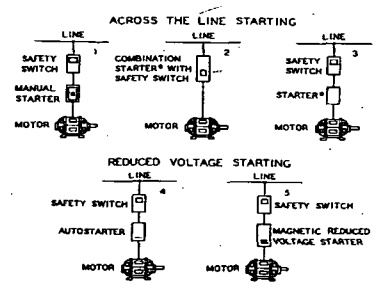


Curve A—Full Voltage Starter
Curve B—Part Winding Starter
Curve C—4-Step Resister Increment Starter

Fig. 6 . . . . Starters for Increment Limitations

The load inertia WK² may also affect the selection of reduced current starting method since the reduced torques may not be sufficient to accelerate the load without overheating the motor or in the case of synchronous motors, may not reach sufficient speed to pull into synchronous speed. In these cases the solution may be to use the wound-rotor induction motor or some means of reducing the load such as suction dampers on compressors. Fig. 7 illustrates recommended control practice for squirrel-cage motors.

Wound-rotor motors require control of both primary and secondary circuits. The primary* control may be the same as for squirrel-cage motors, manual or magnetic, at full voltage. Secondary* control provides means of varying secondary resistance for starting and speed control. The secondary controller should be specified for starting duty only, or for speed regulating duty. If the secondary controller is to be used for speed-regulating duty, the percent speed reduction, the number of speed control points, and the type of load (variable or constant torque) should be specified. Fig. 8 illustrates recommended control practice for wound-rotor motors.



Arrangements 2, 3, 4, and 5 Provide Automatic Push-button Starting.

Fig. 7 . . . . Recommended Controls for Squirrel-Cage Motors

Synchronous motor starters should provide pull-out protection, automatic synchronization or automatic stopping of the motor after pull-out, and assurance of complete starting sequence, as well as overload and low-voltage protection. The control may be either magnetic or semi-magnetic at full or reduced voltage. Semi-magnetic starters provide automatic field control, but require hand operation for closing the line contactors to start and transfer to full voltage.

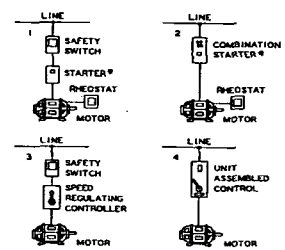


Fig. 8 . . . . Recommended Controls for Wound-Rotor Motors

In applying reduced-voltage starters to synchronous motors it should be remembered that, since these motors are started on damper windings and function during the acceleration period similarly to squirrel-cage motors, the starting torque varies as the square of the applied voltage. Consideration should be given to insure development of sufficient motor torque to accelerate the load.

Multi-speed control may be either manual or magnetic, and at full or reduced voltage. When using automatic magnetic control with two-, three-, and four-speed separate winding or consequent-pole motors, control may be obtained from a remote point by means of a push button master switch. The various speeds of the motor are obtained from the master switch by simply depressing the correct push button. This is known as selective speed control. It is commonly used in the smaller theater installations where the fan and motor are located backstage and the speed control is located in the lobby.

Multi-speed motor controllers may be provided with compelling relays which make is necessary for the operator to press the first speed button before regulating the motor to the desired speed. This ensures that the motor is always started at low speed before adjusting to a higher speed.

Timing relays which provide for automatic acceleration may be used for control. With this feature the motor will always start at low speed and automatically accelerate to the desired speed. Decelerating relays may be used to reduce the shock effect of the braking action on the motor and drive when the speed is reduced from a higher to a lower speed.

Single-phase motor control usually consists only of an across-the-line starter, either manual or magnetic. In some

*Refer to Glossary at end of chapter.

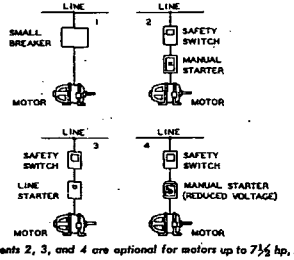
cases it is desirable also to provide a disconnect switch. Fig. 9 illustrates the recommended controls.

MOTOR RATING

The rating of an electric motor depends upon the total temperature which the motor attains under operating conditions. This total temperature depends on both the ambient temperature and the temperature rise of the motor. As motor temperature rise is in turn determined by the ability of the motor to dissipate heat, circulation to the motor should not be restricted. Improper selection of motors with regard to temperature ratings may result in high motor operating temperatures and accompanying reduction in motor life.

FUNCTIONS OF CONTROL EQUIPMENT FOR MOTORS

In general, control equipment for all types of motors should provide (1) means of disconnecting the motor and controller from the power supply, (2) means for starting the motor, (3) overload protection for the motor, (4) protection against low voltage, and (5) means for varying the motor speed.



Arrangements 2, 3, and 4 are optional for motors up to 7½ hp, 220 volts.

Fig. 9 . . . . Recommended Controls for Single-Phase Motors

Full-voltage starting for motors is preferable because of its lower first cost and simplicity of control. Except for d-c machines, most motors are mechanically and electrically designed for full-voltage starting. The starting inrush current, however, is limited in many cases by regulations of power companies because of the voltage fluctuations which may be caused by heavy current surges. It is therefore often necessary to reduce the starting current below that obtained by across-the-line starting. The power supplier should be consulted to determine the allowable inrush current for any given location.

The choice between full-voltage and reduced-voltage starting is governed almost entirely by inrush current limitations. The starting torque of all motors varies with the starting current, and it is therefore necessary to insure that the motor is supplied with sufficient current to develop enough torque to accelerate the load.

In present practice overload protection of motors is obtained by use of thermal-overload inverse-time-limit type protection. The usual setting of such protection devices is not to exceed 125 percent of rated full load current for open 40 C deg rise motors, and not to exceed 115 percent of rated full-load current for all other motors, the element tripping after a definite interval of time. The *National Electrical Code* requires the addition of fuses or circuit breakers to protect the overload elements from severe short-circuit currents.

Two types of protection are available against low voltage at the motor terminals. One type, called low-voltage release, permits the motor line contactor to drop out on low voltage and to close again when the voltage returns to normal, thereby restarting the motor when the abnormal condition is ended. The second type, called low-voltage protection, causes the motor line contactor to drop out on low voltage, but prevents restarting when the voltage returns to normal except by the action of an operator. This latter type of protection is desirable where it is necessary for the operator to make initial starting adjustments on the machine.

Manual control for an alternating- or a direct-current motor is usually located near the motor. When so located an operator must be present to start and stop or change the speed of the motor by operating the control mechanism. Manual control is sometimes employed only as a device to give overload protection, and another device is employed to start and stop the motor. Manual control is used particularly on small motors which operate unit heaters, small blowers, and room coolers in an air-conditioning system. In other cases manual control in the form of drums, when used with multi-speed motors, is used only as a speed setting device, while the *starting and stopping functions operate automatically* through thermostats and pressure switches.

Because of the increasing complexity of air-conditioning systems, the equipment is operated preferably by automatic control, and less dependence is placed on manual operation and regulation.

Automatic control of motor starters may be accomplished by the use of remote push button stations, by a thermostat, float switch, pressure regulator, or other similar pilot devices. An added advantage of automatic control is that the main wiring for the starter may be installed near the motor, while the starter may be operated by a remote control device.

GLOSSARY

General Definitions

NEMA is the abbreviation for the National Electrical Manufacturers Association.

AEIC is the abbreviation for the Association of Edison Illuminating Companies.

EEI is the abbreviation for the Edison Electric Institute.

Speed Regulation (d-c motors) is the change in speed between no load and full load, expressed in percent of full-load speed; for example, a motor having a no-load speed of 1200 rpm and a full-load speed of 1140 rpm would have a speed regulation of 5.26 percent.

Slip (a-c induction motors) is the difference between the motor speed and synchronous speed expressed in percent of synchronous speed, e.g., a 1200-rpm motor operating at 1140 rpm would have a slip of 5 percent.

Torque is an expression of the turning effort developed by the motor at the shaft, and is usually expressed in ounce-feet for fractional horsepower motors, and in pound-feet for motors of larger ratings.

Primary is the term usually applied to the high voltage or line side of a transformer or motor. In the case of the wound rotor motor the primary is the stator winding.

Secondary is the term usually applied to the low voltage or load side of a transformer or motor. In the case of the wound rotor motor the secondary is the rotor winding.

NEMA Classification by Mechanical Protection and Method of Cooling

These classifications are from NEMA publications MGI-1.20 and MGI-1.21.

Open Machine

An open machine is one having ventilating openings which permit passage of external cooling air over and around the windings of the machine.

a. *Drip-proof Machine.* An open machine in which the ventilating openings are so constructed that drops of liquid or solid particles falling on the machine at any angle not greater than 15 degrees from the vertical cannot enter the machine either directly or by striking and running along a horizontal or inwardly inclined surface of the machine.

b. *Splash-proof Machine.* An open machine in which the ventilating openings are so constructed that drops of liquid or solid particles falling on the machine or coming towards it in a straight line at any angle not greater than 100 degrees from the vertical cannot enter the machine either directly or by striking and running along a surface of the machine.

c. *Semi-guarded Machine.* An open machine in which part of the ventilating openings in the machine, usually in the top half, are guarded as in the case of a "guarded machine" but the others are left open.

d. *Guarded Machine.* An open machine in which all openings giving direct access to live or rotating parts (except smooth shafts) are limited in size by the design of the structural parts or by screens, grilles, expanded metal, etc., to prevent accidental contact with such parts. Such openings shall not permit the passage of a cylindrical rod ½ inch in diameter, except that, where the distance from the guard to the live or rotating parts is more than 4 inches, they shall not permit the passage of a cylindrical rod ¾ inch in diameter.

e. *Drip-proof Fully Guarded Machine.* A drip-proof machine whose ventilating openings are guarded in accordance with paragraph d.

f. *Open Externally-ventilated Machine.* A machine ventilated by means of a separate motor-driven blower mounted on the machine enclosure. Mechanical protection may be as defined in paragraphs a to e, inclusive.

g. *Open Pipe-ventilated Machine.* An open machine except that openings for the admission of the ventilating air are so arranged that inlet ducts or pipes can be connected to them. This air may be circulated by means integral with the machine or by means external to and not a part of the machine. In the latter case, this machine is sometimes known as separately- or forced-ventilated machine. Enclosures may be as defined in par. a to e, inclusive.

h. *Weather-protected Machine. Type I*—A. An open machine with its ventilating passages so constructed as to minimize the entrance of rain, snow and air-borne particles to the electric parts and having its ventilated openings so constructed as to prevent the passage of a cylindrical rod ¾ in. in diameter.

Type II—A machine having, in addition to the enclosure defined for a weather-protected Type I machine, its ventilating passages at both intake and discharge so arranged that high-velocity air and air-borne particles blown into the machine by storms or high winds can be discharged without entering the internal ventilating passages leading directly to the electric parts of the machine itself. The normal path of the ventilating air which enters the electric parts of the machine shall be so arranged by baffling or separate housings as to provide at least three abrupt changes in direction, none of which shall be less than 90 deg. In addition, an area of low velocity not exceeding 600 fpm shall be provided in the intake air path to minimize the possibility of moisture or dirt being carried into the electric parts of the machine.

Totally-enclosed Machine

A totally-enclosed machine is one so enclosed as to prevent the free exchange of air between the inside and the outside of the case but not sufficiently enclosed to be termed air-tight.

a. *Totally-enclosed Nonventilated Machine.* A totally-enclosed machine which is not equipped for cooling by means external to the enclosing parts.

b. *Totally-enclosed Fan-cooled Machine.* A totally-enclosed machine equipped for exterior cooling by means of a fan or fans integral with the machine but external to the enclosing parts.

c. *Explosion-proof Machine.* A totally-enclosed machine whose enclosure is designed and constructed to withstand an explosion of a specified gas or vapor which may occur within it and to prevent the ignition of the specified gas or vapor surrounding the machine by sparks, flashes or explosions of the specified gas or vapor which may occur within the machine casing.

Note—See National Electrical Code Article 500—For Hazardous Locations, Class I, Groups A, B, C or D.

d. *Dust-ignition-proof Machine.* A totally-enclosed machine whose enclosure is designed and constructed in a manner which will exclude ignitable amounts of dust or amounts which might affect performance or rating, and which will not permit arcs, sparks, or heat otherwise generated or liberated inside of the enclosure to cause ignition of exterior accumulations or atmospheric suspensions of a specific dust on or in the vicinity of the enclosure.

Note I—Successful operation of this type of machine requires avoidance of overheating from such causes as excessive overloads, stalling or accumulation of excessive quantities of dust on the machine.

Note II—See National Electrical Code Article 500—For Hazardous Locations, Class II, Groups E, F or G.

e. *Water-proof Machine.* A totally-enclosed machine so constructed that it will exclude water applied in the form of a stream from a hose, except that leakage may occur around the shaft provided it is prevented from entering the oil reservoir and provision is made for automatically draining the machine. The means for automatic draining may be a check valve or a tapped hole at the lowest part of the frame which will serve for application of a drain pipe.

Note—A common form of test for a water-proof machine is to play on the machine a stream of water from a hose with a one-inch nozzle delivering at least 65 gpm from a distance of about 10 ft, from any direction, and for a period of not less than 5 min.

f. *Totally-enclosed Pipe-ventilated Machine.* A totally-enclosed pipe-ventilated machine is a totally-enclosed machine except for openings so arranged that inlet and outlet ducts or pipes may be connected to them for the admission and discharge of the ventilating air. This air may be circulated by means integral with the machine or by means external to and not a part of the machine. In the latter case, these machines shall be known as separately- or forced-ventilated machines.

g. *Totally-enclosed Water-cooled Machine.* A totally-enclosed machine which is cooled by circulating water, the water or water conductors coming in direct contact with the machine parts.

h. *Totally-enclosed Water-air-cooled Machine.* A totally-enclosed machine which is cooled by circulating air which, in turn, is cooled by circulating water. It is provided with a water-cooled heat exchanger for cooling the ventilating air and a fan or fans, integral with the rotor shaft or separate, for circulating the ventilating air.

i. *Totally-enclosed Air-to-air cooled Machine.* A totally-enclosed machine which is cooled by circulating the internal air through a heat exchanger which, in turn, is cooled by circulating external air. It is provided with an air-to-air heat exchanger for cooling the ventilating air and a fan or fans, integral with the rotor shaft or separate, for circulating the internal air and a separate fan for circulating the external air.

j. *Totally-enclosed, Fan-cooled Guarded Machine.* A totally-enclosed, fan-cooled machine in which all openings giving direct access to the fan are limited in size by the design of the structural parts or by screens, grilles, expanded metal, etc., to prevent accidental contact with the fan. Such openings shall not permit the passage of a cylindrical rod ½ in. in diameter except that,

where the distance from the guard to the fan is more than 4 in., they shall not permit the passage of a cylindrical rod ¾ in. in diameter.

Motor Speed Classifications

A Constant-speed motor is one in which the speed remains practically constant with changes in load; e.g., a d-c shunt-wound motor or a-c squirrel-cage motor with low slip.

A Varying-speed motor is one in which the speed varies with the load, usually decreasing when the load increases; e.g., a d-c series motor or an induction motor with large slip.

An Adjustable-varying-speed motor is one in which the speed can be adjusted gradually, but when once adjusted for a given load will vary in considerable degree with change in load; e.g., a shunt-wound d-c motor adjusted by armature resistance control.

An Adjustable-speed motor is one in which the speed can be varied gradually over a considerable range, but when once adjusted remains practically unaffected by the load; e.g., a d-c shunt motor with field-resistance control. The standard ratings for open-type, adjustable-speed motors, having a speed range of 3 to 1 and greater are in accordance with the following:

(1) A standard continuous horsepower rating at 150 percent of minimum speed with a temperature rise of 40 C.

(2) The next higher standard continuous horsepower rating at 3 times minimum speed with a temperature rise of 40 C.

(3) Between 150 percent of minimum speed and 3 times minimum speed, the standard continuous horsepower rating with a temperature rise of 40 C will vary with the speed along a straight line connecting these two horsepower ratings. No further increase in horsepower is recognized above 3 times minimum speed.

(4) Below 150 percent of minimum speed the lower continuous horsepower rating (see preceding item 1) will apply with a temperature rise of 50 C.

Example: 20/25 hp, 400 to 1600 rpm. This motor may be rated 20 hp, 40 C at 600 rpm and 25 hp, 40 C from 1200 to 1600 rpm. Between 600 and 1200 rpm the rated horsepower increases directly with speed from 20 to 25 hp.

(5) Motors may also be rated 1 hour with temperature rise of 50 C with the higher horsepower rating (see preceding item 2) throughout the entire speed range.

Example: 20/25 hp, 400 to 1600 rpm. This motor may be rated 25 hp, 50 C 400/1600 rpm; 1 hour.

Mechanical Modifications

Vertical Mountings are available for such applications as pumps and agitators. This type of application may require a special umbrella-type hood to protect against dripping liquids.

Flanged Mountings are available for use where motors are built in as part of machines. Motors may also be supplied with flush plate mountings, suitable for close coupled pump and similar applications.

CHAPTER 46

RESIDENTIAL SUMMER AIR CONDITIONING

Consumer Requirements, Equipment Capacity Selection, Types of Equipment, Types of Systems, Gas Year-round Air Conditioners, Location of Cooling Equipment, Noise, Air Distribution Methods, Operating Costs, Effects on Future House Design

SUMMER air conditioning in residences has become a major factor in the air-conditioning field and has created a heavy interest among home-owners, builders, architects and business men. Residential cooling, particularly for the small home, received its initial impetus when equipment specifically designed for such purposes became available in the 1930's.

The first installations were made by using commercial and industrial methods as the estimating basis and did not prove as satisfactory as the installer or designer desired. Actually, the difference between estimating the residential summer air-conditioning and commercial summer air-conditioning loads is in the design temperature differences used, the type of internal load in the conditioned area, and the method of calculating equipment size.

CONSUMER REQUIREMENTS

Present practice is to design residential summer air conditioning on the basis of an 80 F indoor temperature. In using this basis it is assumed that the home owner will operate the system continuously with a thermostat setting of 75 or 76 F and thereby take advantage of the thermal storage in the house and household furnishings to reduce the peak temperatures. The preference of the individual home owner must, however, be considered in selecting the design indoor temperature and may require that a temperature lower than 80 F be selected. Analog computed studies and field investigations[1] have apparently demonstrated the adequacy of the 80 F indoor temperature as a basis for equipment selection procedure.

To ensure comfort, it is essential that the proposed operation and performance of the system be explained to the user in advance. Usually the greatest degree of comfort can be obtained by keeping the house closed and operating with a fixed thermostat setting throughout the cooling season. In some climates the opening of the house to utilize *night-air cooling* creates humidity difficulties that take several hours to overcome during the following day. The use of *night-air cooling* may lead to unsatisfactory conditions when the equipment is not turned on again sufficiently early the next day. It will also allow unfiltered air to enter the house thus reducing the cleanliness advantage of air conditioning.

EQUIPMENT CAPACITY SELECTION

There are two industry-accepted methods of calculating residential cooling loads. These are as given in Manual 11, *Summer Air Conditioning*, of the National Warm Air Heating and Air Conditioning Association and Standard 610-56, *Application Engineering Standard for Year-Round Residential Air-Conditioning* published by the Air Conditioning and Refrigeration Institute. Published procedures of reliable manufacturers when used in conjunction with their equipment are also adequate.

Both of the industry methods mentioned employ cooling load factors for the various components of the load that are averaged over a peak period of several hours and thereby make allowance for thermal storage effects. They avoid the possibility of pyramiding non-concurrent, instantaneous peak loads.

With most methods it is possible to select the desired design indoor-outdoor temperature difference, using 80 F or the user's preference for the indoor temperature, and the commonly used outdoor temperature for the area as listed in Chapter 13.

Design methods which average load conditions for a period in excess of 12 hr have not been accepted by the industry committees. Such methods require the introduction of factors that are not at present considered to be adequately substantiated by mathematical analysis or to be justified by the thermal storage effects of a typical residence.

The contractor should avoid any capacity selection procedure based on a rule of thumb such as *10 average windows or doors per ton* or *500 sq ft of floor area per ton*. Insulation, glass areas, shading effects, and solar orientation must be given careful consideration by using an accurate, step-by-step procedure for estimating the load.

In residences, peak lighting loads will not usually occur during the period of greatest load from outdoor temperature and solar conditions. Unlike summer air conditioning in commercial or industrial establishments where internal heat producing sources may be a major portion of the load, internal loads in residences may generally be neglected except under the most unusual circumstances.

The heat and moisture introduced by a clothes dryer or water heater should be vented to outdoors. Heat introduced by the kitchen range should be discharged through an exhaust fan. These are major internal loads that must be taken into account if adequate venting is not to be provided.

The load calculation method presented in Chapter 13 is not recommended for residences where it has been found[2] to result in oversizing because internal loads that are generally a prime consideration in commercial applications are not generally significant in residences. Furthermore the method outlined in Chapter 13 gives an instantaneous peak load without consideration of residential usage and heat storage effects applied in the ARI and NWAH & ACA load calculation procedure. Oversizing is objectionable from at least two viewpoints: (1) it increases the capacity and consequently the

cost of the equipment installed, and (2) it results in less desirable humidity conditions within the residence because of the increased size.

TYPES OF EQUIPMENT

Low Side

Three types of low side equipment (the equipment that actually removes the heat from the conditioned space) have been developed for residential work as follows: (1) refrigerated coils, (2) chilled-water coils, and (3) water-cooled coils.

The first and most commonly used method of absorbing heat from the conditioned air is to use direct evaporation of the refrigerant in a coil with the air to be conditioned passing over the outside of the coil surface. The heat is passed directly from the air into the refrigerant for removal. This requires the use of an air distribution system and, although generally most economical in new construction, may present costly problems in modernization work where ductwork is not installed.

In a chilled-water system, water is circulated through a closed circuit containing two coils. In one coil the water removes heat and moisture from the air in the conditioned space. In the other coil (water chiller) the heat is removed from the water by mechanical refrigeration. This type of system is called a duplex system in that the heat is picked up by chilled water and later removed from the water by means of a refrigeration system.

The water-cooled coil using water directly from city mains or wells is the simplest type of installation but, because of limitations imposed by the capacity, cost, or temperature of the available water supply, its use is not generally practicable. Cold water is circulated directly through the coil while air from the conditioned space is passed over the coil to remove heat and moisture. After the water has passed through the coil, it is wasted. To operate satisfactorily, an installation of this type requires water of sufficiently low temperature to chill the air enough to remove the desired amounts of heat and moisture. Where a large supply of cool water is obtainable, the water-cooled coil will operate with satisfactory results. As a general rule a water-cooled coil should not be used unless there is a supply of 52 F or colder water at the peak of the cooling season and in sufficient quantities to take care of the cooling load with a temperature rise of no more than 10 deg. Where water is not obtainable in sufficient quantities or at the required temperatures, it is necessary to chill the water to the desired temperature by using mechanical refrigeration.

High Side

The high side of a conditioning system, the section of equipment that removes the heat from the refrigerant and reclaims the refrigerant for further use in the system may be either the air or water-cooled type. The air-cooled high side is so constructed that the heat is removed from the compressed or hot refrigerant by passing it into a cooling coil over which outside air (air taken from outside the conditioned area and returned to outside of the conditioned area) is circulated.

The major advantage of air-cooled equipment is that no water is required to accomplish the removal of heat from the refrigerant and only an electrical supply is needed. The disadvantage is that operating pressures are higher because of the inability to cool the refrigerant as much by air as by water. This results in an increase in electrical input to the compressor motor.

Water-cooled equipment can be of two types. In one type the water drawn from a well or from city mains is passed through the condenser of the high side and is allowed to flow to a drain, swimming pool, a wetted roof, or to other means of disposal. Where the water supply is sufficient and at low cost, this type of water-cooled unit is the better type installation because of lower equipment cost, lower operating costs and quieter operation. In a second type, if water is in short supply or is expensive, about 95 percent of the cooling water can be reclaimed by evaporating sufficient water to rid the remaining water of the heat picked up in the condensing unit. This can be done by the use of an evaporative condenser or a cooling tower described in Chapters 38 and 40.

TYPES OF SYSTEMS

The present generally accepted method of cooling residences is to employ a central system. The main components of the central plant for year-round air-conditioning systems may be obtained in factory-made assemblies providing matched components for both winter and summer conditioning functions or they may be obtained in one or more separate parts and combined on the job into an integrated system. Summer conditioning equipment may be obtained separately and added to existing heating systems or may be installed independently of any heating system. Cooling coils may be used in conjunction with warm-air furnaces and duct systems, or chillers may be combined with boilers to provide hot and cold water for circulation to room units as required. Refer to other Chapters 15, 16, 19, 42, and 43 for information on specific system components.

Self-contained room coolers, which are discussed in Chapter 16, are used for room or spot cooling and are not generally used to cool an entire residence. Their application and use is not to be considered further, but it may be noted that when only a portion of a residence is cooled the load calculation methods currently used in the industry for central residential systems are not applicable. Refer to Chapter 13 when calculating the load for such applications.

In some areas there is little or no need for heating but there is a distinct need for summer cooling. Both air- and water-cooled package cooling units are available for such installations, ranging in size from one ton to at least 15 tons capacity. To install these units it is only necessary to make plumbing and electrical connections and add the duct work required to convey air to and from the conditioned space. These units can also be used to provide central air conditioning for residences

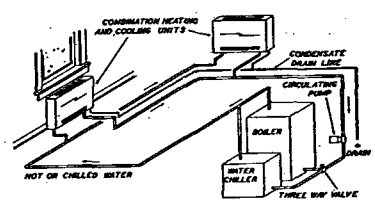


Fig. 1 . . . . Combination Summer Air-Conditioning and Hot Water Heating System

having heating systems which do not contain provision for attaching summer cooling equipment.

Year-round air conditioners require less space than a separate heating plant plus a separate summer conditioner. Usually the heating and cooling portions of a year-round air conditioner operate independently of each other. The air circulated by the fan passes through the cooling or heating portion of the equipment, only one of which is in operation at any one time.

Air conditioning can be installed in homes having a hot-water heating system by use of either of the following:

1. *Room units resembling convectors and containing* both heating and cooling coils. These units are used in place of the conventional radiator or convector. They can be installed in all rooms of the house to provide complete summer cooling, or they can be used in only one or two rooms. This arrangement is shown in Fig. 1.

2. A *split system*. The cooling coil is located in a duct through which air from the rooms is circulated, cooled, and returned to the rooms. Another coil in the same duct can be used for winter heating, or conventional radiators, convectors, or panels can be used to bring heat to the rooms.

When the cooling equipment to be used in houses equipped with forced-circulation hot water heating is of the type which produces chilled water, it may be located in the basement or utility room near the boiler. In most of such systems it is then possible, with the addition of suitable valves, to use the same piping system for circulation of heated water in winter and chilled water in summer. The piping must be insulated to prevent sweating when circulating chilled water.

GAS YEAR-ROUND AIR CONDITIONERS

Gas-fired year-around air conditioners, based on the absorption cycle are available in sizes suitable for residential use and small commercial applications.

The conditioners are built in 3½ ton and 5 ton sizes with heating capacities appropriate for various climates. The 3½ ton and one model of the 5 ton units have heating inputs of 120,000 Btuh. This makes them suitable for homes having design heat losses up to 76,800 Btuh. The other 5 ton model has an input of 180,000 Btuh making it suitable for homes having design heat losses up to 115,200 Btuh.

General Description

The principal components of the gas-fired year-round air conditioner are an absorption unit, a gas burner, a centrifugal fan for air circulation, a filter section, and controls. The gas flame supplies the heat to actuate the absorption unit which provides heating in winter and cooling in summer.

The entire unit operates under a vacuum at all times. The absolute pressure within the generator and condenser on cooling cycle is of the order of 2 to 2½ in. Hg (50 to 60 mm) absolute pressure. The pressure within the cooling coil and absorber is between 0.25 to 0.3 in. Hg (6 to 9 mm) pressure. On the heating cycle, since the same coil is used for both heating and cooling, the absolute pressure is on the order of 14 in. Hg (350 mm) or about ½ atmosphere.

Heating Cycle

The gas flame, when applied to the generator of the absorption unit in winter, boils the solution. The water vapor which is thus produced goes directly (bypasses the condenser) to the single coil over which air is passed by the blower. The heat is taken from the water vapor, and the water is returned to the generator by gravity to be used again.

Cooling Cycle

The gas flame, when applied to the generator of the absorption unit, boils the solution. The water vapor which is thus produced goes to the condenser where this vapor is condensed. It then flows into the single coil (which is the evaporator on the cooling cycle). The air being cooled and dehumidified is blown over this coil. The water vapor (through the process of absorption) is returned to the liquid state and flows by gravity to the generator to be used again. The refrigeration unit is factory-sealed and contains a solution of lithium bromide and water which acts as the absorbent, water vapor being the refrigerant. Circulation of these fluids in the system is accomplished by differences in temperature and height of liquid columns, without moving parts.

Control System

Manual switches permit the owner to select heating or cooling, constant or automatic fan operation, and turn the system *on or off*. For protection of the unit a generator limit switch and a cooling circuit preventive control switch are provided. The former protects the generator from excessive input and is preset to operate at safe temperature levels both on the heating and cooling cycles. The preventive control switch keeps the unit from serving as a heating plant if the condensing water were cut off. Another protective device used in the year-round gas air conditioner is the low temperature control which prevents the unit from freezing as a result of improper equipment operation.

A fan delay control delays fan operation until the heating coil has reached 140 F so as to prevent cold air circulation when heat is desired.

LOCATION OF COOLING EQUIPMENT

The preferred location for summer cooling equipment will depend upon the type. In a combination self-contained unit, the location will depend upon accessibility to the duct system, the fuel supply, the chimney, and the electrical supply. Too often, a heating or cooling unit is located by giving consideration to installation only, without considering the usability of the basement. This may result in breaking up the basement area to such an extent that it becomes practically useless for other purposes. In a remote type combination unit, consideration must be given to the location of the condensing unit, whether it is located in the basement area or whether the unit is located in a remote area such as the garage breezeway. Preferences of the family and their habits of living must be given consideration so as to keep all space as useful as possible and still locate the equipment in such a place that it can do its intended job and be accessible.

NOISE

One of the prime problems with residential air-conditioning installations is the operating noise level. Almost any degree of quietness in an installation can be had for a price. However, the numerical standards for describing sound levels have little meaning for the average person. It is difficult to ascertain from the prospective user in advance of installation just what sound levels will be acceptable and to describe what sound levels will result from any particular arrangement.

Motors and motor drives, compressor, blower, pumps if

any, and flowing air and liquids are potential sources of noise and vibration. The first, and frequently the least expensive and most effective measure in noise reduction, is to put distance between the noise and the listener. Installations of cooling equipment in basements, crawl spaces, utility rooms or attached garages are preferable to those in alcoves or closets adjacent to bedrooms and living rooms. Mounting of units on concrete floors is preferred. Further steps which may be taken to achieve an acceptable low noise level are as follows:

1. Minimize the production of noise and vibration. Use only true, concentric pulleys with smooth belts at the proper tension and use properly balanced blower wheels and other rotating parts, operated at the minimum practical speed. Provide an adequate duct system which promotes smooth air flow at reasonable velocity and is sufficiently rigid in construction so that it does not vibrate unduly.

2. Prevent the transmission of noise and vibration to the house structure and to ducts and piping. Use properly loaded isolation mounts for compressors, even when on concrete floors, and provide vibration isolators on blowers and refrigerant lines and vibration loops in the refrigerant system. Install flexible connections between the cooling unit and the ductwork.

3. Control airborne noise with sound-absorbing material or other acoustical treatment. Install sound-absorbing material within ductwork and, where practicable, use one or more elbows between the equipment and the registers or grilles.

AIR DISTRIBUTION METHODS

Systems should be designed and installed by competent engineers in accordance with authoritative information such as given in the design manuals of the Air-Conditioning and Refrigeration Institute[6] and the National Warm Air Heating and Air Conditioning Association.[4] The sizes of the ducts of year-round air-conditioning systems are often larger than those of equivalent winter heating systems since large quantities of cool air, at 55 F to 65 F, must be circulated through them.

Several manufacturers have developed reliable methods of duct design. Such methods should be used when recommended, since they take into account the specific air-flow and static pressure characteristics of the systems to which they apply. They should, however, be applied with caution to systems and equipment other than those for which they were developed.

Cooled air admitted to rooms from ducts through supply outlets is from 15 F to 25 F colder, and therefore heavier, than the air in the room, and has a natural tendency to drop toward the floor. Unless the supply outlets are properly selected, located, and adjusted, the air distribution is likely to result in a low velocity of cool air falling on the occupants or a high velocity stream of cool air striking a wall or other obstruction and deflecting on the occupants. A further complication is introduced when the same outlets are used for heating in which case the delivered air at temperatures considerably above room temperature has a tendency to rise.

Where cooling is installed with ducts and air outlets that serve the cooling function only, the practice and experience of comfort air conditioning for offices and commercial buildings apply directly to residential installation. Although cooling and heating by means of the same ductwork and air outlets has been common practice in commercial and industrial applications, greater care must be taken in residential applications. It has been found that low air-flow rates (much lower than would be used for cooling systems) are desirable for warm air heating systems.[5] It is therefore recommended that, in a residence located in a moderate or cold climate, provision be made for different air-flow rates for summer and winter. In warmer climates, however, where air conditioning is the primary consideration, the summer air-flow rate may be used the year around.

The registers may be placed either in the high side-wall or in the perimeter location. For the high side-wall applications, it is necessary that the deflection of the air be different for summer than it is for winter to prevent the jet from dropping into the living zone (60 in. above the floor and below) where the velocity should be not greater than 35 fpm. For registers located at the perimeter of the residence,[5] it is not necessary to change the deflection from summer to winter.

The air distribution system should be designed for the largest air-flow rate that is to be used, as outlined in *Manual* 9 of the National Warm Air Heating and Air Conditioning Association. For small-pipe perimeter systems, it is necessary to have more outlets than for the larger duct systems. In some applications, it may be necessary to close off some registers during the winter so that adequate register air velocities will be obtained. These registers would then be opened during the summer to give a sufficient number of outlets.

Recent investigations have shown[7] that the use of perimeter-duct systems for air conditioning gives excellent results, and that objectionable air motion does not occur within the living zone except in the immediate vicinity of the registers. Care must be taken, however, to use only registers or diffusers that are specifically designed for perimeter application. This is necessary to permit the introduction of the air into the room in a pattern that does not cause drafts. Floor diffusers which distribute the air in a fan-shaped pattern along the wall or low side-wall diffusers which blanket the wall are recommended. For supply-air temperatures not more than 15 deg below room-air temperatures, diffuser face velocities of 500 fpm are recommended.[5] For supply-air temperatures 20 deg below room-air temperature, the diffuser face velocity should not be less than 700 fpm.

Many registers designed for outside wall application may not be satisfactory for cooling when applied to low-velocity systems because the cool air is apt to remain at the floor.

Ceiling diffusers may also be used, but they may lead to difficulty during the heating season if the warm air remains at the ceiling, resulting in cold floors.

High return locations at inside walls are very effective with outside wall or floor perimeter distribution systems for both the heating and cooling season. In multi-story or split-level dwellings there should be a minimum of one return on each level with a high central return on the upper level to minimize the effects of hot air rising through halls and stairways to upper levels. With ceiling or high-inside-wall distribution, the returns should be located on the outside walls under windows for best results.

CONTROLS

Control devices and systems and their application in residential air conditioning are discussed in Chapter 43, Automatic Control.

OPERATING COSTS

The operating cost of residential cooling equipment during a particular summer depends on variables such as the amounts of sunshine and rain, the number of abnormally hot or cool days, the efficiency of the equipment, and the local power rate. It is also influenced by human factors such as operation of equipment only during the hottest weather, opening windows at night,[5] and difference in preferred indoor

Table 1 . . . Approximate kw Input per 12,000 Btuh Cooling Capacity

Item of Power Load	Type of Heat Rejection[a] Water Cooled Condenser			
	City Water	Atmospheric Cooling Tower	Mechanical Draft Cooling Tower	Air-Cooled Condenser
Refrigeration Compressor	1.00	1.00	1.00	1.35
Blower for Air Conditioner	0.10	0.10	0.10	0.10
Blower for Rejection Air	—	—	0.15	0.15
Pump, Cooling Water	—	0.15	0.15	—
Total kw per 12,000 Btuh	1.10	1.25	1.40	1.60

[a] Other heat rejection means include wells, spray ponds, evaporative condensers, water coils in the ground, finned water coils in air, etc.

temperatures.[16] Nevertheless, it is important that lending agencies and prospective buyers of equipment be given a reasonably accurate estimate of the operating cost during normal summer weather and under usual operating conditions. Adjustments can then be made for any special conditions anticipated. The approximate electrical power inputs for the various motorized components in mechanical cycle air conditioners are shown in Table 1.

Power cost per hour can be estimated by multiplying the estimated power per ton, the cooling capacity in tons and the cost per kilowatt-hour. Thus the estimated cost per hour for a 3-ton air conditioner cooled with city water will be 1.1 kw × 3 tons × the power rate. It is essential to use the correct step of the utility residential rate structure to get a good estimate. The basis of the method used[16] requires the use of a table of the estimated annual hours of operation for properly sized equipment in typical cities. See Table 2.

The values in Table 2 have been substantiated by utility records of actual performance in several major cities when the average indoor temperature is maintained at 80 F. Low indoor temperatures increase operating cost as do weather conditions above normal.

Water usage is another important factor with water-cooled equipment. Various manufacturers have published water usage data for their equipment at varying summer water temperatures. Representative water consumption values are given in Table 3.

For a given application the power rate and water rate may be obtained from the local utilities. Since the average power requirement depends on the particular equipment used and

Table 2 . . . . Estimated Annual Hours of Operation for Properly Sized Equipment in Typical Cities during Normal Cooling Season

City	Hours	City	Hours
Atlanta, Ga.	750	Jacksonville, Fla.	1600
Boston, Mass.	200	Minneapolis, Minn.	350
Chicago, Ill.	400	New Orleans, La.	1500
Cleveland, Ohio	450	New York, N. Y.	350
Dallas, Texas	1400	St. Louis, Mo.	1000
Fresno, Calif.	900	Washington, D. C.	800

Table 3 . . . . Water Usage for Water-Cooled Equipment

Summer Water Temperature, Normal Extremes F	Gallons Per (Hour) (Ton)	Summer Water Temperature, Normal Extremes F	Gallons Per (Hour) (Ton)
55	34	75	58
60	39	80	70
65	44	85	88
70	50		

the condensing method, values obtained from the manufacturer should be used in preference to those in Table 1.

If the evaporator fan cycles with the compressor, the power requirement as found from Table 1 also represents the total power requirement for air-cooled condenser applications. For continuous fan operation the number of hours of compressor operation should be subtracted from the number of hours in the cooling season to obtain the extra hours of fan operation. This value may then be used with the appropriate value of power consumption and power cost for the fan motor only, to find the added power cost for continuous fan operation and the result should then be added to the power cost during compressor operation.

For water-cooled condenser applications the water cost, as determined from the operating hours, water rate, and manufacturer's data on equipment water usage, is added to the power cost to obtain the estimated total operating cost.

Another method has been developed using cooling degree days above 70 F as a criterion but tabulated values of cooling degree days for various localities are not available.[13] Approximate values can be obtained from degree-day maps.[14, 15] The daily range of temperatures for various localities is also required with this method.[16] It is important that the method used in estimating operating cost be consistent with the method used in calculating the heat gain of the structure.

EFFECTS ON FUTURE HOUSE DESIGN

Air conditioning, like automatic heating, has brought about improved practice in residential construction, particularly in those regions where cooling is of primary importance. Although it is true that construction practices which reduce the heating load also reduce the cooling load, there are certain considerations that are unique in their effect upon the cooling load. Special attention should be paid to window orientation and shading. South windows may be shaded by awnings or roof overhangs, but it is not practicable to shade large window areas having an east or west orientation except by exterior solar screening of some type. The use of insulation in the walls and ceiling of the residence will reduce the cooling load appreciably. In addition, attention should be paid to adequate ventilation of the attic. Double-glazed and weather-stripped windows, heat-absorbing glass, planned shade tree locations, and light-colored exterior wall and roof surfaces, also help to reduce the cooling load.

REFERENCES

[1] T. N. Willcox, C. T. Oergel, S. G. Reque, C. M. toeLaer, and W. R. Brisken: Analogue computer analysis of residential cooling loads (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 505).

[5] H. T. Gilkey, D. R. Bahnfleth, and R. W. Roose: Cooling a small residence with a two-horsepower mechanical condensing unit (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 283).

[4] *Application Engineering Standard for Year-Round Residential Air Conditioning* (Air Conditioning and Refrigeration Institute, 610-56).

[5] *Summer Air Conditioning* (National Warm Air Heating and Air Conditioning Association, *Manual* No. 11, 1955).

[6] *Continuous Air Circulation* (National Warm Air Heating and Air Conditioning Association, *Manual* No. 6, 1955).

[7] M. E. Childs, R. W. Roose, H. T. Gilkey, and S. Konzo: *Comparative Performances of Two Warm Air Perimeter Systems and Three Convection Systems* (University of Illinois, *Engineering Experiment Station Bulletin* 403, 1942).

[8] D. R. Bahnfleth, C. F. Chen, and H. T. Gilkey: Cooling a small residence using a perimeter-loop duct system (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 271).

[9] H. E. Straub and S. F. Gilman: Room air distribution research for year-round air conditioning, Part II—Supply outlets at three floor locations (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 249).

[10] A. P. Kratz, M. K. Fahnestock, and S. Konzo: *Investigation of Summer Cooling in the Warm Air Heating Research Residence* (University of Illinois, *Engineering Experiment Station Bulletin* 290, 1930).

[11] R. A. Gonzales: How power usage varies for summer air conditioning of identical residences (*Refrigerating Engineering*, January 1957, p. 35).

[12] T. N. Willcox: Annual utility cost operating residential cooling system (*Fuel Oil News*, June 1954).

[13] S. F. Gilman, L. A. Hall, and E. P. Palmatier: The operating cost of residential cooling equipment (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 525).

[14] S. S. Visher: Relative cooling requirements for American homes (*The Scientific Monthly*, November 1945, p. 211).

[15] E. N. Kemler and S. Oglesby: *Heat Pump Applications* (McGraw-Hill Book Co., New York, 1950, 1st ed., p. 246).

[16] *Summer Weather Data* (The Marley Co., Kansas City, Kansas, 1939, Chapter VI).

CHAPTER 47

SCHOOL SYSTEMS

General Considerations, Criteria for Design, Design Considerations, Preliminary Design, Mechanical Plant Design, Types of Ventilating Systems, Types of Heating and Cooling Systems, Automatic Control, Operation, Maintenance

The purpose of this chapter is to outline the factors affecting the selection of heating, ventilating, and air-conditioning systems for schools. The components of these systems are the same as used for other types of buildings. Their combination in school systems, in some cases, will be the same as for other buildings, but may differ considerably in others in order to provide optimum conditions for effective school work and to comply with local regulations.

GENERAL CONSIDERATIONS

The proper criteria for heating, ventilating, and in many instances air conditioning, of school buildings, particularly classrooms, are often not recognized fully.

Many designers and school administrators approach the problem as one of heating, whereas analysis may show that cooling, rather than heating, is required during periods of occupancy throughout most of the year. Ventilation systems are frequently designed to supply air at 72 F for ventilation and, when feasible, at 55 F to 60 F to provide the cooling required during occupancy.

With regard to ventilation, code requirements vary from acceptance of natural ventilation, such as obtained by opening of windows, to a requirement of mechanical supply of a specified quantity of air, which sometimes is as high as 30 cfm of outdoor air per occupant. The variations in quantity of outdoor air specified are due to the basis adopted whether it is the quantity needed *to replace oxygen for breathing* or that required for the *dilution of odors*. A discussion of air requirements will be found in Chapter 6.

When planning for introduction of ventilation air into rooms, careful consideration should be given to air distribution and circulation within the occupied space, as well as the natural convection and temperature effects.

Elementary schools of recent design are generally single-story buildings, often placed at considerable distance from each other. They are often provided with windows 7 ft or more in height extending the full length of one or more classroom walls. These large glass areas constitute one of the *greatest problems in maintaining an optimum indoor* atmospheric environment.

Secondary or high schools are more often two- and three-story structures but present problems only slightly less severe than grade schools. Separation of school buildings results in increased cost of the mechanical systems.

A further characteristic of school buildings is a wide variation in use and occupancy. Classrooms (the largest single group of occupancies), lecture rooms, assembly rooms, auditoriums, and other similar areas have uniformly dense occupancies. Gymnasiums, cafeterias, laboratories, shops, and similar functional spaces have lighter occupancies and also have different and varying ventilation temperature requirements. Many schools include multipurpose rooms serving as auditoriums, gymnasiums, and community centers.

The normal periods of operation are five days per week for nine months per year, with complete shutdown during week-ends and vacation periods. Few schools operate during the summer months and therefore do not require complete air conditioning, except in southern climates. Systems for administration areas, gymnasiums, auditoriums, and multi-purpose rooms, *suitable for community activities*, will often be operated at times when the systems for classrooms and other general educational areas may not be in operation.

Automatic control of temperature and system equipment is a necessity if comfortable conditions are to be obtained. Proper application of controls may return the cost of investment by savings in labor and fuel within a year or two.

Frequently, insufficient consideration is given to provisions for access and maintenance with the idea that the mechanical system is a fixed part of the structure and should last as long as the structure, without attention.

School boards are often criticized by their constituents for spending large sums of money for so-called luxury items, with the result that economy in construction and design becomes a paramount factor. This often tends to limit size of the mechanical equipment selected below that required to maintain a proper indoor atmospheric environment. However, many schools in areas having warm spring and fall climates are now being designed to include air conditioning, at least in spaces of densest occupancy and greatest use.

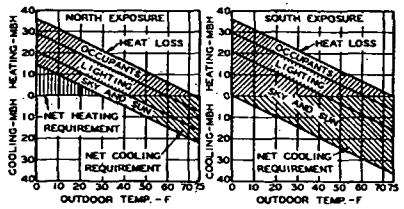
CRITERIA FOR DESIGN

Comfort, cleanliness, reduction of odors, maintenance of alertness, compliance with codes, installation cost, ease and cost of upkeep, cost of operation, and adaptability to control in relation to periods of use as related to outdoor conditions are the chief criteria by which a design for maintenance of indoor environment for learning may be judged.

Comfort, which may be defined as freedom from strain in accommodating to the environment, is a necessity. Uniformity of air movement and temperature within the space is required to provide a comfortable environment. Fundamentals in this respect are discussed in Chapter 6, Physiological Principles, and Chapter 20, Air Distribution. Intermittent currents of air or air movement at varying velocities may result in discomfort. Similarly, variations in temperature may contribute more to discomfort than uniform maintenance of slightly higher or lower temperatures.

Odors, while they may not be deleterious to health, are disagreeable and distracting, even to the point of being nauseating. Odors should not be perceptible in schoolrooms.

Local codes may establish requirements for construction, glass areas, ventilation rates, etc., which exceed values used in normal design practice.



Basis: Floor area—1000 sq ft. Windows on one side only—240 sq ft. Heat transmission factors, U; roof—0.15, wall—0.25, glass—1.13. No infiltration. Heat gains based on artificial lighting at 2.3 watts per sq ft of floor area. Occupants, 35. Solar intensity and incidence at 45 deg N. latitude, December 1. Windows shaded by venetian blinds. Since the consideration is cooling, a room temperature of 75 F, instead of the 72 F shown in Table 1, was used in the above charts. Net cooling requirement for the south exposure approaches that of the north exposure on a day when the sky has a heavy cloud cover.

Fig. 1 . . . . Typical Net Heating and Cooling Requirements of Classrooms

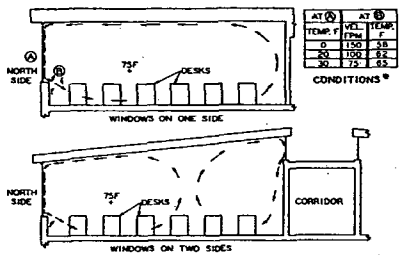
Installation cost is a prime consideration in design. It should be considered in relation to the overall cost of school operation. The cost of maintenance is reflected in man hours required to provide proper operation and reasonable life of the equipment. Proper location, access, and working space for the equipment are reflected in reduced operator or custodian time. Some systems require more frequent attention, overhaul, or replacement than others. Operating costs may be held to a minimum by: (1) selection of the proper fuels for system requirements based on fuel economics in a particular area; (2) providing flexibility for operation of sections of systems according to use; (3) provision of adequate automatic temperature control and system control.

Periods of use contemplated for the entire school building, or sections of it, due to summer school, or community use of an auditorium throughout the summer or during spring and fall months (in a region where these are mild) may influence a decision to include air-conditioning systems.

DESIGN CONSIDERATIONS

Design for the required indoor environment for learning must be made in consideration of the heat losses, heat gains, and air movements; ventilation means and ventilation rates; space temperature requirements; variability of occupant density; independent operation of various spaces; practicability of cooling; means of cooling (if it is to be included in the design); and automatic control of the environment.

A typical illustration of heat losses and heat gains in classrooms is presented in Fig. 1, applicable for sun effect to about 45 deg north latitude and design temperature for heating of 0 F. For the classrooms shown and based on the construction and conditions described in the footnote, heat gains in the classroom with north exposure equal the heat losses at 32 F. In the south classroom, practically no heating is required after the room is occupied and lighted. Accordingly, heat is required in the south classroom only for warm-up. Cooling is required in north classrooms at outdoor temperatures above 32 F and, in south classrooms, at all temperatures above 0 F. Conditions shown for the north classroom would apply to a south classroom when the sky is fully overcast.



** Conditions shown at windows are conservative; vary with number and shape of cross-members. Temperatures are often lower; velocities often higher.*

Fig. 2 . . . . Natural Air Currents in Classrooms

Natural air movements or air currents prevailing in the space must be taken into account in design. Fig. 2 illustrates two conditions: the first having windows located along one side, the second having windows for cross-lighting. Measured conditions are shown for classrooms of the first type. It is significant, as will be noted in these illustrations, that, even though the room temperature at the control point may be at the desired level, uncomfortable conditions could be expected for the students in the first or second rows paralleling the windows. Direct radiation installed at the glass exposures cannot be fully effective in warming the descending currents of cold air unless it extends the full length of the windows, is located close to the window sill, and is in operation whenever the outdoor temperature is low enough to cause these downdrafts.

Ventilation means and rates are largely fixed by state and municipal codes and vary considerably with type and intended use of any building. Applicable ventilating systems are shown in Table 1. Natural ventilating systems are those relying on opening of windows to admit and exhaust air. Mechanical supply or exhaust systems are those using fans to supply or exhaust air. Natural ventilating systems are not suitable for spaces such as gymnasiums, auditoriums, cafeterias, and meeting rooms, because of size or density of occupancy; nor for laboratories, large toilet rooms, locker rooms, or kitchens in which intense odors are common; nor in shower rooms where high vapor concentrations are likely to occur. It is not possible to obtain consistently good room conditions in classrooms by means of any ventilating system that depends on openable windows for introduction of air. This is the case with each of the natural ventilation systems shown in Table 1. The problems are due to the following factors:

1. At low outdoor temperatures, the windows are kept closed, resulting in high odor intensities.
2. As outdoor temperatures rise, up to temperatures of 60 F, uncontrolled, uncontrollable, and variable drafts will occur in the occupied zone. Typical conditions are shown in Fig. 3.
3. Frequent adjustment of window openings by the instructor, with consequent distraction from teaching duties, will be necessary to provide adequate ventilation without excessive drafts and to maintain proper room temperature.
4. Manual readjustment of window openings is an erratic method of controlling ventilation.
5. The advantages of saving time and obtaining greater accuracy of temperature control are lost.

Table 1 . . . . Applicable Ventilating Systems and Room Temperatures

Space	Natural	Natural with Gravity Exhaust	Natural with Mechanical Exhaust	Mechanical Supply and Exhaust	Mechanical Supply, Gravity Exhaust	Room Temp F
1. Classroom	No[a]	No[a]	No[a]	Yes	Yes	72
2. Gymnasium	No	No	No	Yes	Yes	60–70
3. Auditorium	No	No	No	Yes	Yes	72
4. Kitchen	No	No	No	Yes	No	65–70
5. Cafeteria	No	If Code[b, a]	If Code[a]	Yes	Yes	65–72
6. Meeting Rooms	No	No	If Code[a]	Yes	Yes	72
7. Offices	Yes	Yes	Yes	Yes	Yes	72
8. Locker Rooms	No	Yes[c]	Yes[c]	Yes	No	75–80
9. Laboratories	No	No[a]	No[a]	Yes	No	72
10. Shops	If Code[c]	If Code[a]	If Code[a]	Yes	Yes	65–70
11. Toilet Rooms	Yes[d]	Yes[c]	Yes[d]	Yes	No	70
12. Showers	No	No	No	Yes	No	75–80

[a] Subject to conditions. See text. [b] If permitted by code. [c] Unless rooms are interior. [d] If rooms are small and have openable windows.

Temperature requirements of various spaces depend on occupancy, function, degree of activity, and type of clothing. Values generally accepted are listed in Table 1.

In meeting spaces such as auditoriums, cafeterias, gymnasiums, and the like, the variability of occupant density may warrant the use of separate units or systems for the several spaces in order to permit their individual operation without operating the entire school system.

It may be practicable to counteract the net heat gains shown in Fig. 1 or some of them by cooling with ventilation air or other means besides refrigeration. This would require the use of mechanical air supply with sufficient cooling capacity at a 55 F outdoor air temperature to offset the net heat gains. If the amount of air required results in an excessive cost of the mechanical system, a study should be made of means of reducing the various cooling load sources, particularly the solar load. When outdoor temperatures are in excess of 65 F, windows with ample opening to produce rapid changes of air in the space are necessary to maintain comfortable conditions, unless cooling by refrigeration is installed. When the outdoor temperature is above 65 F, relatively high velocity air movement in the space is not objectionable from the standpoint of comfort. Even under the most favorable conditions, ventilation systems cannot maintain comfort when the outdoor temperature exceeds 72 F.

DESIGN OF THE MECHANICAL PLANT

Heating, ventilating, and cooling loads are computed according to the procedures given in Chapters 9, 11, 12, and 13. The performance requirements of the various parts of the system, whether combined or operating separately, according to use, are then established, with due consideration being given to the method of control and operation of the entire school project. Heating requirements should be determined for design conditions and for warm-up periods, the heaviest warm-up load being that which occurs after a weekend shutdown when the outdoor temperature is at design value.

The load applicable during occupancy, considering net heating and cooling loads, is determined and evaluated as shown in Fig. 1. If planned use of the building extends through the summer, cooling design dry-bulb temperatures used are those established by the ASHAE Technical Advisory Committee on Weather Design Conditions which are shown in Column 6 of Table 2, Chapter 13, and referred to as "TAC 2½% Basis." Design temperatures in common use, both dry- and wet-bulb, may also be found in the same table in Columns 7 and 8. For detailed analysis of frequency of occurrence of extreme conditions by months, reference may be made to *Region Climate Analyses and Design Data*, published in 1951 by the American Institute of Architects. These analyses are available for all United States areas.

The requirements of each of the mechanical systems incorporated in a school plant are as follows:

1. An adequate warm-up rate for heating, the ability to raise room temperatures to required conditions shown in Table 1 within an acceptable interval of time when the outdoor temperature is at design value. Usually, warm-up requirements after a weekend shutdown are used as a basis.
2. Maintenance of indoor design temperature when cooling is required. Considering the heat gains shown in Fig. 1, a typical analysis of ability to provide cooling with outdoor air is shown in Fig. 4.
3. Uniformity of air motion, discussed under Criteria for Design. Generally this requirement is satisfied by a system providing constant air flow, with proper distribution of the air supplied.
4. Uniformity of temperature of supply air, discussed under Criteria for Design.
5. Introduction of ventilation air, according to applicable codes; otherwise on a rational basis, as discussed in Chapter 6 for quantity and Chapter 20 for distribution.
6. Interception of deflection of window drafts in rooms having extensive fenestration, such as classrooms. (See Fig. 2.)
7. Acceptable noise levels which will depend on location of the school and on the equipment provided.

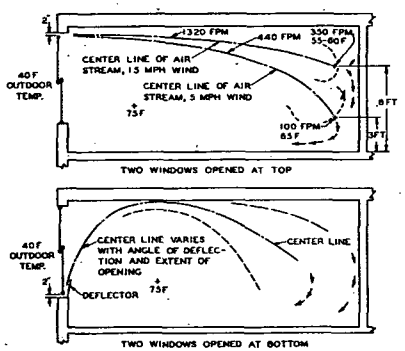
TYPES OF VENTILATING SYSTEMS

Any ventilating system provides air circulation and thus includes means of supply and exhaust. Ventilation systems may be classified by the type of supply (window or mechanical) or exhaust (window, gravity, or mechanical).

Air Supply Methods

Windows are generally of double hung type or consist of metal architectural sections, the bottom section being arranged to swing inward to deflect air upward and the top section to swing outward from a hinge at its top or center. If double hung windows are used, a deflecting device, generally of glass, to prevent interference with light transmission, is provided at the bottom to deflect incoming currents upward, as illustrated in Fig. 3, and thereby prevent them from blowing directly on room occupants at desk level. Since windows require manual opening, proper control of air currents cannot be provided.

Mechanical air supply is generally provided by a central



Wind[a] Mph	Air Volume Cfm	Cooling in MBh			
		40 F Outdoor Temp.		55 F Outdoor Temp.	
		Required[b]	Obtained	Required[b]	Obtained
0	0	19.0	0	27	0
5	730	19.0	25.3	27	14.5
15	2190	19.0	69.0	27	39.5

[a] Velocity vector normal to window. [b] South Classroom, Fig. 1.

Based on wind normal to exposure; room 25 ft deep, windows double-hung, 5 ft wide, 2 windows opened as shown.

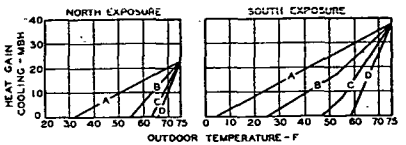
Fig. 3 . . . . Typical Air Flow in Classrooms with Natural Ventilation

duct system, the air being delivered through wall grilles or ceiling diffusers, or by self-contained units within the rooms, such as unit ventilators and unit conditioners located at the outside walls. Grilles are generally located most conveniently at the inside or corridor wall. They must be of the deflecting type, deflecting upward in consideration of the drop in the air stream when air is introduced at lower than room temperature. Diffusers are generally located in ceilings of rooms in a pattern to provide proper air distribution. In location of both wall grilles and diffusers, it is important that the air-delivery path be free of obstructions, which might be light fixtures and crossbeams, since presence of obstacles will result in uncontrollable and turbulent currents of air extending into the occupied zone and causing drafts. Grilles and diffusers are generally selected to provide a terminal velocity of 50 fpm at approximately one-fourth the distance from an opposite wall or other obstacle to air flow.

Exhaust Methods

Windows, when used for exhaust, are opened at the top to take advantage of natural rise in the warm air to be exhausted. Windows are not fully effective for exhaust purposes, except in completely still weather. Typical conditions are shown in Fig. 3, indicating that, if the windows are on the windward side, air is introduced rather than exhausted.

Gravity exhaust ventilation is obtained through exhaust grilles in the ceiling (or in a wall away from the windows)



Basis: Cooling requirements shown in Fig. 1.
A = Net heat gain with ventilation air controlled at room temperature of 75 F.
B, C, and D = Net heat gains using 525, 1000, and 1500 cfm, respectively, controlled to 55 F, minimum. Cooling air temperature will be higher as outdoor air temperature rises above 55 F.

Fig. 4 . . . Typical Cooling Capacities, Using Outdoor Air

connected to roof ventilators. Air is exhausted by wind effect and chimney effect due to differential pressure resulting from thermal head. This type of exhaust is used in conjunction with air supply obtained by admitting air at the windows. Satisfactory air changes can be obtained by this method within limits shown in Fig. 3, if the room ventilators are adequately large. However, room conditions cannot be fully acceptable, since objectionable cold drafts are certain to occur in cold weather. Control of conditions depends on manual adjustment of window openings, a method that is reasonably satisfactory only in mild climates, but is not recommended where the design temperature is lower than 30 F. Gravity exhaust ventilation may be used with satisfactory results in combination with mechanical air supply systems.

In *mechanical exhaust systems* the roof ventilators described in preceding paragraph on *gravity exhaust ventilation* are replaced with motor-driven exhaust fans. If this system is used in conjunction with *window supply of air*, it should be remembered that when windows are closed, replacement air is drawn from the corridor or the ventilation rate is greatly reduced. At such times, negative pressures will prevail in the building, with adverse effects on incinerators, hooded equipment, and other local exhaust and fuel-firing equipment.

TYPES OF HEATING AND COOLING SYSTEMS

Systems may be classified according to methods of heating and cooling. Each may be combined with means for ventilation in various ways. Only a few states have specific requirements with respect to types of systems. Accordingly, except in these states, engineers and administrators of school construction are free to analyze and choose kinds of systems. The possible combination of such systems are extremely numerous. A discussion of those encountered most often will follow.

Direct Radiation

This type of heating includes cast-iron radiators, cast-iron or nonferrous convectors, and continuous finned-pipe heating elements in connection with steam or water heating systems (see Chapters 14, 26, and 28). The radiation must extend the full length of the outside or window walls and the full width of high windows used in cross-lighted designs illustrated in Fig. 2. Control is obtained by manual valves, by individual automatic control valves for each room, or by an automatic valve or valves controlling zones of the system according to exposure.

Direct radiation may be used with natural ventilation, natural ventilation with gravity exhaust, and natural ventilation with mechanical exhaust. It is designed for the warm-up load and to provide a warm curtain of rising air to mix with the incoming cold air to temper it. However, it is inadequate for this purpose, because heating requirements are satisfied at relatively low outdoor temperatures as soon as classrooms are occupied and lighted, as illustrated in Fig. 1. The system requires the teacher to adjust openings for ventilation. Installation costs can be relatively low.

Electric Heating

This consists of electrical heating elements in baseboard type of radiation extending the full length of the window walls, electric ceiling panels, or panels formed of heating cable buried in the ceiling plaster. Control of heating is automatic by means of individual room thermostats. Any of the ventilating systems discussed previously can be combined with electrical heating, much as discussed under Direct Radiation. Electrical heating has the same shortcomings as direct radiation; namely, that when classrooms are occupied and lighted, heat is not available for counteracting downdrafts at windows. The system requires the teacher to adjust openings for ventilation. The cost of energy will depend on the local electrical rates.

Panel Heating

Floor panel heating is adaptable to the slab-on-ground type and to the open-corridor type of school, particularly in the warmer climates. Design data are given in Chapter 30, Panel Heating, which also describes features of panel heating control systems. (See also Chapter 43, Automatic Control.) The floor system lends itself well to individual room control, although zone control, including several rooms of the same exposure in the same zone, is practicable. *Floor panel heating systems* are adaptable in areas having design temperatures down to 0 F. Since the requirement for heating is materially reduced when the spaces are occupied and lighted, as illustrated in Fig. 1, the panels should be constructed to have minimum mass and thus minimum storage effect. This is accomplished by using a slab, only sufficiently thick to enclose the tubing or piping, underlaid by a slab of insulating concrete. Even with light concrete panels, there is enough storage effect to cause overheating after warm-up and to make it difficult to follow rapidly changing loads. Any of the types of ventilating systems already discussed may be combined with panel heating systems. However, the panel heating system cannot be effective in combating the natural condition of downdrafts at the windows. The system requires the teacher to adjust openings for ventilation. The panel system is well adapted for use in primary classrooms with slab-on-ground floor construction, where students spend much time on the floor.

Central Indirect Air Systems

In these systems, air is supplied for ventilation and heating. At a central point or points, outdoor ventilation air and recirculation air is mixed and heated by automatically controlled steam or hot water coils. Gravity or mechanical exhaust is coordinated with the air supply system. The central system can be arranged to recirculate air from window sills to intercept the flow of uncontrolled cold window drafts into the classroom. Such returns are fully effective only if the openings are located in the window sills; if located below the sills, the natural air currents will not be intercepted. There are many variations of the system, which may include features of central systems described in Chapter 19. Air supply may be controlled by zones of similar occupancy and identical orientation. Individually controlled booster or reheat coils may be located within ducts leading to each of the spaces. Individual room control may also be provided by a double duct system, equipped with mixing dampers for each room located within a short distance ahead of the supply grilles or diffusers serving the room. Central air conditioning can be included.

Split Systems

The central indirect air systems may be combined with direct radiation or convection units described in the previous section, Direct Radiation. In this combination, air may be supplied by the ventilating system at constant temperature, usually 55 F, when the outdoor temperature is less than 55 F. In this case direct radiation may be designed to take care of the building heat loss. In order to prevent window drafts, the temperature of the ventilation air should be controlled to require operation of the direct radiation when the outdoor temperature is below 55 F. The direct radiation is controlled by automatic valves operated by a room thermostat for the individual room. This system has wide flexibility.

Central Direct-fired Air Systems

These systems are forced circulation warm air furnace systems. They may be arranged for single or double duct air distribution, as discussed under Central Indirect Air Systems. This type of system is further discussed in Chapter 18, Warm Air Heating. They are usually installed with zone controls and arranged to permit use of any desired proportion of outdoor air during periods of space occupancy. Mechanical cooling may be obtained by installation of cooling coils in the main supply ducts.

Unit Ventilators

Unit ventilators are used in classrooms to provide heating, controlled ventilation, and cooling, using outdoor air as the cooling medium. Auditorium unit ventilators are used in large rooms such as auditoriums, gymnasiums, and large lecture rooms. Classroom unit ventilators are often combined with matching storage cabinets or may be combined with ducts extending the full length of the windows and arranged either to discharge air upward along the window sills or withdraw air at the window sills and thereby prevent downdrafts from the windows. In order to effectively intercept natural downdrafts, the return grille must be located with the top of the unit at the level of the window sill. Classroom unit ventilators may also be combined with extended finned-pipe radiation at the window sills to combat cold window downdrafts. Electrical unit ventilators are also available. Unit ventilators using hot water for heating and chilled water for cooling are used for school buildings where year-round use is contemplated or where warm fall or spring seasons warrant the installation of air-conditioning equipment. These units and their controls are described in Chapter 15, Unit Ventilators and Unit Heaters. The noise level of the unit ventilators under the operating conditions of the installation should be considered. These systems have a high degree of flexibility.

Unit Conditioners

Unit conditioners are essentially the same as unit ventilators, discussed above, except that the cabinets contain refrigeration equipment, a direct-expansion coil, and a steam or hot water coil piped from the central boiler source. Noise

levels of unit conditioners are generally somewhat greater than those of unit ventilators. Unit conditioners are described in Chapter 16, Unitary Air-Conditioning Equipment.

AUTOMATIC CONTROL

Discussion of automatic control in this chapter will be limited to general factors peculiar to school systems, and their effect on control or the need for it. Chapter 43, Automatic Control, provides a detailed discussion of the proper application of automatic control to the various system elements normally encountered.

Automatic temperature control equipment is justified in school systems because manual control will not maintain the temperature or environment within desirable limits.

Automatic control, made an integral part of the system design, can reduce the *initial cost* of the mechanical installation. Examples of this type of control include *warm-up clocks* or low-limit thermostats, which prevent the use of outdoor air until the building has been heated for a set period of time, or until a minimum space temperature is reached. Such controls permit selection of the minimum size of equipment such as boilers, furnaces, burners, coils, piping, etc., having the least excess capacity for warm-up purpose. Controls to prevent the heating of service water in large storage tanks during the warm-up period accomplish a similar result.

Controls which reduce the man-hours of attendance at the heating plant or reduce fuel costs, can reduce *operating costs*. Time clocks, relays, temperature and pressure control, which allow the starting of the heating plant without the presence of supervising personnel, are included in this category, together with controls to reduce the use of outdoor air during warm-up periods.

Where fans and extended surface coils are used to maintain the temperature in an occupied space having outside exposure, it is not possible to control the capacity of the equipment manually without producing alternately high and low space temperatures because of the rapid changes in load due to shifting occupancy, variation in solar intensity, and wind variation in speed or direction, and the relatively high capacity of this type of equipment. Where occupancy is particularly dense, the changes in load due to changes in occupancy alone are enough to make proper automatic control necessary in order to maintain space temperatures within acceptable limits. This applies to a large percentage of the spaces in a school building.

There are many other factors or conditions that make proper automatic control necessary in order to achieve acceptable environmental conditions. Those already discussed illustrate some basic considerations in applying temperature control to mechanical systems for schools.

OPERATION

The proper method of operation of a system depends upon its design and the automatic control applied to it and will vary with each individual system. To insure good operation throughout the life of a system, it is essential that adequate written instructions be prepared and made readily available to the operating personnel. There should be at least two copies of the instructions, one of which should be placed in a secure depository to insure its preservation in good condition during the life of the building.

The operating personnel should be thoroughly instructed regarding proper operation of the equipment upon completion of the installation. Someone having authority in the school administration, in addition to the operating personnel, should be instructed in the operating characteristics of the system, not only to insure that a change in the operating personnel does not result in the loss of knowledge required for the proper operation of the system, but also to insure that the operating personnel adheres to the instructions.

It is seldom that the personnel in charge of the operation of a school system has training particularly applicable to the characteristics of mechanical equipment or the automatic control applied to it. Therefore, it is essential that this type of operating personnel be made to understand the exact limits that must be observed in adjustment and operation of the system. Where temperature control is applied to a system, it must be realized that any instrument used for control purpose is of necessity very sensitive to any change or adjustment made in the setting of the instrument. Therefore, it is necessary that only personnel fully trained in the proper handling of the instruments should make adjustments other than those expressly designated as proper for the operating personnel.

It is common practice for officials in large school districts where there are many buildings to employ specialists acquainted with the adjustment of critical items of equipment such as temperature and automatic control. Where individual schools do not have such personnel available, it is necessary that the operating personnel be instructed to communicate with qualified service agencies to make changes in automatic control settings and instrument characteristics when they are found to be necessary by experience in the operation of the system. Continuing service contracts for proper maintenance of equipment can be had from companies in this field.

When systems are installed and designed on the basis of natural or gravity ventilation, the teacher or person in charge of the individual room or space must perform the functions of adjusting window openings or ventilator settings to provide the conditions of comfort and proper environment in the space. To accomplish efficient operation of the system and maintain a suitable environment, it is necessary that these people be instructed in detail regarding the techniques necessary to achieve the proper environment. It is necessary that the administrative authorities supervise the maintenance of such conditions by periodic checks on the efficiency of the people responsible for the spaces.

MAINTENANCE OF MECHANICAL SYSTEMS

The proper maintenance of a mechanical system will result in a minimum of repair and the extended life of the system equipment. As is the case with operating instructions, it is essential that written instructions for the maintenance of the system be prepared and given to the owner when the system is completed and placed in his charge. These instructions should include detailed information on the lubrication and care of each item of equipment. A schedule showing items of maintenance to be performed for each month of the year should be prepared.

Name plates and identification for each valve and item of equipment should be prepared. Manufacturers' catalogs and parts lists for the equipment should be obtained. One copy of each of these items should be placed in a secure depository, together with the operating instructions.

A list of the proper service agencies should be prepared and placed in the hands of the operating personnel. Any changes in service agencies should be recorded for the benefit of the administration as well as the operating personnel.

CHAPTER 48

TRANSPORTATION AIR CONDITIONING

Railway Passenger Car Air Conditioning; Passenger Bus Air Conditioning; Automobile Air Conditioning; Aircraft Air Conditioning; Ship Air Conditioning, Heating and Ventilating, Air-Conditioned Space Treatment, Systems and Controls

THE principles of air conditioning applying to stores, restaurants, hospitals, theaters, and homes are applicable to railway passenger cars, passenger buses, automobiles, streetcars, trolley coaches, airplanes, and ships. However, equipment used for mobile applications differs from that used for stationary purposes in that it must meet additional requirements. Equipment must be compact, accessible for quick inspection and servicing, light-weight and unaffected by vibration and impact. Freedom from vibration which could be transmitted to supporting vehicle and thus to passengers, is essential.

RAILWAY PASSENGER CAR AIR CONDITIONING

The railway passenger car represents a very difficult air-conditioning problem. Space is strictly limited so that all equipment and ducts must be reduced to minimum size. Electric power supply and water supply also are limited. All equipment must withstand severe vibration and shock, and must be very reliable since servicing points are frequently far apart.

During the heating season it is necessary to heat conventional cars with steam from the locomotive at pressures that may vary from 250 psig to only 5 or 10 psig on the last car in long trains. Passengers in window seats sit only a few inches from cold outside walls and windows, and are close to heating surfaces installed along sides of cars. Sudden changes in load may be caused by changes in sun, wind, or train movement. Even in coldest weather, outside doors must be opened frequently.

During the cooling season, the problem is further complicated by a highly concentrated internal load due to the passengers. Air distribution problems are increased by low ceilings and short air throws.

Heating

The heating of passenger cars is accomplished by using a split system consisting of an overhead air-circulating system with heating and cooling coils, and heating surfaces (floor heat) along car sides. The floor heaters, which usually consist of finned tubing, may be made more effective by addition of covers designed to increase gravity air circulation, and to direct the warm air from finned heating surface along cold outside walls and car windows. In some new cars, wall convector panels are used and extend the full length of the car, with air intakes along the floor and outlets at window sill height and at window head height in dead-light panels. The heated panel protects passengers from cold outside walls, and the chimney effect of the panel duct increases air flow and improves heating surface effectiveness.

Floor heat is supplied by introducing steam into an inner tube within a finned tube or by means of a separate steam-to-liquid heat exchanger. The liquid (usually an antifreeze) is mechanically circulated through plain finned tubing. When steam is used directly, the tube-within-a-tube construction makes it possible to obtain uniform distribution throughout the length of the car. It is achieved by means of heat transfer between the steam within the inner tube and the condensate returning in the annular space between the tubes.

The finned tubing at the floor must have sufficient capacity to offset effects of cold walls and windows during normal operation, and to heat the entire car to a minimum temperature of 60 during standby when the overhead system is not operating. The maximum capacity required (determined by standby requirements) varies with car construction and design temperatures, but is approximately 90,000 Btu per hour. This requires a heating capacity in finned tube of approximately 650 Btu per linear foot.

The overhead air heating coil must have sufficient capacity to heat the outdoor air brought into the car for ventilation, and to supply approximately 20 percent of the internal heat loss of the car so as to permit supply of floor heat at all times at an output that will not be objectionable to passengers sitting near it. The usual capacity of the overhead heating coil is approximately 100,000 Btu based on 2400 cfm of circulated air, with 600 cfm of this being outdoor air for ventilation. All Btu values are approximations of actual heating requirements, and do not include heat losses in the trainline (or leakage) or losses in the undercar piping. Present car designs have enabled the car builder to run the steam supply lines in recesses within the car body, thus greatly diminishing under-car losses.

Refrigeration

For cooling and dehumidification during summer, refrigeration may be obtained from ice bunkers, steam-jet systems, or mechanical compressors (driven directly from car axle by electric motors or by gas engines). Refrigeration required varies with load conditions, but 7½ tons per car is one capacity frequently used. Evaporative-type condensers are sometimes used in combination with the usual air condenser on either steam-jet or mechanical refrigeration.

When an electric motor (approx. 10 hp) is used to

drive the air-conditioning compressor, the electric power source is a problem. If the power source is an axle-driven generator, there is an appreciable increase in the drag on the locomotive. The power available for refrigeration, when train is stopped at a station, it limited to storage batteries. One solution to this problem is to use a direct-current generator driven by a gas engine. Another is to use a Diesel-driven alternator in a special head-end car to furnish power to the entire train. Recently there have been installations in which a Diesel-driven alternator is mounted on an individual passenger car to supply the power requirements of the car. The attractiveness of this type of installation can be increased by utilizing spare alternator capacity in winter for electric heating. If this capacity is supplemented by exhaust heat from the Diesel engine, there is sufficient capacity to heat the entire car and to provide hot water for washrooms when outdoor temperatures are above approximately 30 F. This feature is important on trains using Diesel locomotives, since it eliminates the need for operating the steam heating boiler in the locomotive during a portion of the year.

Air Distribution and Cleaning

Railway cars present critical problems in air distribution because air space per passenger is small (60 to 190 cu ft), and the sun load is great. An average passenger car contains approximately 5000 cu ft of air, and may seat as many as 80 passengers. The occupants are continually liberating heat, carbon dioxide, moisture, odors, and some organic matter from their breath, skin and clothing. The heat and moisture can be removed by cooling and dehumidification, but the other constituents can be successfully handled only by proper ventilation and air cleansing. In an average car, from 2000 to 2500 cfm are circulated by the air-conditioning unit. Some of this air may be recirculated, but a portion of it should always be brought in from outdoors. The amount of outdoor air desirable depends upon the type of car, number of passengers, air temperature, humidity, odors, and whether or not the occupants are smoking. It will vary from 15 to 90 percent of the total air circulated.

Careful attention must be exercised in specifying the rate of outdoor air taken in so as to fit the type of service adequately, and yet not supply more ventilation than is necessary. Conditioning of this outdoor air is a major factor in determining size of both summer and winter equipment.

For normal conditions, 10 cfm of outdoor air per passenger are provided. When smoking is permitted, at least 15 cfm should be admitted. In some dining cars, and deluxe sleeping cars, outdoor air rates as high as 20 to 30 cfm per occupant are used. A ceiling duct lengthwise along the center of the car is usually used to distribute the air to the interior by fans or blowers. A perforated ceiling supplied from an overhead duct, or delivery grilles and plaques designed to give considerable entrainment and mixing, are used to deliver air to the car space.

Smoking rooms present a special problem. The cloud of smoke that usually hangs near the ceiling can be broken up by directing incoming air along the ceiling at a velocity somewhat higher than that used for the rest of car. The air is exhausted through the washroom or lavatory. For compartments, provision is made in the door or partition for removal of used air. Lower berths are provided with a low-velocity air outlet.

Recirculating air grilles are usually of straight-flow types. Outdoor air intakes are usually located in the vestibule, on the side of the car, or on the roof, depending upon location of the cooling coils. On many air-conditioned cars, there are no dampers or shutters at the outdoor air intakes; the percentage of the outdoor air is controlled by adjusting flow through the recirculating grille.

Many coach cars are now being equipped with return-air ducts fitted in the structure of baggage racks. Part of the air circulated is returned to the blower unit through these ducts, and part through the car body. This arrangement reduces quantity and velocity of air returning through the car body, and removes smoke fumes at the source. This, and any other design features aimed at taking recirculated air at the floor and adjacent to both end doors (rather than drawing all recirculated air to one end of the car) also reduces infiltration of cold air in ankle-height strata, when doors are opened during the heating season.

All air circulated by the blower is filtered before passing over the cooling and heating coils. In some cars outdoor air and recirculated air are filtered separately before mixing, while in others air from the two sources is mixed before passing through a common filter. Filters in use are combinations of metal, wool, cloth, spun glass, hemp, paper, hair, and wire screen. Most filters have a viscous coating of oil for greater cleaning efficiency. Some types may be cleaned, re-treated, and re-used, while other types are discarded when dirty. Applications are also being made of electric precipitation for air cleaning. In this system the coarser particles are removed from the air by mechanical separation; finer materials, by electrostatic action.

Activated carbon units sometimes are used in addition to the regular filters for adsorbing odors and other impurities, thus reducing the amount of outdoor air necessary for ventilation.

Temperature and Humidity Control

Controls in a passenger car should be as automatic as possible. The regular train crew cannot be relied on to make adjustments for the comfort of passengers. For this reason the latest systems of temperature control have only an off-on switch to be operated by the train crew. When the system is in operation, heating or cooling is provided automatically as required.

When heating, it is important that floor-heat finned tubing be controlled at stable temperatures. Wide fluctuation in its temperature is highly objectionable because of location close to the passengers. Stable operation may be secured by cycling the floor heat on the basis of indoor conditions in conjunction with an overhead air-circulating system to maintain final car temperatures.

Because of window condensation and other problems, no attempt is usually made to raise relative humidity in a railroad car in winter time.

When cooling, the steam-jet refrigeration system is controlled in an on-off manner. Some means are ordinarily provided for operating mechanical compressor systems at partial capacity. In this case split evaporators are used, so that evaporator surface and compressor capacity can be reduced together under light load conditions.

Attempts have been made in the past, largely on an experimental basis, to control the relative humidity in railway cars. This was done by operating the air-conditioning equipment according to outdoor temperatures

and then re-heating the air to an acceptable temperature. This was found to be an expensive method of operating and the results did not justify the cost. A common method of controlling heating and cooling is to provide a thermostat to control heating only and a separate thermostat set at a slightly higher temperature to control the cooling equipment. This obtains heating control, ventilation, and cooling control and the reverse, as the case may be. A car in service can go from heating into cooling in a very short time. An example of this is found on the full-dome cars in service on several railroads in the country. In the early morning hours before sunrise, the car may be requiring a considerable amount of heat. Shortly after sunrise, the car may go into the ventilating cycle and as the sun load increases, cooling may be required. On the majority of car heating systems, frequent cycling of the compressor is prevented by the one degree difference between the heating and cooling control points and by lag imposed on the system by applying artificial heat to the heating thermostat when cooling is required. On some car heating systems, a degree of modulation is obtained by the use of split evaporation. This requires two cooling thermostats with one of them set at a somewhat higher temperature than the other. The lower temperature cooling thermostat controls a portion of the cooling capacity and, if unable to maintain the car temperature, the higher thermostat adds the remaining capacity.

PASSENGER BUS AIR CONDITIONING

The passenger bus designed for urban transportation operation presents a greater problem to the designer of heating systems than does the inter-urban bus. More frequent stops, and rapidly changing passenger load create this problem on urban vehicles. Provision of heat for the driver independent of the passenger heating system, is a further problem. The inter-urban bus, however, is usually a deluxe vehicle and may require a comfort cooling system. Space and weight limitations and vibration must be considered.

Heating

Recent designs of bus heating systems obtain improved air distribution. Heat in the engine coolant liquid is used to warm air by means of suitable finned coils and this heated air is distributed throughout the passenger space by ducts and outlets directed toward the floor. Some designs include finned surface near the floor in an application similar to that in railway passenger cars. Forced air circulation over this finned floor-heating surface has been provided to increase its effectiveness. Oil-burning booster heaters have been applied to many Diesel-powered buses to raise the temperature of the engine coolant for maximum engine operating efficiency, and to provide sufficient heat for the passenger space.

Ventilation

Air for ventilation is usually brought into a bus at the front, and distributed throughout the length of the passenger space by a duct or ducts near the ceiling. Except for a few designs employing 100 percent outdoor air for heating, no heating of the ventilating air has been provided. One type of distribution system for an inter-urban bus provides for a fixed minimum of outdoor air, and is arranged to increase the percentage of outdoor air to 100 percent when the heating or cooling load diminishes. The distribution ducts and diversion-damper arrangement of this system make available two supply ducts and one return duct for heating and for cooling, with a change-over to all three ducts to supply air during the intermediate ventilating cycle. This system permits utilization of atmospheric cooling and ventilation to the greatest degree when it can be most economically employed in the interval between the heating and cooling demand.

Conventional throw-away type filters or renewable filters are used in intake air ducts for many vehicles. Electrostatic filters have been successfully used in some installations. The need for elimination of dirt is great, but the problem is complicated by space limitations and limited power.

Refrigeration

Summer conditioning systems for inter-urban vehicles range in cooling capacity from 36,000 to 48,000 Btu per hour. Mechanical compression systems using refrigerants are used, and are powered by water-cooled gasoline engines of approximately 14 hp.

Complete systems add from 800 to 1300 lb to the weight of the coach. Sometimes an auxiliary generator driven by the refrigeration system engine is used and serves to help charge the bus battery, thereby offsetting power drain imposed by the ventilating blower. Belted reciprocating compressors and direct-driven V-type and rotary compressors are used, with engine speeds up to about 1800 rpm. Air-cooled condensers for this service require about 5000 cfm of outdoor air, and this is provided by either centrifugal or propeller type fans belted or direct-driven by the air-conditioning engine. Preventing noise and vibration from affecting passengers is of vital importance. Installations must be made for quick daily engine servicing. In all cases fuel is obtained from the main bus tanks, and in some, the main engine cooling system cools the air-conditioning engine.

Control

The simplest control system for heating of buses consists of a single thermostat, located in the passenger section to start and stop the blower of the heating unit. This method of control is not generally satisfactory because, without continuous air circulation, temperature gradients from floor to ceiling and from front to rear are quite pronounced. A more satisfactory method of control provides for continuous fan operation whenever heating is required. In this method a thermostat, usually in the return air to the heater, operates a modulating valve in the supply line to the heater to throttle the flow of engine coolant to the heater coil in accordance with bus interior temperature. On some systems a means of remotely adjusting the control point of the thermostat is provided at the operator's location. Automatic means are available for starting and stopping booster heaters in the engine coolant system, under control of the thermostat. Also, some systems provide for automatically stopping the blower of the heater unit and the booster coolant pumps when no heat is required in the bus. Outdoor air for ventilation, drawn into the bus by the heating unit blower, is usually not automatically controlled but is preset, either by means of a manual damper or by sizing the outdoor-air intake to provide the desired amount of outdoor air.

On buses equipped with mechanical refrigeration for air

conditioning one method of control includes a thermostat located in the passenger section to start and stop the refrigeration compressor. If the compressor is driven by the bus engine a clutch is provided which can be energized or de-energized by operation of the thermostat. On systems employing a compressor, driven by a separate engine, automatic means of starting and stopping the engine are controlled by the thermostat. Modulating control of engine driven compressors may be accomplished by automatic control of the engine throttle by a thermostat in the passenger section. Some systems provide for continuous operation of the compressor when cooling is required, the compressor being manually started by the operator. Temperature in the interior of the bus is maintained by reheating the air from the cooling coil. Automatic control of temperature is provided by a thermostat in the passenger space or the return air to the blower which operates a modulating valve on the engine coolant supply to the reheat coil.

AUTOMOBILE SUMMER AIR CONDITIONING

The basic components used for automobile air conditioning are fundamentally the same as in any stationary air conditioning system using the vapor compression cycle. For automobile application the source of power is always a major obstacle to overcome. The best solution to date for automobile applications is to operate the condensing unit by belt drive from the car motor and then control the variable refrigerating capacity caused by the variable car motor speed by suitable means.

One to seven or more horsepower are required. Compressors must be capable of operating at speeds of 4000 to 5000 rpm to last in this service. Refrigeration capacity required is about 1¼ to 1½ tons. Radiant heat, as well as high conduction heat gain due to single-glazed large glass areas, and lack of adequate insulation in car bodies require this much capacity. Proper air distribution to avoid undesirable drafts on passengers also requires careful design. Air distribution fans and electrical controls secure power from the car's electrical system which in some cases may require heavier generators and batteries to handle the load. Most systems weigh about 200 lb or less.

Early designs, and many current ones, have the cooling units installed in the rear of the car with the cooled air introduced through the package shelf at the rear of the passenger compartment. Cooled air is then distributed with or without ducts. Warm air is returned through a grille in the package shelf, or below the rear seat, thence through a filter into the cooling units. Refrigerant lines from condensing unit to cooling unit are carried from the engine compartment to the luggage compartment along the car frame under the body.

Performance at open-road speeds has been very satisfactory (even with early systems—when they were operating properly). Twenty percent or more outdoor air should be supplied, especially if passengers smoke. This air may be picked up by scoops on the side of the car or cowl, carried into the cooling unit, and mixed with returned warm air from the car.

Car temperature control, compressor capacity control to compensate for the variable compressor speed, and prevention of ice formation on the evaporators (a common fault of early systems) are accomplished by bypassing compressor discharge gas into the low-pressure side of the system, or cycling the compressor with a magnetic clutch. The solenoid bypass valve or a magnetic clutch is electrically operated by a thermostat, usually controlled by the return air to the cooling unit. The bypass is opened or the compressor stopped when the air gets too cold.

A recent development is a complete factory-assembled system, charged with refrigerant and ready to run, which is dropped into position ahead of the *fire wall* of the car and bolted into place. Flexible suction, liquid, and discharge lines are used to connect the refrigeration components together. This facilitates initial installation. It also simplifies removal for service and maintenance and eliminates the necessity to open the refrigeration components to the atmosphere with resultant loss of refrigerant or infiltration of moisture into the system.

This system combines winter heating with summer cooling. It also permits use of all or limited quantities of outdoor air, with or without heating or cooling. Outdoor air is taken in through cowl openings. This air is then pulled through a combined heating and cooling coil assembly by a fan inside the car on the fire wall. Cooled air is discharged from adjustable nozzles on the dash inside the car. Hot air for heating is discharged onto the floor as with conventional car heaters. Dampers change the air distribution from heating to cooling, and dash controls activate or shut off either heating or cooling.

Temperature control, prevention of evaporator icing and compensation for variable compressor speed are controlled by a thermostat with bulb in the air leaving the evaporator. This cycles the compressor on a magnetic clutch.

Public acceptance of current offerings has been excellent. Performance has been very satisfactory. What service difficulties have been encountered have been readily cared for by car dealers who have trained personnel to render this type of service, or by competent refrigeration service organizations. Future demand is estimated as high as 10 percent of total car production by one major manufacturer, while another estimates a million or more of its cars will be equipped in the not too distant future.

AIRCRAFT AIR CONDITIONING

In recent years, heating, cooling and ventilating of airplanes has progressed from comparatively simple systems to highly complex multi-purpose designs. The attendant control problem has become correspondingly complex. On older, non-pressurized planes, the heating system consisted either of a steam boiler and radiator or a single stage or double stage heat exchanger. On both types, the cabin temperature was adjusted by positioning the face and bypass dampers. While these were sometimes moved by an automatic modulating control, in the majority of cases they were positioned by one of the ship's crew. As these planes cruised at less than 200 mph and normally operated at low altitudes, changes in outdoor air temperatures were generally gradual enough so that manual readjustment of controls could maintain reasonably comfortable cabin conditions. Nearly all of these heating systems were marginal in respect to heat available, and the main problem was lack of heat, rather than inadequate control.

Non-Pressurized Cabins

With the advent of the combustion type heater, and use of larger and faster planes, use of manual controls became impracticable. The combustion type heaters reach full rating in less than a minute after being turned on, and as they are rated at 100,000 Btu per hr and up, and since several heaters are generally used, it would take full time of one crew member to keep cabin temperature regulated. As ships of this type are not pressurized, the heating system is still comparatively simple.

In one type, two 100,000 Btu heaters are placed in parallel positions and the ram air from an external scoop is passed through the heaters and discharged through a series of distributing outlets located in the cabin ceiling. The cabin air is discharged through grilles located in the bottom walls of the cabin. An auxiliary nose heater is used by the crew to obtain additional heat for the cockpit or for windshield defrosting.

The cabin is maintained at the desired temperature by means of an automatic control which operates both heaters simultaneously. This control consists of two duct thermostats, one being mounted in the air inlet duct between the air scoop and the heaters so that it is affected by outdoor ambient temperatures, and the other being mounted in the heater outlet duct so that it is affected by the discharge air temperatures. A thermostat in the cabin is so located that a continuous stream of cabin air passes through it. This type of control has been found to respond to a 1 deg temperature change in less than a second.

As the outdoor temperature starts to drop, the outdoor air duct thermostat decreases in resistance, unbalancing an electronic bridge. This unbalance is amplified by vacuum tubes and causes a power tube to close a relay, turning on the combustion heaters. The resulting increase in temperature is sensed by the warm air duct thermostat which increases in resistance, thus rebalancing the bridge and causing the relay to open. If there were no loss by radiation or convection from the aircraft cabin, these two duct thermostats would be sufficient for adequate control. However, the cabin thermostat is given approximately 30 times more influence than the duct thermostats and so acts as the master controller, and the duct thermostats prevent overheating or underheating and keep the discharge air from alternating between extreme cold and extreme heat.

In a slightly more elaborate system, two combustion heaters supply a plenum chamber which is maintained at a constant temperature. Air from the plenum chamber is then mixed with outdoor air to maintain desired cabin temperature. All of the warm air is discharged into the cabin through the walls. The discharge grilles are located on the floor under seats, and a modulating-type controller varies proportions of heated and outdoor air necessary to maintain desired cabin temperature. The same type of control system as previously described is used, except that an amplifier operates a two-phase motor capable of positioning control dampers instead of operating a relay which would merely open and close the fuel valve. An auxiliary duct, running from the plenum chamber, is used by the pilot as a source of windshield defrosting air.

Pressurized Cabins

With the advent of the new high-speed pressurized transport planes, and the addition of cabin cooling in addition to heating, the control problem becomes more complex. On all of these airplanes, the heat of compression from cabin supercharger must be controlled, the air cycle or expansion turbines must be turned on and also, the heat exchanger or combustion heaters, which are used in the system when cooling is required for additional heat, must be automatically controlled.

Assuming that one of these airplanes is operating in an extremely cold climate, the sequence of operation would be as follows:

The automatic controller for the supercharged-air intercooler would be in full closed position, so that none of the heat of compression would be removed, and the air would bypass the expansion turbine and its compressor and the secondary after-cooler. An additional automatic controller would be operating the combustion heater and supplying the additional heat necessary to maintain the desired cabin temperature. If a heat exchanger were used as a supplemental source of heat, a modulating control operating a damper on this exchanger would run towards full heat position.

As the airplane enters a warm climate and heat requirements drop, the combustion heater would cease operation or the heat exchanger would go to full cold position, and the modulating control on the supercharger compressor would move towards the cold position. When the outdoor ambient temperature rises so high that cooling is desired, the cabin supercharger intercooler would be opened wide. If further cooling were required, the air would go into an air cycle turbine, which is modulated to deliver the required amount of cold air to maintain a comfortable cabin temperature.

Pressure in the cabin is maintained by providing a controlled, constant rate of air flow into the cabin sufficient to maintain ventilation, and adjusting the cabin-pressure relief valve setting, by means of a cabin pressure selector, to maintain the desired cabin pressure. Limits on maximum inside to outside pressure may be of the order of 4 or 5 psi, and safety controls should be provided to prevent exceeding this limit. There is a maximum rate at which the cabin can change to a newly selected value, this rate being in some cases also adjustable.

The requirements for controls of this nature are extremely rigid. It is commonplace for ships of this type to experience changes in outdoor ambient temperatures of as much as 100 deg in a space of 5 min. For this reason, speed of sensing a change and rapidity of response in the control system is essential if satisfactory control is to be accomplished. The older type thermostats cannot be used in airplanes, due to mass of the thermostat and to vibration experienced on all airplanes. All modern controls use some type of bridge system with temperature-sensitive resistors as sensing elements. In some types of controls, the bridge system feeds a sensitive balanced relay, which in turn runs a modulating motor or controls an on-off power relay. A recent sensitive and quickly responding type uses an electronic amplifier, which in turn controls a two-phase motor, or, through relays, controls a d-c motor, or merely closes and opens a power relay for on-off applications.

In addition to extreme speed and accuracy which are required of all aircraft temperature controls, they must be able to operate under great extremes of temperature, pressure and humidity, and also withstand continuous extreme vibration. Heaters should have, in addition to control from thermostats, suitable limit controls to prevent overheating due to failure of air supply, or any other cause. Also, there should be safety devices to shut off fuel in case of flame failure.

On the latest *high speed jet airplanes*, the temperature control problem is still more severe than on the latest transports; as in addition to the accuracy required, control response must be phenomenally fast. For example,

on some, the air going to the cabin from the jet engine compressor can change the temperature at the rate of 150 deg per second. This, coupled with the fact that on smaller size pursuit ships air is changed in the cabin as much as four times per minute, makes the instantaneous sensing of change and an extremely rapid control movement essential. Also, in airplanes operating at Mach numbers in excess of 0.7, the control must react to the large adiabatic temperature rises encountered.

SHIP AIR CONDITIONING

In air conditioning a ship, the designer is faced with all problems that would normally arise on shore installation plus additional factors. Mechanical ventilation is an absolute necessity for the comfort of passengers and ships' personnel, and for utility and preservation of cargo and stores. Ships are constructed with water-tight bulkheads dividing the vessel into several compartments. This complicates the running of ductwork and results in a multiplicity of both supply and exhaust fans. Temperature and humidity requirements of various spaces aboard ship vary widely. Passenger staterooms and public spaces must have year-round air conditioning which is also being applied more and more to the quarters of the officers and crew. Boiler rooms, galleys, laundries, etc., must have ventilation, and some must have tempered air. Cargo spaces are quite likely to need dehumidification in addition to ventilation.

Inasmuch as ventilation is such a necessary factor aboard ship, the majority of recently built vessels have been utilizing the air distribution system for heating purposes.

A ship is a self-contained structure quite likely to be away from repair ports for long periods. Adequate spare parts, therefore, form an integral part of equipment furnished. The same type of equipment should be used in as many places as possible throughout the ship, in order to reduce the number of spare parts to be carried.

Preliminary system design is simplified by uniformity of conditions which apply to most ships. Among such conditions are: (1) the necessity for the vessel to supply its own power, (2) the availability of an unlimited supply of low cost steam at suitable pressure, (3) limitations of available space and permissible weight.

The problem of heat transfer and insulation must be given careful consideration. The thermal conductivity of ship building material, such as steel, copper and brass, is many times the value of building material used ashore. The length of ducts between heat sources and fans necessitates extra duct insulation. Hull insulation must be of high quality, with attention given to fireproofness, low density, low thermal conductivity, ruggedness, vermin resistance, and ease of application. Board types are most common.

Duct insulation must have the same characteristics as hull insulation. Semi-rigid and rigid board are most common. Corrugated asbestos is not used because in the presence of moisture, it tends to disintegrate. There is a growing use of natural cork on chilled air ducts because of great difficulty in applying an adequate vapor seal due to space limitations.

SHIP HEATING AND VENTILATING

It is usually most economical in weight and space to use steam duct heaters for heating spaces served by mechanical supply systems. Spaces which do not have mechanical supply, or do not require ventilation in cold weather, are heated by steam convectors or, where the load is large, by unit heaters. Ventilation air, except that supplied to auxiliary and main machinery spaces, is usually preheated to temperatures of 50 to 70 F. No recirculation is used in ventilation and heating systems, but a manual reduction (25 to 50 percent) of air quantity is made during the heating cycle. All heaters are automatically controlled, and preheaters are designed and installed to minimize possibility of freezing of condensate. Preheaters are frequently located close to the outdoor air intake in order to conserve insulation and, for the same reason, zone reheaters are located as close as possible to each zone. Where a reheater serves only one space, it is commonly located in the space.

Table 1 .... Design Conditions for Ships

Area	Outdoor Design Temperatures			Indoor Conditions		
	Heating	Ventilating	Cooling	DB & R.H.		Effective Temp.
	F	F	F	F	%	F
Living Quarters	0	95	95 (DB) 80(WB)	80	50	73–74
Public Spaces	0	95	95(DB) 80(WB)	80	55	73½–74½
Naval Vessels	+10	88(DB) 80(WB)	88(DB) 80(WB)	85	50	75–78

Combinations or variations of duct type and convection or radiant heaters are used, depending upon basic design requirements, such as weight and space limitations, and the economic justification of the cost of the type selected.

Some design temperatures and humidities for various spaces aboard ship, are given in Table 1. The resulting quantities of air should be checked against typical heating and ventilation practices for ships.*

Living Spaces

The minimum quantity of ventilation air provided for any sleeping or office space, including hospital space, should be based on a temperature rise over the outdoor air of not more than 10 deg (a rise of 7 deg is more satisfactory), with not less than 30 cfm per person.

In spaces fitted for eating, recreation, or manual work, the rise may be taken at 10 deg with not less than 20 cfm per person. The same requirement applies to mechanical exhaust, although natural exhaust may be used where only a short run of duct exists.

Heat should be furnished to maintain the following temperatures:

Staterooms, Berthing, Messing and Office Spaces 70 F
Working Spaces and Shops 60 F
Hospital Spaces 75 to 78 F

Machinery Spaces

The prime purpose of machinery space ventilation is to maintain a habitable temperature for the operating

* See Ventilation and Heating of Maritime Commission Ships, by J. W. Markert (*Heating and Ventilating*, Feb. 1943, p. 333).

personnel. It is more practicable to use *spot cooling* of personnel at working areas than to attempt to obtain uniform ambient temperature. The permissible temperature rise (above outdoors) at working stations is usually 15 deg, while the overall temperature rise is usually between 30 and 50 deg.

These spaces must be exhausted adequately, preferably by mechanical means. Every attempt should be made to *remove air at or close to the heat sources. The capacity* of the mechanical exhaust systems should be greater than the supply, taking into consideration the expansion of the supply air, to insure an indraft through access openings to the space.

Heat is not required for machinery spaces, except for those fitted with electrically operated equipment, which may remain inactive during periods while in port when heating to about 50 F will be required.

Storerooms and Cargo Spaces

The ventilation provided for these spaces depends on the type of vessel, location of space, and nature of cargo.

Ventilation is required for all closed spaces. Even if ventilation is not necessary to preserve the stores or cargo, it is required to prevent the accumulation of toxic or combustible gases and odors. One air change in 15 to 30 min is common practice, except where inflammable liquids or proximity to hot spaces requires additional ventilation. Mechanical supply and natural exhaust are usually used. However, where inflammable gases may exist, natural supply and mechanical exhaust are provided.

Many ships are fitted with dehumidification facilities for eliminating damage to the dry cargo by preventing condensation and dampness. The dehumidification load consists of moisture removed from the ventilation air passed through the dehumidifier, plus the moisture on, or given off by, the cargo, packaging, dunnage, battens, and other materials in the ship's holds. The most severe outdoor condition requires a moisture removal of 90 grains (140–50) per pound of dry air, with 88 F cooling water. The largest cargo ships are provided with equipment for removing about 250 lb of water per hour.

The dehumidifying systems generally utilize silica gel or lithium chloride with inhibitor. (See Chapter 42.) In most cases central drying equipment is provided. On large passenger ships consideration is given to the use of two dehumidifying units, because the cargo-carrying spaces are usually concentrated at the extreme ends of the vessel. A simple duct system distributes the dry air to the hold supply ventilation system. These ventilation systems use outdoor air when weather conditions are favorable. Recirculation and dehumidification are used only when necessary, i.e., when the weather dew point approaches or exceeds the temperature in the hold. Two control stations are generally provided, one in machinery space and one in chart or wheelhouse. These stations are arranged so that either may change the cargo conditioning system controls from use of outside air to use of conditioned air.

AIR-CONDITIONED SPACE TREATMENT FOR SHIPS

The application of air conditioning to new American passenger ships is well established. All passenger staterooms, except steerage and third class, are usually air conditioned. This includes staterooms for ship's personnel and offices within conditioned passenger areas. Third class staterooms are air conditioned on some ships, depending on the particular trade. Theaters, lounges, smoking rooms, beauty shops, barber shops and similar closed public spaces are usually air conditioned.

All messrooms, recreation rooms, officers' offices, crew's inboard rooms, and those having fixed portlights are usually air conditioned on passenger vessels. The treatment *depends on the requirements of the operator and the* proposed itinerary of the vessel.

SHIP SYSTEMS AND CONTROLS

The types of comfort conditioning systems used to date generally have followed conventional lines, except for those serving staterooms, offices, and similar small spaces. Large public spaces are fitted with individual systems which supply dehumidified and cooled air during *the cooling cycle, and warm air during the heating cycle.* In many cases these rooms are fitted with large glass windows and doors, and require direct radiation to offset the downdraft which would occur in cold weather. Finned-tube radiation running the full length of the glass area is commonly used for this purpose. Introduction of warm air at the sill, in lieu of direct radiation, is also used.

Systems serving most public spaces are designed to provide all outdoor air as long as the refrigeration load is less than the capacity of the cooling equipment. Many central systems are simplified by using 100 percent outdoor air all-year-round. The important problem in ship air conditioning concerns the treatment of the small spaces such as passenger staterooms, offices, and crew quarters. Low headroom, congested quarters, double berths, and unsymmetric arrangements make each space a problem in air distribution and treatment.

The simplest system used for small spaces consists of a central filter bank, supply fan, preheater, and cooling and dehumidifying coil. The preheater steam valve and cooling coil water valve are controlled in sequence by a duct thermostat, in the fan discharge, set to maintain a constant outlet air temperature. Zone reheaters are provided to take care of variations in heating loads. The reheater steam valve is controlled by a submaster thermostat at the reheater outlet. Control of room temperature is obtained by operating manual dampers in the air supply to the space. A recirculation exhaust fan is frequently provided, and operates in conjunction with automatic dampers to utilize the maximum quantity of outdoor air consistent with capacity of the cooling coils.

One system utilizes the same central supply equipment and recirculation exhaust system as the one just described, except that zone reheaters are replaced by individual space hot water reheaters. Each reheater is provided with a control valve, controlled by room thermostat. Generally, a forced-circulation single-pipe hot water system is used. This system is generally used for staterooms and small spaces devoted to first and second class passengers. The average total air per person is about 60 cfm, and average outdoor air per person is about 18 cfm.

A third system, used to a limited extent, is similar to the system just described, except that each room is provided with an induction unit (floor type where possible) which reheats the primary air supply. Control of heating coil in the induction unit is the same as noted for the system described in the previous paragraph. The average primary air supply is about 40 cfm per person. Recircula-

tion is not always used. The amount of induced air varies with unit design.

A fourth less common arrangement is similar, but makes use of hot water in the induction unit in winter and cold water in the induction unit in summer. Control of the valve on the unit is obtained by use of a summer-winter type thermostat. The unit is provided with drip pan and drain piping to remove condensation. No recirculation is used.

On cargo ships, tankers and vessels not carrying passengers or not having air conditioning, heating of crews' spaces and officers' staterooms is usually obtained through a central system having filters, preheaters and reheaters. A minimum temperature of air leaving the preheater is controlled by a duct thermostat. The reheater is controlled by a sub-master discharge-duct thermostat, reset by outside master control. Relationship between sub-master and master control (wherein discharge temperature is raised as outdoor temperature drops), is set according to a schedule based on the ship's itinerary.

Ductwork for all systems described is designed for conventional velocity. However, if power is available, and suitable duct construction and adequate sound absorbing facilities are provided, high velocity systems may be used.

BIBLIOGRAPHY

Railway Passenger Cars

Report on Performance and Cost of Operation of 1937 Internal Combustion Engine Mechanical Compression Equipment for Air Conditioning Railroad Passenger Cars (Division of Equipment Research, Association of American Railroads, May 1, 1937).

Report on Relative Performance of Air Filters (Mechanical Division, Association of American Railroads, January 15, 1938).

L. W. Wallace and G. G. Early, Jr.: Air conditioning of railroad passenger cars (*ASME Transactions*, November 1937).

Kenneth Cartwright: Passenger car cooling methods (*Refrigerating Engineering*, February, p. 83; March, p. 158, 1936).

J. R. Hornaday: Diesel drive for passenger air conditioning (*Refrigerating Engineering*, March 1942, p. 139).

G. T. Wilson: Railroad air conditioning (*Refrigerating Engineering*, May 1943, p. 323).

M. R. Eastin: Railway air conditioning (*Railway Elec. Engr.*, August-December 1942).

F. L. Sahlmann and E. M. Bill: Head-end power for railway cars (*Railway Elec. Engr.*, May 1939).

J. D. Loftis: *Head-End Power for Streamlined Passenger Trains* (ASME Raleigh Section, October 25, 1946).

P.C.C. Car Ventilation (Westinghouse Electric Corp., B-3697, September 1946).

Buses and Automobiles

E. T. Todd and F. O. Gadd: Bus heating (*Heating and Ventilating*, December 1946, p. 83).

L. W. Child: Air conditioning of automobiles and buses (*Society of Automotive Engineers Journal*, June 1938).

Jerry Hicke: Bus air conditioning (*Heating, Piping and Air Conditioning*, October 1938, p. 639).

A. J. Mallinckrodt and Lars Hanson: Bus air conditioning (*Refrigerating Engineering*, June 1939, p. 388).

O. G. Tinkey: What has been done in auto air conditioning (*Refrigerating Engineering*, January 1953, p. 31).

M. W. Baker, D. C. McCoy, H. V. Joyce, and P. J. Kent: Cars that beat the heat (*Society of Automotive Engineers Journal*, July 1953, p. 19).

P. J. Kent: Automobile air conditioning—progress and problems (ASHVE Transactions, Vol. 60, 1954, p. 37).

M. W. Baker and D. C. McCoy: Passenger automobiles (*ASRE Air Conditioning Refrigerating Data Book*, 1954-55, Chapter 50).

Airplanes

D. W. Tomlinson: Comfort in high altitude flying (ASHVE Transactions, Vol. 47, 1941, p. 57).

A. J. Hess: Heat exchangers for aircraft (*Refrigerating Engineering*, September 1944, p. 192).

B. M. Brod: Heating and ventilating for transport airplanes (ASHVE Transactions, Vol. 52, 1946).

A. A. Arnhym: *Comfortization of Aircraft* (Pitman Publishing Corp., New York, 1945).

B. L. Messinger: Refrigeration for air conditioning pressurized transport aircraft (*Heating and Ventilating*, January 1946, p. 63).

Ships

Ventilation and air conditioning of the S. S. Panama (*Heating and Ventilating*, September 1939, p. 47).

Air conditioning the new Mauretania (*Heating, Piping and Air Conditioning*, July 1939, p. 431).

J. H. Clarke: Heating, ventilating and air conditioning on shipboard (*Heating, Piping and Air Conditioning*, August, p. 467; September, p. 529; October, p. 610, 1940).

O. D. Colvin, W. H. E. Hahne, and M. R. Colby: Care of cargo at sea (*Society of Naval Architects and Marine Engineers Transactions*, Part I, Vol. 46, 1938, p. 109; Part II, Vol. 49, 1941, p. 208).

Modern Marine Engineers Manual (Cornell Maritime Press, Cambridge, Maryland, 1943, Vol. II, Sections 16-19).

Comdr. T. H. Urdahl, U.S.N.R., and W. C. Whittlesey: Warship ventilating, heating and air conditioning (ASHVE Transactions, Vol. 49, 1943, p. 35).

Comdr. R. H. Urdahl, U.S.N.R., and W. C. Whittlesey: De-U.S.N.R.: Standardized heating and ventilating equipment for fighting ships (*Heating, Piping and Air Conditioning*, July 1943, p. 333).

Comdr. R. H. Urdhal, U.S.N.R., and W. C. Whittlesey: Designing warship ventilation with standardized equipment (*Heating, Piping and Air Conditioning*, August 1943, p. 419).

J. W. Markert: Modern air conditioning (*Marine Engineering and Shipping Review*, November 1945, p. 177).

W. H. Carrier and L. E. Starr: Modern marine refrigeration and air conditioning (*Marine Engineering and Shipping Review*, April 1946, p. 132).

Capt. T. H. Urdahl and Comdr. E. R. Queer: Dehumidification protects U. S. Navy's inactive fleet (*Heating, Piping and Air Conditioning*, March 1946, p. 71).

Robert Tate: *Reconversion of Liner S. S. Lurline* (Society of Naval Architects and Marine Engineers, May 12, 1949).

J. W. Markert: Air conditioning of P-2 American President liners (*Pacific Marine Review*, August 1946).

J. W. Markert: Export lines air conditioning of four aces (*Marine Engineering*, March 1949).

CHAPTER 49.

SNOW MELTING

Design: Heating Requirement, Hydraulic Requirement, Installation: Safety, Internal Corrosion, Slab Construction, Thermal Stresses, Control, Testing, Draining, Drifting Snow, Design Example

THE practicability of melting snow by means of heated coils has been demonstrated in a large number of installations in sidewalks, roadways, ramps, and runways. In addition to eliminating the need for snow removal, other advantages gained are greater safety to pedestrians and vehicles, and reduction of labor in removal of slush from floors.

The design of a snow-melting system must determine and satisfy two primary requirements: (1) the heating requirement, and (2) the hydraulic requirement. Each of these is treated in this chapter. Several points regarding installation practices are also discussed.

DESIGN

Heating Requirement

The heating requirement for snow melting is affected by four atmospheric factors: (1) rate of snowfall; (2) air temperature, (3) wind velocity, and (4) humidity. The effects of these factors can be evaluated by consideration of the action of snow falling on a warmed surface.

The first flakes fall on a dry, warm surface, and are then warmed to 32 F and melted. The water from the melted snow soon forms a film over the entire area and starts to evaporate. The evaporation of the film is a *mass* transfer from the surface to the atmosphere. In addition, there is a *heat* transfer from the film to the ambient air and surfaces.

Mass transfer due to evaporation. The rate of evaporation of the melted snow from the snow-melting slab is affected by the wind speed and the vapor-pressure difference between the air and the melted snow. The vapor pressure of air, however, is fixed by the relative humidity and the temperature of the air. If the slab surface temperature is fixed, then the evaporation loss varies with changes in air temperature, relative humidity, and wind speed.

Heat transfer by convection and radiation from the melted snow to the ambient air and surfaces. A combined film coefficient has been used with sufficient accuracy to determine the combined convection and radiation loss. This coefficient is based on a heat transfer from a *wetted* surface, such as the film of melted snow, to air. The coefficient is a function of wind speed alone. The heat transfer, of course, is dependent upon the film coefficient and the temperature difference between the surface and the air. Since the surface temperature is fixed, the convection and radiation losses vary with changes in air temperature and wind speed.

To determine evaporation and heat transfer from the melted snow to the air it is necessary to know three of the four climatic factors: (1) wind speed, (2) air temperature, and (3) relative humidity, and (4) rate of snowfall. The fourth factor, rate of snowfall, is necessary to determine the heat required to warm the snow to 32 F, and to melt it.

Before deriving equations to give quantitative values for the effects of these four factors, it is necessary to consider the insulating effect of the snow before it is melted. As stated previously, the first flakes fall on a dry, warm surface, and are then warmed to 32 F and melted. During the time that the flakes are being warmed, and before they are completely melted, they can be considered as tiny blankets or insulators. The effect of this insulation has been measured[1] and can be of considerable magnitude. Since the snowflakes cover a fraction of the surface area, it has been convenient to think of the insulating effect as an area ratio. The area covered by the snowflakes is the insulated area, and the uncovered area is the uninsulated area. The term free area ratio A_r has been adopted to represent the ratio of the uncovered, or free, area to the total area or,

$$A_r = \frac{A_f}{A_t} \tag{1}$$

where

A_r = Free area ratio.
A_f = Free area, sq ft.
A_t = Total area, sq ft.

hence

$$0 \leq A_r \leq 1$$

For $A_r = 1$, the system would have to melt the snow so rapidly that the snow accumulation would be absolutely zero. This is impossible from a theoretical standpoint, but for all practical purposes it is permissible to have $A_r = 1$ as a maximum. For $A_r = 0$, the surface would be completely covered with snow to a depth sufficient to completely prevent evaporation and heat transfer losses. Research on the insulating effects of snow indicate that there are just three practical values for the free area ratio, 0, 0.5, and 1. This will be covered in greater detail in a later section.

The equations for the heating requirements of a snow-melting system have been derived and thoroughly explained in Reference 2. The general equation for slab output, q_s, as derived and discussed thoroughly in Reference 1, is

$$q_s = q_s + q_m + A_r(q_e + q_h) \tag{2}$$

where

q_s = sensible heat transferred to snow, Btu per (hour) (square foot).
q_m = heat of fusion, Btu per (hour) (square foot).
A_r = ratio of snow-free area to total area, dimensionless.
q_e = heat of evaporation, Btu per (hour) (square foot).
q_h = heat transfer by convection and radiation, Btu per (hour) (square foot).

Table 1 .... Data for Determining Operating Characteristics of Snow-Melting Systems[a, 1]

City	Period of No Snowfall: Air Temperature, F[2]: Over 32	Below or equal to 32	Mean during freezing period[3]	Wind speed, freezing period,[3] mph	Period of Snowfall[2]: Hours of snowfall[2]: Percent	Hr per year	Free area ratio, A_r	Required output, Btu per (sq ft) (hr)[4]: 0 to 49	50 to 99	100 to 149	150 to 199	200 to 249	250 to 299	300 to 349	350 to 399	400-up	Maximum output, Btu/(sq ft) (hr)
	% of winter hours with no snow at above temperatures							Frequency distribution of snowfall hours at above outputs, percent[5]									
Albuquerque, N. M.	74.7	24.7	26.2	8.5	0.6	22	1	62.0	25.4	7.6	4.2	0.0	0.8	—	—	—	259
							0	94.1	5.9	—	—	—	—	—	—	—	82
Amarillo, Tex.	73.1	26.0	24.6	13.3	0.9	33	1	33.7	35.4	15.4	10.7	3.0	1.8	—	—	—	260
							0	88.1	10.1	1.8	—	—	—	—	—	—	143
Boston, Mass.	64.6	31.4	24.7	14.2	4.0	145	1	51.5	30.0	12.3	4.3	1.2	0.6	0.1	—	—	320
							0	83.2	14.0	2.0	0.3	0.3	0.1	—	0.2	—	370[6]
Buffalo Niagara Falls, N. Y.	46.5	46.9	23.9	10.8	6.6	240	1	50.7	32.6	11.2	3.7	1.4	0.2	0.2	—	—	309
							0	95.9	3.4	0.2	0.5	—	—	—	—	—	192
Burlington, Vt.	39.0	54.5	19.6	10.8	6.5	236	1	53.7	29.9	13.2	2.5	0.6	0.1	—	—	—	280
							0	91.8	7.6	0.6	—	—	—	—	—	—	142
Caribou Limestone, Me.	21.4	70.6	16.5	10.0	8.0	290	1	35.0	39.7	16.0	5.7	2.0	1.0	0.5	0.1	—	378
							0	92.0	7.5	0.5	—	—	—	—	—	—	138
Cheyenne, Wyo.	46.4	49.8	21.5	15.3	3.8	138	1	16.5	26.2	19.4	13.1	8.6	4.7	4.2	4.7	2.6	499
							0	94.3	5.4	0.3	—	—	—	—	—	—	129
Chicago, Ill.	45.4	50.9	21.4	11.5	3.7	134	1	45.8	37.4	11.4	3.1	1.4	0.6	0.2	0.1	—	368
							0	91.5	8.1	0.3	0.1	—	—	—	—	—	165
Colorado Springs, Colo.	54.3	43.6	22.1	11.5	2.1	76	1	26.8	36.3	19.0	7.5	4.4	5.5	0.5	—	—	311
							0	98.4	1.6	—	—	—	—	—	—	—	63
Columbus, Ohio	59.0	38.1	24.5	10.0	2.9	105	1	65.8	22.4	8.0	1.7	1.7	0.4	—	—	—	261
							0	97.7	2.3	—	—	—	—	—	—	—	72
Detroit, Mich.	47.0	49.3	24.1	10.6	3.7	134	1	60.4	27.7	9.3	1.5	0.8	0.3	—	—	—	278
							0	95.9	3.5	0.6	—	—	—	—	—	—	140
Duluth, Minn.	12.6	80.5	14.5	12.0	6.9	250	1	23.7	32.9	20.6	13.7	4.3	2.5	1.7	0.6	—	382
							0	94.8	4.7	0.0	0.3	0.2	—	—	—	—	206
Falmouth, Mass.	68.5	29.5	25.5	12.8	2.0	73	1	50.0	33.9	14.2	1.6	0.3	—	—	—	—	204
							0	91.5	7.4	1.1	—	—	—	—	—	—	144
Great Falls, Mont.	49.0	46.2	16.5	14.4	4.8	174	1	26.2	27.6	16.7	16.4	7.5	4.6	0.3	0.5	0.2	451
							0	94.6	4.8	0.6	—	—	—	—	—	—	138
Hartford, Conn.	56.4	38.9	24.4	8.2	4.7	171	1	43.4	34.6	11.2	4.3	0.8	0.7	—	0.1	—	396
							0	80.4	16.7	2.2	0.5	—	0.1	—	0.1	—	383
Lincoln, Neb.	45.0	52.5	20.8	10.1	2.5	91	1	32.7	26.2	20.0	13.9	5.7	1.5	—	—	—	293
							0	97.2	2.6	0.0	0.0	0.2	—	—	—	—	202
Memphis, Tenn.	87.2	12.5	27.0	11.5	0.3	11	1	43.4	28.3	6.7	13.3	3.3	—	—	—	—	227
							0	85.0	8.3	6.7	—	—	—	—	—	—	144
Minneapolis St. Paul, Minn.	23.6	70.8	16.9	11.1	5.6	203	1	28.4	31.4	21.7	14.1	3.5	0.6	0.3	—	—	313
							0	96.5	3.1	0.3	0.1	—	—	—	—	—	155
Mt. Home, Idaho	56.3	42.6	24.9	9.5	1.1	40	1	74.2	21.9	3.9	—	—	—	—	—	—	143
							0	98.1	1.9	—	—	—	—	—	—	—	90
New York, N. Y.	55.7	42.2	24.2	11.8	2.1	76	1	53.1	31.8	9.4	2.2	1.5	1.7	—	0.3	—	355
							0	87.6	9.6	1.5	0.7	0.3	0.3	—	—	—	268
Ogden, Utah	50.0	45.6	24.3	9.4	4.4	160	1	64.6	29.2	5.8	0.3	0.1	—	—	—	—	216
							0	88.8	9.4	1.4	0.3	0.1	—	—	—	—	216[6]
Oklahoma City, Okla.	79.0	19.8	24.6	15.8	1.2	44	1	27.8	18.7	17.0	12.6	14.3	5.9	2.7	1.0	—	394
							0	95.7	4.3	—	—	—	—	—	—	—	81
Philadelphia, Pa.	75.8	22.6	26.7	9.7	1.6	58	1	62.3	23.6	10.4	2.3	0.9	0.5	—	—	—	296
							0	84.3	14.0	1.1	0.2	0.4	—	—	—	—	229
Pittsburgh, Pa.	55.2	39.8	24.3	11.6	5.0	182	1	53.6	30.8	8.4	4.6	1.9	0.7	—	—	—	282
							0	93.3	5.9	0.7	0.1	—	—	—	—	—	157
Portland, Ore.	92.9	6.1	28.9	8.4	1.0	36	1	78.0	16.9	5.1	—	—	—	—	—	—	125
							0	91.5	8.5	—	—	—	—	—	—	—	97
Rapid City, S. D.	45.2	51.6	19.3	12.9	3.2	116	1	29.7	29.0	16.0	8.4	6.3	3.6	1.9	2.0	3.1	581
							0	97.6	2.2	0.2	—	—	—	—	—	—	102
Reno, Nev.	56.0	41.6	24.3	5.6	2.4	87	1	82.6	15.4	1.8	0.2	—	—	—	—	—	152
							0	90.2	8.0	1.6	0.2	—	—	—	—	—	154[6]

Table 1 .... Data for determining Operating Characteristics of Snow-Melting Systems[a, 1] (Concluded)

City	Period of No Snowfall: Air Temperature, F[2]: Over 32	Below or equal to 32	Mean during freezing period[3]	Wind speed freezing period,[3] mph	Period of Snowfall[2]: Hours of snowfall[2]: Percent	Hr per year	Free area ratio, A_r	Required output, Btu per (sq ft) (hr)[4]: 0 to 49	50 to 99	100 to 149	150 to 199	200 to 249	250 to 299	300 to 349	350 to 399	400-up	Maximum output Btu/(sq ft) (hr)
	% of winter hours with no snow at above temperatures							Frequency distribution of snowfall hours at above outputs, percent[5]									
St. Louis, Mo.	68.7	30.4	25.0	11.5	0.9	33	1	42.9	31.4	16.7	7.1	1.9	—	—	—	—	225
							0	85.2	11.6	2.6	0.6	—	—	—	—	—	152
Salina, Kan.	60.0	38.5	23.3	10.9	1.5	54	1	44.9	31.9	12.7	7.6	2.2	0.7	—	—	—	286
							0	93.5	6.2	0.3	—	—	—	—	—	—	129
Sault Ste. Marie, Mich.	21.3	69.2	18.6	9.4	9.5	345	1	45.7	32.8	14.3	5.7	1.4	0.1	—	—	—	262
							0	97.9	2.0	0.1	—	—	—	—	—	—	144
Seattle Tacoma, Wash.	88.0	10.8	28.5	5.9	1.2	44	1	86.3	12.3	1.4	—	—	—	—	—	—	137
							0	91.0	8.1	0.9	—	—	—	—	—	—	128
Spokane, Wash.	48.5	46.1	25.7	10.7	5.4	196	1	62.6	28.7	7.4	1.1	2.0	—	—	—	—	205
							0	92.0	7.8	0.2	—	—	—	—	—	—	127
Washington, D. C.	77.9	21.2	26.8	9.6	0.9	33	1	59.0	29.8	10.6	0.6	—	—	—	—	—	154
							0	85.7	11.8	2.5	—	—	—	—	—	—	121

[a] From *Air Conditioning, Heating and Ventilating*, August, 1957, p. 87.

[1] The period covered by this table is from No. 1 to March 31, incl. with February taken as a 28¼ day month. Total hours in period = 2520. [2] The percentage in Columns 2 and 3 plus the percent under hours of snowfall total 100 percent. Note that *Hours of Snowfall* does not include idling time, and is not actual operating time. See text. [3] Snowfalls of trace amounts are not included; hence, *Hours of Snowfall* includes only those hours of 0.01 inches water equivalent per hour snowfall. [4] Output does not include allowance for back or edge losses since these depend on slab construction. [5] Percentages total 100 percent of the number of hours of snowfall. [6] *Freezing Period* is that during No Snowfall when the air temperature is 32 F or below. [7] When heat output for $A_r = 0$ equals or exceeds the heat output for $A_r = 1$, the heat transfer q_s is *from* the air *to* the slab. This occurs when snowfall is at temperatures above 32 F.

The sensible heat, q_s, to raise the temperature of the snow to 32 F is

$$q_s = 2.6s\,(32 - t_a) \qquad (3)$$

where

s = rate of snowfall, inches of water equivalent per hour.
t_a = air temperature, Fahrenheit.

The heat of fusion, q_m, to melt the snow is

$$q_m = 746s \qquad (4)$$

The heat of evaporation, q_e, (mass transfer) is

$$q_e = h_{fg}(0.0201v + 0.055)(0.185 - p_{aw})A_r \qquad (5)$$

where

h_{fg} = heat of evaporation at the film temperature, Btu per pound.
v = wind speed, miles per hour.
p_{aw} = vapor pressure of moist air, inches of mercury.

The heat transfer, q_h, (convection and radiation) is

$$q_h = 11.4(0.0201v + 0.055)(t_f - t_a)A_r \qquad (6)$$

where

t_f = water film temperature, Fahrenheit.

In addition to determining the four heating requirements, it is necessary to make allowance for back and edge losses. These losses vary from 30 to 50 percent, depending upon the slab construction and fluid temperature.

The equation for the required fluid temperature to provide an output q_o has been derived in Reference 2. For construction similar to Fig. 1, the equation is

$$t_m = 0.5q_o + t_f \qquad (7)$$

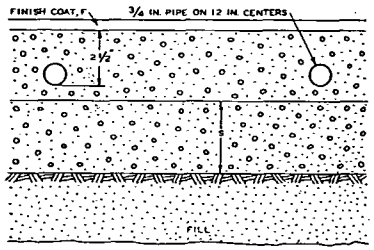
where

t_m = mean fluid temperature (antifreeze solution), Fahrenheit.

Equation 7 will apply to 1 in. as well as ¾ in. IPS pipe—see Fig. 1.

Equations 2 to 7 permit the designer to determine the heating requirement of a snow-melting system. The solutions of these equations, however, require the *simultaneous consideration* of the four climatic factors: (1) wind speed, (2) air temperature, (3) relative humidity, and (4) rate of snowfall. It is not satisfactory to use annual averages or maximums for the climatic factors. If averages or maximums are used there is no assurance that they will *ever* occur simultaneously. It is necessary, therefore, to make a frequency analysis of the solutions to Equation 2 for all the occurrences of snowfall for a period of several years. Such an analysis for 33 cities appears in Table 1 which contains the operating information for a snow-melting system. For freezing temperatures (32 F and below) without snowfall the system may be *idling* which means that some heat is supplied to the slab so that there will be immediate melting when snow starts to fall. Column 4 of Table 1 gives the average temperature during freezing periods. This temperature is used in calculating the *idling load*. The other term needed to calculate *idling load* is the wind speed during the period of freezing temperatures.

The column, Hours of Snowfall, indicates the number of hours that snow is falling at rates equal to or greater than 0.01 in. of water equivalent per hour. There are snowfalls of



F = Depth of Finish Coat—Assumed as ¼ in. of concrete. Finish Coat may be asphalt but then cover slab should be reduced from 2 in. Depth of slab should always keep thermal resistance equal to 2 in. of concrete.
S = Depth required by structural design (should be a minimum of 2 in. of concrete).

Fig. 1 .... Detail of Snow Melting Panel

trace quantities about twice as often as there are for measurable quantities of 0.01 in. or more. It is assumed that these light falls can be handled by the *idling load*.

The remaining columns of Table 1 represent the frequency distribution of required heat output. This distribution is based on the solution to the basic equation for two values of the free area ratio, A_r. This distribution represents the basis of the analysis and is also the basis for Tables 3 and 4.

For specific conditions, or for cities other than those given in Table 1, the solutions to Equation 2 are given in Table 2.

Table 2....Heat Output and Fluid Temperature for Snow-Melting System

Rate of Snowfall, s	A_r		t_a = 0 F			t_a = 10 F			t_a = 20 F			t_a = 30 F		
			Speed v			Speed v			Speed v			Speed v		
			5	10	15	5	10	15	5	10	15	5	10	15
0.08	1.0	q_s	151	205	260	127	168	209	102	128	154	75	84	94
		t_m	108	135	162	97	117	138	85	97	110	70	75	79
	0.0	q_s	66	66	66	64	64	64	62	62	62	60	60	60
		t_m	66	66	66	65	65	65	64	64	64	63	63	63
0.16	1.0	q_s	218	273	327	193	233	274	165	191	217	135	144	154
		t_m	142	169	198	129	149	170	117	129	142	100	105	109
	0.0	q_s	133	133	133	129	129	129	125	125	125	121	121	121
		t_m	99	99	99	97	97	97	95	95	95	93	93	93
0.25	1.0	q_s	292	347	401	265	305	346	235	261	287	203	212	221
		t_m	179	206	234	165	186	206	151	163	176	134	139	144
	0.0	q_s	208	208	208	202	202	202	195	195	195	188	188	188
		t_m	137	137	137	134	134	134	131	131	131	127	127	127

Note: This table is based on a relative humidity of 80 percent for all air temperatures.

s = rate of snowfall, inches of water equivalent per hour.
A_r = free area ratio.
q_s = slab output, Btu per (hour) (square foot).
t_a = air temperature, Fahrenheit.
t_m = fluid temperature, Fahrenheit. Based on construction as shown in Fig. 1.
v = wind speed, miles per hour.

Table 3....Design Data for Three Classes of Snow-Melting System*

City	Design Output, Btu per (Hr) (Sq Ft)		
	Class I system[1]	Class II system[2]	Class III system[3]
Albuquerque, N. M.	71	82	167
Amarillo, Tex.	98	143	241
Boston, Mass.	107	231	255
Buffalo-Niagara Falls, N. Y.	80	192	307
Burlington, Vt.	90	142	244
Caribou-Limestone, Me.	89 (93)	138	307
Cheyenne, Wyo.	83	129	425
Chicago, Ill.	89	165	350
Colorado Springs, Colo.	49 (63)	63	293
Columbus, Ohio	52	72	253
Detroit, Mich.	69	140	255
Duluth, Minn.	83 (114)	206	374
Falmouth, Mass.	93	144	165
Great Falls, Mont.	84 (112)	138	372
Hartford, Conn.	115	254	260
Lincoln, Neb.	64 (67)	202	246
Memphis, Tenn.	134	144	212
Minneapolis-St. Paul, Minn.	63 (95)	155	254
Mt. Home, Idaho	50	90	140
New York, N. Y.	121	298	342
Ogden, Utah	98	216	217
Oklahoma City, Okla.	66	81	350
Philadelphia, Pa.	97	229	263
Pittsburgh, Pa.	89	157	275
Portland, Ore.	86	97	111
Rapid City, S. D.	58 (86)	102	447
Reno, Nev.	98	154	155
St. Louis, Mo.	122	152	198
Salina, Kan.	85	120	228
Sault Ste. Marie, Mich.	52 (78)	144	213
Seattle-Tacoma, Wash.	92	128	133
Spokane, Wash.	87	127	189
Washington, D. C.	117	121	144

From *Air Conditioning, Heating and Ventilating*, August 1957, p. 92.

* Where idling rate is greater than Class I design rate, idling rate is given in parenthesis and should be used as Class I design output.

[1] For Class I (residential) Systems, the design output is set at that required heat output (see Table 1) when A_r = 0 at the 98th percentile of the frequency distribution; that is where 98% of the hours have this output or less.

[2] For Class II (commercial) Systems, the design output is the maximum output when A_r = 0 in Table 1 (last column).

[3] For Class III (industrial) Systems the design output is determined by the following four requirements: (1) Output is never exceeded for two consecutive hours; (2) Output for A_r = 1; Table 1, is at least 1 Btu per hour per sq ft greater than maximum output for A_r = 0; (3) A_r is greater than or equal to 0.5 maximum requirement shown in Table 1 for A_r = 1. That is, $q_s = q_r + q_m + 0.5(q_c + q_k)$ for the conditions where $q_r = q_c + q_m + q_k + q_r$ are a maximum; (4) The free area ratio A_r is unity for at least 98% of the hours listed in Table 1.

Table 4 .... Yearly Operating Data

City	Idling			Melting			City	Idling			Melting		
	Time,[1] hr per year	Rate,[2] Btu per (hr) (sq ft)	Output,[3] Btu per (year) (sq ft)	Time,[4] hr per year	Class[5]	Output,[6] Btu per (year) (sq ft)		Time,[1] hr per year	Rate,[2] Btu per (hr) (sq ft)	Output,[3] Btu per (year) (sq ft)	Time,[4] hr per year	Class[5]	Output,[6] Btu per (year) (sq ft)
Albuquerque, N. M.	897	32.5	29100	22	I	908	Mt. Home, Idaho	1546	41.9	64800	40	I	1260
					II	969						II	1530
					III	1150						III	1590
Amarillo, Tex.	944	51.1	48200	33	I	2150	New York, N. Y.	1532	50.7	77700	76	I	4180
					II	2520						II	4690
					III	2770						III	4710
Boston, Mass.	1140	52.1	59400	145	I	8000	Ogden, Utah	1655	44.6	73800	160	I	7050
					II	9080						II	7370
					III	9100						III	7370
Buffalo-Niagara Falls, N. Y.	1702	50.2	85500	240	I	11800	Oklahoma City, Okla.	719	56.2	40400	44	I	2380
					II	14600						II	2800
					III	14900						III	5400
Burlington, Vt.	1978	76.9	152000	236	I	11800	Philadelphia, Pa.	820	31.4	25700	58	I	2710
					II	13500						II	3100
					III	13800						III	3110
Caribou-Limestone, Me.	2563	93.0	238000	290	I	17800*	Pittsburgh, Pa.	1445	49.5	71500	182	I	9050
					II	20600						II	10700
					III	22500						III	11100
Cheyenne, Wyo.	1808	77.7	140000	138	I	9730	Portland, Ore.	221	17.4	3840	36	I	1300
					II	13200						II	1330
					III	20200						III	1360
Chicago, Ill.	1848	67.8	125000	134	I	7200	Rapid City, S. D.	1873	66.4	162000	116	I	7450*
					II	8390						II	8250
					III	8700						III	13400
Colorado Springs, Colo.	1583	63.4	100000	76	I	3960*	Reno, Nev.	1510	36.9	55700	87	I	2970
					II	3960						II	3030
					III	7390						III	3030
Columbus, Ohio	1383	45.0	62200	105	I	3590	St. Louis, Mo.	1104	44.8	4950	33	I	2190
					II	4180						II	2290
					III	5350						III	2380
Detroit, Mich.	1790	49.0	87600	134	I	5540	Salina, Kan.	1398	53.9	75400	54	I	2920
					II	6350						II	3370
					III	7070						III	3810
Duluth, Minn.	2922	113.8	332000	250	I	33200*	Sault Ste. Marie, Mich.	2512	77.7	195000	345	I	17600*
					II	38100						II	22200
					III	39500						III	23200
Falmouth, Mass.	1071	44.2	47400	73	I	3830	Seattle-Tacoma, Wash.	392	17.2	6750	44	I	1410
					II	4250						II	1430
					III	4290						III	1430
Great Falls, Mont.	1677	111.6	187000	174	I	13700*	Spokane, Wash.	1673	39.1	65500	196	I	8650
					II	15400						II	9350
					III	19100						III	9560
Hartford, Conn.	1412	41.9	59200	171	I	9830	Washington, D. C.	770	30.6	23600	33	I	1650
					II	10800						II	1660
					III	10810						III	1690
Lincoln, Neb.	1906	67.2	128000	91	I	4750*							
					II	8350							
					III	8520							
Memphis, Tenn.	454	32.0	14500	11	I	702							
					II	721							
					III	792							
Minneapolis-St. Paul, Minn.	2570	95.1	244000	203	I	14200*							
					II	17600							
					III	18400							

[1] From Table 1, Column 3 × 3630 (hr per year).
[2] Rate when idling, Btu per (Hr) (Sq Ft) = $(0.27v + 3.3)(32 - t)$, where v = wind speed from Column 5, Table 1, and t = air temperature from Column 4, Table 1.
[3] Product of the two preceding columns.
[4] Hours of Snowfall, from Column 7, Table 1.
[5] See footnote for Table 2.
[6] Based on the condition that surface temperature is maintained at 32 deg until required output exceeds designed output, at which time design output is used regardless of required output. Distribution of required output based on Table 1.
* Based on Idling Rate rather than Design Rate.

Air Conditioning, Heating and Ventilating, August 1957, p. 94.

Notice that these solutions are for a relative humidity of 80 percent. For other values of relative humidity, corrections can be obtained by using Equations 8 and 9 as follows:

$$\frac{dq_s}{dp_{ss}} = -A_r(0.0201v + 0.055)h_{fg} \qquad (8)$$

$$\frac{dt_m}{dp_{ss}} = \frac{-A_r}{2}(0.0201v + 0.055)h_{fg} \qquad (9)$$

Snow-melting installations are classified in Table 3 according to types as Class I, II, or III. These classes have been discussed thoroughly in Reference 3, and are defined in

the footnotes to Table 3. Briefly, snow-melting systems are classified as to the urgency for melting as follows:

Class I (minimum): Residential walks or driveways and *interplant areaways*.

Class II (moderate): Commercial (stores and offices) sidewalks and driveways, and steps of hospitals.

Class III (maximum): Toll plazas of highways and bridges, and aprons and loading areas of airports.

These classifications depend upon the allowable rate of snow melting. For example, a residential system does not have to melt snow as rapidly as a commercial system. In fact, a depth of snow of an inch for an hour or so during a heavy storm might not be objectionable with a residential system. On the other hand, a store manager would consider the system inadequate if half an inch of snow accumulated on the sidewalk in front of the store. The difference, then, between a Class I system and a Class II system is in the required ability of each system to melt snow. The one feature that is common to all classes is that the systems must be adequate for some combination of weather factors. The designer may select equipment having capacity to melt snow whenever conditions are milder than some critical values, but be willing to have an inadequate system for a given fraction of the time. In other words, the designer will take a calculated risk providing he knows the odds of that risk. For a residential system, where initial cost must be at a minimum, the designer must accept more frequent snow accumulations.

Table 3 contains the design heat requirements for the 3 classes of snow-melting systems. Under Class I systems, the values in parentheses are *idling rates* and, since they exceed the Class I design rates, should be taken as design output for this classification. Design rates may be altered by the designer if he feels that a particular job should have different design criteria from those given in the footnotes of Table 3. Any change in design conditions used should be based on the frequency distribution given in Table 1.

Use of Tables 1, 3, and 4 is illustrated by *Example 1*.

Example 1: An engineer has been retained to design snow-melting systems for the service areas of a turnpike running from the eastern edge of the Wisconsin-Illinois border northwest to the Wisconsin-Minnesota border just east of St. Paul. He decides that Chicago data will be adequate for the southern terminus and that Minneapolis-St. Paul data will be adequate for the northern terminus. His problem is to determine the heat and hydraulic requirements of the systems for service areas between Chicago and St. Paul.

Solution: Assume, for this example, that the city in question is Madison, Wis. Weather bureau records indicate that the annual average number of days with snow cover of an inch or more would be 100, and that the engineer can assume an average snowfall of 40 inches. In addition, he can estimate about 11 days per year with a snowfall of an inch or more (see Reference 3).

For the walkways to the restaurant from the parking area a Class I design rate could be used. This rate could be taken as 90 Btuh per sq ft. This is in good agreement with data in Tables 3 and 4 which give the rate at 89 (design rate) for Chicago and 95 (idling rate) for Minneapolis.

The lanes leading from the turnpike to the gasoline pumps and parking areas should be rated as Class II areas. A check of Tables 1 and 3 would indicate that 160 Btuh per sq ft would be adequate.

If an emergency area were included, for a wrecking truck, ambulance, or police garage, it would be wise to consider a Class III rate for such areas. An inspection of Table 1 for Chicago shows that a rate of 275 Btuh per sq ft would be adequate for $A_r = 1$ for 99.4 percent of the time. Similarly, 275 would be adequate 99.4 percent of the time in St. Paul. Therefore, 275 Btuh per sq ft seems sufficient for the emergency areas. Table 3 in Class III column lists 350 Btuh per sq ft for Chicago and 254 for St. Paul but for uses similar to the areas in this example, 275 should be adequate.

Table 5 .... Physical Properties of Antifreeze Solutions[a]

Fluid	Freezing Protection Temp. F	Item	Fluid Temperature—Fahrenheit							
			−20	0	20	40	120	140	160	200
Ethylene Glycol 15.3% by vol.	+20	$\nu \times 10^5$	—	—	—	2.64	0.840	0.687	0.577	0.458
		c	—	—	—	0.935	0.956	0.960	0.962	0.969
		w	—	—	—	64.0	62.8	62.5	62.1	58.8
Ethylene Glycol 31.4% by vol.	0	$\nu \times 10^5$	—	—	6.86	4.18	1.19	0.955	0.784	0.609
		c	—	—	0.833	0.850	0.895	0.905	0.910	0.923
		w	—	—	65.7	65.4	64.1	63.7	63.3	60.0
Ethylene Glycol 42.7% by vol.	−20	$\nu \times 10^5$	—	16.1	9.47	5.75	1.46	1.20	0.950	0.748
		c	—	0.764	0.775	0.788	0.832	0.845	0.856	0.884
		w	—	66.8	66.7	66.3	65.3	64.9	64.4	60.6
Ethylene Glycol 51.2% by vol.	−40	$\nu \times 10^5$	46.3	21.3	12.13	7.54	1.77	1.46	1.14	0.870
		c	0.682	0.717	0.726	0.745	0.809	0.823	0.835	0.854
		w	68.2	67.6	67.4	67.2	65.9	65.4	65.0	61.1
Heat Transfer Oil	−40	$\nu \times 10^5$	105	43.1	29.7	14.0	3.13	2.51	2.06	1.48
		c	0.382	0.390	0.400	0.408	0.444	0.452	0.462	0.480
		w	62.9	62.5	62.0	61.6	59.1	58.6	57.6	55.3
Water	+32	$\nu \times 10^5$	—	—	—	1.71	0.603	0.494	0.413	0.328
		c	—	—	—	1.005	0.999	0.999	1.001	1.005
		w	—	—	—	62.4	61.7	61.4	61.0	60.1

[a] Based on data given in Reference 6.
ν = Kinematic viscosity, (feet squared per second) (e.g. for oil at 40 F, ν = 0.000140 ft² per sec.). c = specific heat, Btu per (pound) (Fahrenheit degree). w = specific weight, pounds per cubic foot.

Hydraulic Requirement

After determining the heating requirements, it is necessary to determine the hydraulic requirements of the system. This can be done by means of the procedures explained in Chapter 4, Fluid Flow, but it is necessary to use the proper physical properties of the antifreeze solution. A complete discussion of the hydraulic problem is given in Reference 4.

The main consideration is the proper allowance for viscosity. Table 5 gives viscosities for typical fluids used as antifreezes for snow-melting systems. Viscosities are given in feet squared per second. Table 6 can be used in conversion of viscosity units. A large increase in viscosity will be noted for glycols and oils—about 20 times—as the fluid temperature changes from 160 F to 0 F. This viscosity change has two effects. First, an increase in viscosity will increase the fluid friction in the piping circuit. Second, an increase in viscosity will decrease the pump capacity—in both volume and head. The effect of viscosity on fluid friction in the piping circuit is illustrated in Fig. 2.

Table 6 .... Conversion of Kinematic Viscosity Units[a]

Centistokes	(Ft²/Sec) × 10^5	SSU[b]	Centistokes	(Ft²/Sec) × 10^5	SSU[b]
2	2.15	32.6	31	33.4	145.7
2.5	2.69	36.0	32	34.4	150.2
3	3.23	36.0	33	35.5	154.7
3.5	3.77	37.6	34	36.6	159.2
4	4.30	39.1	35	37.7	163.7
4.5	4.84	40.8			
5	5.38	42.4			
6	6.46	45.6	36	38.7	168.2
7	7.53	48.8	37	39.8	172.7
8	8.61	52.1	38	40.9	177.3
9	9.68	55.5	39	42.0	181.8
10	10.8	58.9	40	43.0	186.3
11	11.8	62.4	41	44.1	190.8
12	12.9	66.0	42	45.2	195.3
13	14.0	69.8	43	46.3	199.8
14	15.1	73.6	44	47.3	204.4
15	16.1	77.4	45	48.4	209.1
16	17.2	81.3	46	49.5	213.7
17	18.3	85.3	47	50.6	218.3
18	19.4	89.4	48	51.6	222.9
19	20.4	93.6	49	52.7	227.5
20	21.5	97.8	50	53.8	232.1
21	22.6	102.0	55	59.2	255.2
22	23.7	106.4	60	64.6	278.3
23	24.7	110.7	65	69.9	301.4
24	25.8	115.0	70	75.3	324.4
25	26.9	119.3			
26	28.0	123.7			
27	29.1	128.1			
28	30.1	132.5			
29	31.2	136.9			
30	32.3	141.3			

Over 70.0 centistokes, SSU = 4.635 × centistokes.
[a] Kinematic viscosity in (ft²/sec = 1.076 × 10^{-5} × centistokes.
[b] Values listed for SSU (Saybolt Seconds—Universal) are for fluid temperatures of 100 F. To obtain the Saybolt Universal viscosity equivalent to a kinematic viscosity determined at a Fahrenheit temperature t, multiply the equivalent Saybolt Universal viscosity at 100 F by $1 + (t - 100)$ 0.000064; for example, 10 centistokes at 210 F are equivalent to 58.9 × 1.0070 or 59.3 sec Saybolt Universal at 210 F. (Taken from *ASTM* D 445 - 53.)

For large installations, the friction losses should be calculated by the Fanning equation

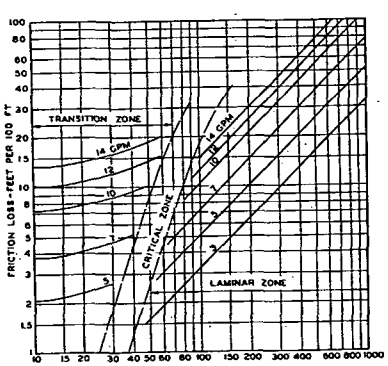
$$h_f = \frac{flV^2}{2gD} \qquad (10)$$

where

h_f = the loss in head of the fluid under conditions of flow, in feet.
l = the length of the pipe, in feet.
V = the velocity, in feet per second.
g = the acceleration due to gravity = 32.174 ft per (second) (second).
D = the internal diameter of the pipe in feet.
f = a dimensionless friction coefficient which can be determined from Fig. 4, Chapter 4. The Reynolds number can be computed from data in Table 5 of this chapter.

Solutions for the pipe friction should be plotted for temperatures at the starting condition (probably 0 F) and at the operating condition (use either 120 or 160 F). Then on the same graph, the operating curve of the pump should be plotted (see Reference 4 for such a graph). The intersection of the fluid friction curve and pump operating curve will give the operating point for the system. Table 7 can be used to allow for the viscosity effect on the pump.

The designer must decide on the tolerable viscosity limit. Generally it is between 300 and 500 SSU, although for



(For 1-in. Pipe)

Fig. 2 .... Effect of Viscosity on Friction Loss

Table 7 . . . . Viscosity Effect on Centrifugal Pump Characteristics

Kinematic Viscosity SSU[a]	Correction Factors		
	Pump Head	Pump Capacity	Pump Efficiency
0 to 30	1.00	1.00	1.00
50	1.00	1.00	0.94
100	0.98	1.00	0.88
200	0.96	0.98	0.79
300	0.95	0.97	0.73
400	0.93	0.96	0.68
500	0.92	0.96	0.65
600	0.91	0.95	0.62
700	0.90	0.94	0.59
800	0.89	0.94	0.57
900	0.88	0.93	0.55
1000	0.87	0.92	0.53

[a] SSU = Saybolt Seconds—Universal.

Table 8 . . . . Correction Factors for Pipe Size (For Laminar Flow $N_{Re} < 2000$)

Pipe Size	Inside Diameter		D_i^4 Ft[4]	$K = \frac{5.84 \times 10^{-6}}{D_i^4}$ (See Figure 2)
	Inches	Feet		
½	0.622	0.0518	7.18×10^{-6}	8.13
¾	0.824	0.0687	2.23×10^{-5}	2.62
1	1.049	0.0874	5.84×10^{-5}	1.00
1¼	1.380	0.115	1.75×10^{-4}	0.334
1½	1.610	0.134	3.26×10^{-4}	0.179
2	2.067	0.172	8.75×10^{-4}	0.0667
2½	2.469	0.206	1.81×10^{-3}	0.0322
3	3.068	0.256	4.30×10^{-3}	0.0136
3½	3.548	0.296	7.68×10^{-3}	0.00760
4	4.026	0.335	1.26×10^{-2}	0.00462
5	5.047	0.421	3.14×10^{-2}	0.00186
6	6.065	0.505	6.50×10^{-2}	0.000898

commercial or private systems (where A_s is 0.5 or 0) it may go to 750 SSU.

Efficiency loss is not important, but head and capacity losses are. Reduced flow means a longer period of time for the system to become operative from a cold start.

The viscosity limit is controlled by means of a low-limit thermostat. For example, if it is desired to hold the viscosity of the solution to less than 200 SSU (43.1 ft sq per sec), then for the oil shown in Table 5, the low-limit control would be set at 0 F.

For small installations, a quick method of determining the fluid friction for a 1-in. IPS pipe circuit is given in Fig. 2.

The required pump capacity, in pounds of fluid per hour is given by the equation:

$$C = \frac{A_p q_t}{c_m \Delta t} \tag{11}$$

where

C = pump capacity, pounds per hour.
A_p = area of slab, square feet.
q_t = total heat requirement (slab output, q_s, plus back and edge losses) Btu per (hour) (square foot).
c_m = specific heat at temperature t_m, Btu per (pound) (Fahrenheit degree).
Δt = temperature drop through circuit, Fahrenheit degrees.

For a temperature drop of $\Delta t = 20$ F, Equation 11 becomes

$$G = \frac{A_p q_t}{160 w_m c_m} \tag{12}$$

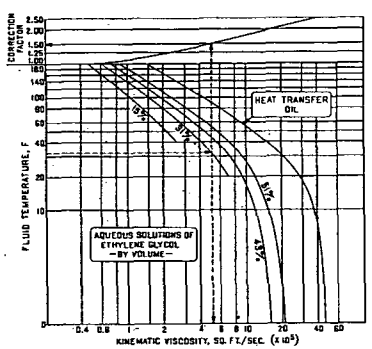
where

G = gallons per minute.
w_m = mean specific weight at temperature t_m, pounds per cubic foot.

An approximate solution for systems using pipe sizes other than 1-in. IPS can be obtained by using Table 8 or Fig. 3. Table 8 is for use when the flow is in the laminar zone. Fig. 3 is for flow in the turbulent zone. For flows where Reynolds number falls between 2000 and 3000, no reliable prediction for pressure loss can be made. It is generally safe to assume the flow to be increased to a Reynolds number of 3000 with the pressure loss estimated at that flow rate.

Figure 2 is based on a pipe size of 1 in. IPS. If another pipe size is used and the flow is *laminar*, the correction factor K from Table 8 can be used. For example, assume a viscosity of 300 SSU, a flow of 5 gpm, and a ¾-in. IPS pipe. Fig. 2 shows that for 1-in. IPS, friction loss would be 15.50 ft/100 ft. Table 8 gives $K = 2.62$ for ¾-in. pipe; therefore, friction loss would be $15.5 \times 2.62 = 40.6$ ft/100 ft.

If the flow is in the *transition* zone or is *turbulent*, then Fig. 5 of Chapter 28 can be used in conjunction with Fig. 3. For example, assume a viscosity of 0.0001 sq ft/sec (56.5 SSU), a flow rate of 14 gpm and a 1¼-in. IPS pipe size.



(For $N_{Re} > 3000$)
(Based on data given in Reference 6)

Fig. 3 . . . . Friction Correction Factor for Viscosity

Fig. 5, Chapter 28, shows a head loss for water to be 190 milinches per foot or 1.58 ft/100 ft. Fig. 3 gives a correction factor K of 1.75 for a viscosity of 0.0001 sq ft/sec; hence, the pressure loss is $1.58 \times 1.75 = 2.77$ ft/100 ft.

INSTALLATION

There are certain precautions that must be taken during installation. They concern internal corrosion, flammability, toxicity, cleaning, joints, and hookup. A comprehensive discussion of these precautions may be found in Reference 5.

Safety

Since ethylene glycol and petroleum distillates are slightly toxic, the system should be installed and maintained independently. There should be no permanent connection between the snow-melting system and the drinking water supply.

Ethylene glycol is not considered flammable. In fact, aqueous solutions of less than 60 percent glycol are used for fire sprinkler systems. These solutions do not freeze, and they are effective fire extinguishing agents.

Petroleum distillates suitable for fluids in snow-melting systems are classified as non-flammable, but have fire points between 300 and 350 F. When using fluids of this type, care should be taken to collect any oil dripping from the seals on the pump. It is good practice also to provide some barrier between the oil lines and the boiler so that in the event of a leak, a flash back from the boiler will not ignite the oil.

There are other non-flammable fluids, such as those used in some transformers, that can be used as the antifreeze. These fluids are three or four times as expensive as the glycols or oils, but combine the fire protection of the glycols with the corrosion protection of the oils.

Internal Corrosion

Ethylene glycol solutions tend to become corrosive in service; therefore, rust inhibitors are generally included. Even with an inhibitor, the solution should be tested annually to determine any change in acidity. If the test indicates that the inhibitor has been exhausted, the entire system should be drained and a fresh solution installed.

To increase the life of the inhibitor, the heat exchanger surfaces should be kept below 285 F which corresponds to about 40 psig steam. Temperatures above 300 F accelerate the deterioration of the inhibitors.

Slab Construction

It has been found satisfactory to use ¾-in pipe or tube on 12-in. centers as a standard coil. If pumping loads require a reduction in friction the pipe size may be increased to 1-in., but the slab depth must be increased accordingly.

The piping should be supported in such a manner that there is a minimum of 2 in. of concrete above and below the pipe. This requires a 5-in. slab for ¾-in. pipe and 5½-in. for 1-in. pipe.

If an insulating material is used beneath the structural slab, it is generally good practice to provide a moisture barrier between the insulation and the fill. A roofing material (such as a 55 lb felt) is often used as a moisture barrier. The joints in the barrier should be mopped, and the fill made smooth enough so that there will be no holes or gaps for moisture transfer. Also, the edges of the barrier should be flashed to the surface of the slab so that the ends are sealed.

If the pipe must pass through an expansion joint, a protective coating should be applied to the pipe for a foot or two on both sides of the joint.

If the pipe is kept dry at all times in both summer and winter, no external corrosion problems will occur.

Thermal Stresses

This problem is discussed in Reference 7. In general, there will be no ill effects from the thermal stresses if these rules of installation and operation are followed:

1. Keep the temperature difference between the fluid and the slab surface to a minimum by:
   a. Close pipe spacing (see Fig. 1).
   b. Low temperature drop in fluid, $\Delta t \leq 20$ F deg.
   c. Continuous operation (if economically feasible).
2. Keep pipe near surface to obtain about 2 in. cover.
3. Use reinforcing steel designed for thermal stress if high structural loads are expected (such as on highways).

Control

Normally snow-melting systems are manually controlled. They are started by an operator when snow is anticipated or is falling, and they are shut off when the snow has stopped falling and the area is clear. This is strictly two-position control either 0 or 100 percent operation. One version, with some degree of control, provides a surface-temperature control. The system is started manually, as before, but a thermostat controls the heat input to maintain a surface temperature of approximately 33 F.

There are several fully automatic control systems, however. In general, these systems provide some means of detecting precipitation and simultaneously sensing air temperature. One method is to set a thermostat for a 35 F air temperature and arrange to have the signal from this thermostat activate a heating element on a receptacle. This warm receptacle collects the precipitation, and melts it if it is frozen. The water then runs through a tube and activates another circuit. This second circuit can be used to start the snow-melting system. Another thermostat can be used to control the surface temperature of the snow-melting slab.

For basic information on control equipment, such as thermostats, and for fundamentals of control circuits, see Chapter 43.

Testing

After installation and before pouring concrete, all piping should be tested to about 100 psig. This pressure should be maintained until all welds and connections have been checked for leaks. If an oil is used as an antifreeze, the test should be performed with air or some gas (only 50 psi required with an air test) but *not with water*. The danger of using water is that the pipe may not be thoroughly dried when the oil is introduced. The water will tend to collect, and when the temperature falls below 32 F the collected water may freeze and cause damage.

Draining

Proper drainage is necessary for both the slab surface and the coil.

The slab must be sloped so that the water from the melted snow can run off. Puddles are objectionable on sidewalks or drives. They cause splashing, and they retard heat flow from the embedded pipes.

The pipes must be placed so that they may be drained. If the antifreeze becomes corrosive it must be drained. It is not

good practice to rely entirely on blowing of air through the system.

Drifting Snow

It is quite likely that some drifting will occur on every system that is adjacent to a wall or vertical surface. The designer should try to anticipate this condition and add extra piping in these areas. If possible, coils should be added in the vertical surface. Another expedient is to carry the drainage to the area expected to be drifted. The drainage will tend to wash away some of the snow.

Design Example

Example 2: Assume that there are three independent hydraulic systems to be used in the snow-melting application described in *Example 1*. One system will be for the Class I area, another for the Class II, and the third for the emergency area, or Class III area.

Determine the hydraulic requirement for the Class II system, assuming an area of 100,000 sq ft.

Solution: The heating requirement has been determined as 160 Btuh per sq ft. Since an area of this type has a large perimeter, assume the edge and back losses add about 40 percent or 64 Btu per sq ft. The total heat requirement, then, will be taken as 224 Btuh per sq ft or 22.4×10^6 Btuh for the 100,000 sq ft area.

The fluid temperature is determined by using Equation 7:

$$t_m = 0.5\ q_o + t_f$$

Since

$$q_o = 160 \text{ Btuh per sq ft, and } t_f = 33 \text{ F}$$
$$t_m = 0.5 \times 160 + 33 = 113 \text{ F.}$$

If an oil antifreeze is used, the specific heat, c_m, the kinematic viscosity ν_m, and the specific weight w_m, at 113 F can be found by interpolation in Table 5 as

$$c_m = 0.440 \text{ Btu per (lb) (F deg)}$$
$$\nu_m = 3.88 \times 10^{-4} \text{ sq ft per sec.}$$

An accurate determination of viscosity cannot be made by interpolation in Table 5 and should therefore be made by use of *ASTM Standard Viscosity Chart B*.

$$w_m = 59.4 \text{ lb per cu ft}$$

Then from Equation 12

$$G = \frac{A_p q_t}{160 w_m c_m} = \frac{10^6 \times 225}{160 \times 59.4 \times 0.440} = 5380 \text{ gpm}$$

Equation 12 is based on a temperature drop of 20 F deg, therefore, the fluid must enter the slab at 123 F and leave at 103 F in order to have the desired mean temperature of 113 F.

Assume that the coil with the longest *equivalent* length is 300 ft and has a flow of 12 gpm. Such an arrangement would suggest the use of a coil of 1 in. IPS. From Table 1, Chapter 31, the ID = 1.049 in. = 0.0875 ft which gives an internal cross-sectional area of 0.00600 sq ft.

For 12 gpm the flow is 0.0268 cu ft per sec and the velocity is

$$V = \frac{0.0268}{0.00600} = 4.47 \text{ fps}$$

and Reynolds number, N_{Re}, is

$$N_{Re} = \frac{VD}{\nu_m} = \frac{4.47 \times 0.0875}{3.88 \times 10^{-4}} = 10{,}100$$

From Fig. 4, Chapter 4, using a roughness factor $e = 0.00015$, whence $e/D = 0.00172$, the friction factor f is found to be 0.0333. Substituting in Equation 10 yields

$$h_f = f \times \frac{L}{D} \times \frac{V^2}{2g} = 0.0333 \times \frac{300}{0.0875} \times \frac{4.47^2}{2 \times 32.16}$$

$$= 35.8 \text{ ft of fluid} = 14.8 \text{ psia}$$

The friction loss could have been approximated two ways: (1) by using Fig. 2, and (2) by using Fig. 3 of this chapter in conjunction with Fig. 5 of Chapter 28.

In order to use Fig. 2, the viscosity of the oil must be known in SSU. As given above the kinematic viscosity of the oil at 113 F is 3.88×10^{-4} sq ft per sec. From Table 6, 3.88×10^{-4} sq ft per sec is equal to 37.9 SSU. Entering Fig. 2 at 37.9 SSU and reading friction loss for 12 gpm gives 12.8 ft per 100 ft. For a 300-ft coil this is 38.4 ft of fluid. This compares fairly well with the more exact figure of 35.8 ft as found by using the Fanning equation.

If the friction loss were approximated by Fig. 3 and Fig. 5 of Chapter 28, the result would have been 40.5 ft which is determined as follows: From Fig. 5 of Chapter 28, the loss for *water* flowing at 12 gpm in a one-inch pipe is 1200 milinches per foot or 10 ft per 100 ft. In this case, then, the loss for *water* would be 30 ft.

The correction factor for oil as compared to water is given in Fig. 3 as 1.35. The friction loss then would be 1.35×30 or 40.5 ft. Both of the approximation methods are conservative, but for large systems, this conservative approach may be costly.

The pump must deliver 5380 gpm against a 14.8 psia head. The heat exchanger must deliver 22.5 million Btuh with an inlet temperature of 103 F and an outlet temperature of 123 F. Actually some allowance should be made for the temperature drop in the piping between the slab and the heat exchanger.

Both the pump and heat exchanger manufacturers should be advised that the fluid is an oil and will operate over a wide temperature and viscosity range. If a low-limit thermostat is placed in the fluid line so that the minimum fluid temperature is −10 F, it can be seen from Tables 5 and 6 that the viscosity can reach about 300 SSU. From Table 7, it can be found that such a viscosity can reduce the pump head 5 percent and the capacity 3 percent. This means that the pump should be rated at $\frac{14.8}{0.95} \approx 15.6$ psia and $\frac{5380}{0.97} = 5550$ gpm.

The expansion tank should be designed in accordance with the procedure outlined in Chapter 28, Hot Water Heating Systems. The amount of expansion should be based on the change in volume from the low temperature limit to the high temperature limit. Assuming the temperature range to be −10 F to +123 F, the expansion would be $\frac{62.7 - 59.1}{59.1} = 6.1$ percent of the total volume in the system.

REFERENCES

[1] W. P. Chapman and S. Katunich: Heat requirements of snow melting systems (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 359).

[2] W. P. Chapman: Design of snow melting systems (*Heating and Ventilating*, April 1952, p. 95, and November 1952, p. 88).

[3] W. P. Chapman: Calculating the heat requirements of a snow melting system (*Air Conditioning, Heating and Ventilating*, September 1956 through August 1957).

[4] W. P. Chapman: Snow melting system hydraulics (*Air Conditioning, Heating and Ventilating*, November 1955).

[5] P. B. Gordon: Antifreeze protection for snow melting systems (*Heating, Piping and Air Conditioning Contractors National Association Official Bulletin*, February 1950, p. 21).

[6] C. S. Cragoe: *Properties of Ethylene Glycol and Its Aqueous Solution* (National Bureau of Standards, Society of Automotive Engineers, CKC report No. 9).

[7] W. P. Chapman: Are thermal stresses a problem in snow melting systems? (*Heating, Piping and Air Conditioning*, June 1955, p. 104, and August 1955, p. 92).

CHAPTER 50

PROCESS AND PRODUCT AIR CONDITIONING

General Requirements for Manufacture, Processing, and Preservation; Design Conditions and Application Data; Classification of Problems; Moisture Content and Regain; Conditioning and Drying; Chemical and Biochemical Reactions; Crystallization; Control for Machining, Polishing, and for Static Electricity Elimination; Laboratory Conditions; Calculations

PROCESS and product air conditioning is concerned with the design and application of equipment for obtaining proper conditions for the manufacturing, processing, and preserving of material, equipment, and commodities. This chapter includes a general discussion of these conditions and also a comprehensive list of specific requirements for various types of products.

GENERAL REQUIREMENTS FOR MANUFACTURE, PROCESSING, AND PRESERVATION

In order to apply air conditioning to industrial processes, the air-conditioning engineer must have a thorough understanding of the processing problems involved. Since individual processes and machines are changing rapidly, air conditions must be revised constantly to meet new requirements.

Table 1 lists the temperatures and relative humidities required for storage of certain commodities, and for manufacturing and processing of others. In some cases the temperatures and relative humidities listed in Table 1 have no direct influence upon the product itself, but do affect the efficiency of employees and, in turn, the workmanship, uniformity, and cost of production. Sometimes, a compromise between the known optimum condition for processing and that required for worker comfort is unavoidable.

Air conditioning for industrial processes is so extensive and involved that a detailed treatment is beyond the scope of this chapter. In many industries the exact conditions to be maintained are determined and known only by the manufacturer who specifies them. In other industries there is a wide variance between manufacturers' requirements, depending on results desired, experience, and cost considerations.

CLASSIFICATION OF PROBLEMS

In general, any industrial air-conditioning problem in processing may be classified under one or more of the following:

1. Control of regain.
2. Control of rate of chemical reactions.
3. Control of rate of biochemical reactions.
4. Control of rate of crystallization.
5. Control of temperature for close tolerance machining, and grinding.
6. Control of dew point for protection of highly polished surfaces.
7. Control of humidity for static-electricity elimination.
8. Control of conditions for material-testing laboratories.

(*Text continued on p. 687*)

Table 1 .... Temperatures and Humidities Applicable to Industrial Air Conditioning*

Process	Temp. F	R.H.%
BAKERY		
Mixer (bread dough)	75–80	40–50
Fermenting	75–80	70–75
Proof box	92–96	80–85
Bread oven	375–450	
Cake oven	300–430	
Bread cooler (room or tunnel) Vacuum 28.6 in.	70–75	80–85
Cold room	40–45	
Make-up room	75–80	65–70
Cake mixing	70–75	65
Cake mixing (sponge)	95–110	
Crackers and biscuits	60–65	50
Wrapping	60–65	60–65
Dried ingredients, storage	70	55–65
Fresh ingredients, storage	30–45	80–85
Flour storage	65–80	50–65
Shortening (depending on type), storage	45–70	55–65
Sugar, storage	80	35
Water, storage	32–35	....
Wax paper, storage	70–80	40–50

Methods of Mixer Cooling

1. 35–40 F water circulated through mixer jacket.
2. 15–25 F brine.
3. Direct expansion, refrigerant circulated through mixer jacket.
4. Cracked ice added to dough in mixer.
5. Cold air introduced into mixer during mixing process.
6. Cooled agitators are used in mixers.

Mixer Load Calculations

Refrigeration is required to remove: excess ingredient, if any; heat generated by the beating and mixing of dough; excess heat in mixer body; heat of hydration of flour and water; and heat absorbed by mixer from atmosphere during mixing.

Additional factors are the design and speed of mixer, consistency, kind and mass of dough.

Data Used in Calculations:

1 bbl. flour = 200 lb.
Heat of hydration = 6.5 Btu per lb of flour.
Specific heat of flour = 0.42 Btu per lb.
Water is 65% of weight of flour.
Flour is 65% of batch.
Sponge is 60% of batch.

Total motor output is converted to heat in the mixer.

In fermenting rooms recent practice is to use direct radiation for heating, atomizing sprays for humidifying, and gravity con-

* Information in Table 1 is drawn from many sources. See bibliography at end of chapter.

Table 1 . . . . Temperatures and Humidities Applicable to Industrial Air Conditioning—(*Continued*)

BAKERY (*Continued*)

vection cooling surfaces for temperature reduction and dehumidifying—thereby eliminating objectionable air currents.

Proof box and bread cooler conditions vary slightly for dark bread.

Cakes are sterilized by ultra-violet rays before wrapping.

Process	Temp. F	R.H.%
BANANAS		
Ripening	68	90 to 95
Storage	60	85 to 90
Shipping	54 to 56	85 to 90

Forced ripening is accomplished in 4 to 5 days while slow ripening is extended to 7 to 9 days with lower temperatures.

The green fruit with a pulp temperature of 56 F is placed in a ripening room at 68 F and 90 to 95 percent relative humidity until it begins to change color. The temperature is then lowered sharply to 60 F (or slightly less) dependent upon how quickly the fruit is to be moved.

Typical refrigeration load for ripening room per carlot, with an 85 F ambient temperature:

Insulation and electric load	= 0.4 ton.
Live load	= 0.9 ton.
Pull down load (from 68 F to 60 F)	= 1.6 ton.

Minimum carlot—20,000 lb which represents about 300 stems.

Since bananas give off minute quantities of ethylene and possibly other gases necessary for ripening, ventilation during the early stages of ripening is undesirable.

Cooling unit to be sized to provide a complete air change every 1½ to 2 min.

Provide a heating system with a rated capacity to warm the fruit at a rate of not less than 2 deg per hr.

Specific heat—0.9 Btu per (lb) (F deg).

Approximate rate of evolution of heat by bananas when stored at temperatures indicated:

Bananas at 54 F	= 3,300 Btu (per ton) (24 hr).
green at 68 F	= 8,360 Btu (per ton) (24 hr).
turning at 68 F	= 9,240 Btu (per ton) (24 hr).
ripening at 68 F	= 8,360 Btu (per ton) (24 hr).

Room	Temp. F	R.H.%
BREWING		
Storage: Hops	30 to 32	55 to 62
Grain	80 max.	60 max.
Liquid yeast	32 to 34	
Lager	32 to 34	75 min.*
Ale	40 to 46	75 min.*
Fermenting Cellar: Lager	40 to 45	75 min.*
Ale	55	75 min.*
Racking Cellar	32 to 35	75
*. If wooden tankage is used, otherwise humidity controlled to prevent condensation on walls and ceiling.		

Wort cooled to 47 F for lager, 54 to 59 F for ale, by evaporative cooling or use of double pipe or plate type coolers with counterflow of cooling medium.

Fermentation produces 250 Btu per lb of sugar fermented. Lager fermented five days at 55 to 60 F, ale at 68 to 75 F, then cooled to storage temperature.

Bottled beer pasteurized by heating to 140 F in twenty minutes, maintaining temperature for eighteen minutes, then cooling to 80 F in twenty minutes. Canned beer requires one-third less time.

Cold water, brine, direct-expansion ammonia, or propylene-glycol and water solutions may be used as the cooling medium.

Process	Temp. F	R.H.%
CANDY (CHOCOLATE)		
Candy centers for coating	80 to 85	40 to 50
Hand dipping room	60 to 65	50 to 55
Enrober room	75 to 80	55 to 60
Enrobing		
Loading End	80	50
Enrober	90	13
Stringing	70	40 to 50
Tunnel	40 to 45	
Packing	65	55
Panned specialty room	70 to 75	45
General candy storage	65 to 70	40 to 50
Tempering room = Ventilation only		

Tunnel discharge room requires a dew point lower than the temperature of the product leaving the tunnel.

During the summer months the product is usually held in the tempering room for 24 hours prior to shipping.

Recovery of sugar fly in coating kettle rooms is accomplished by use of cyclone-type dust collecting devices. Supply air to coating kettles is maintained at 85 F dry-bulb and 61 F wet-bulb temperature.

Bacteriological control is employed.

Load calculations for hand dipping rooms are based on 100 lb of 90 F chocolate per (worker) (hr).

Specific heat of chocolate = 0.30 to 0.56 Btu per (lb) (F deg).
Latent heat of fusion = 34 to 40 Btu per (lb) (F deg).
Freezing point =
Sweet milk chocolate = 86 F.
Dark chocolate = 90 to 92 F.

Process	Temp. F	R.H.%
CANDY (HARD)		
Manufacturing	75 to 80	30 to 40
Mixing and cooling	75 to 80	40 to 45
Tunnel	55	
Packing	65 to 75	40 to 45
Storage	65 to 75	45 to 50
Tempering = (Ventilation only)		

Hot rooms used in the drying of jellies and gums are maintained at 120 to 150 F. A purge system using 100 percent outside air, bypassing the heating coil, is incorporated to produce rapid cooling of both the product and the room.

Cold rooms for cooling marshmallows and cast creams are maintained at 75 to 80 F with a relative humidity of 45 to 50 percent. Uniform air distribution is essential.

Standard starch drying equipment is employed.

Filtration of air is required.

Process	Temp. F	R.H.%
CERAMICS		
Refractory	110 to 150	50 to 90
Molding room	80	60 to 70
Clay storage	60 to 80	35 to 65
Decalcomania production	75 to 80	48
Decorating room	75 to 80	48

Relative humidity has no effect on the manufacture of the products.

Temperature and humidity must be controlled in the decorating shop in the whiteware plants, and the decalcomania production room.

Dust control is essential, and the dust count must be held down to four million particles per cubic foot due to the danger of silicosis.

Table 1 . . . . Temperatures and Humidities Applicable to Industrial Air Conditioning—(*Continued*)

Process	Temp. F	R.H.%
CEREAL		
Packaging	75–80	45–50
CITRUS FRUIT		
Lemons (storage)*	58	84 to 88
Grapefruit (storage)*	60	86 to 88
* Prior to transcontinental shipment		

Careful consideration must be given to air volumes, air temperatures, humidity, ventilation, and distribution.

In regions where end rot is prevalent, it may be necessary to store grapefruit at temperatures of 32 to 34 F for a period not exceeding six weeks.

Lemon and grapefruit storage requires an air-conditioning system to (1) maintain a constant temperature, (2) maintain a high relative humidity, (3) ventilate to maintain 0.1% CO_2 content, (4) obtain uniform air distribution, and (5) provide air washing or air filtration.

Load calculations include (1) transmission losses, (2) internal load-fruit cooling and respiration, cooling fruit boxes, electric load, and people, and (3) outside air load—72 cfm per carlot.

Specific heat—Btu per (lb) (F deg):

Lemons 0.94, Grapefruit 0.87, Boxes 0.40

HEAT OF EVOLUTION IN BTU PER (TON) (24 HR)

Temp.	Lemons	Grapefruit
32	580	460
40	810	1070
60	2070	2770
80	6200	4180

Approximately 10 percent of the cooling load is considered as latent heat load.

In a conventional system the air required is one cfm per storage box or 650 cfm per carload, resulting in a small temperature rise in the supply air making possible the required high humidities.

In a combination system the air required is one cfm per sq ft of floor area or 300 cfm per carlot with the addition of auxiliary humidifying nozzles to maintain the required high humidities.

Installation of metal ducts is required. Ductwork made of insulation board is sometimes preferred.

Process	Temp. F	R.H.%
DISTILLING		
Storage:		
Grain	60	35 to 40
Liquid Yeast	32 to 34	
General Manufacturing	60 to 75	45 to 60
Aging	65 to 72	50 to 60

Mashing done at 150 to 155 F, then cooled to 64 to 68 F.

Yeast culture fermented at 170 F, then cooled to 70 to 72 F.

Yeast propagated at 150 to 154 F.

Mash heated to 165 F and then cooled to 80 F, then pitched with yeast and fermented at maximum temperature of 85 F.

Cooling for various distilling processes normally accomplished by use of river or well water depending on temperatures and availability.

Low humidity and dust control important where grains are ground.

Viscous filters preferred as mold spores and bacteria are trapped in the viscous film, preventing propagation.

Process	Temp. F	R.H.%
ELECTRICAL PRODUCTS		
Electronics and X-Ray:		
Coil and transformer winding	72	15
Tube assembly	68	40
Electrical Instruments:		
Manufacture and laboratory	70	50 to 55
Thermostat assembly and calibration	76	50 to 55
Humidistat assembly and calibration	76	50 to 55
Small Mechanisms:		
Close tolerance assembly	72	40 to 45
Meter assembly and test	74 to 76	60 to 63
Switchgear:		
Fuse and cutout assembly	73	50
Capacitor winding	73	50
Paper storage	73	50
Conductor wrapping with yarn	75	65 to 70
Lightning arrestor assembly	68	20 to 40
Thermal circuit breakers assembly and test	76	30 to 60
Water wheel generators:		
Thrust runner lapping	70	30 to 50
Rectifiers:		
Processing selenium and copper oxide plates	74	30 to 40
Dust control is essential in these processes		
FLOOR COVERING		
Linoleum:		
Mechanical oxidizing of linseed oil	90 to 100	20 to 28
Printing	80	30 to 50
Stoving process	160 to 250	60 gr*
* grains per pound abs. hum.		

Some operations are stabilized to prevent mold growth.

Precise control of temperature and humidity is required for mechanical oxidizing of linseed oil. The rate of flow and the temperature of cooling water in the jacket surrounding the tank must be controlled.

Air filtration is recommended for the stoving process.

Process	Temp. F	R.H.%
FOUNDRIES		
Core making	60 to 70	
Mold making:		
Bench work	60 to 70	
Floor work	55 to 65	
Pouring	40	
Shakeout	40 to 50	
Cleaning Room	55 to 65	
* Winter design temp.		

The charging room is usually unheated.

In core making fume-exhaust hoods are required for oven and for cooling areas adjacent to ovens.

In mold making provide hoods at transfer points with wet collector dust removal system. Use 600 to 800 cfm per hood.

Pouring rooms require two-speed powered roof ventilators. Design for minimum of two cfm per sq ft floor area at low speed. Shielding is required to control radiation from hot surfaces. Proper introduction of air will minimize preheat requirements.

In shakeout room provide hoods with wet-collector dust-removal system. Exhaust 400 to 500 cfm per sq ft grate area. Roof ventilators are generally not effective.

In cleaning room provide hoods for grinders and cleaning equipment with dry cyclones or bag-type collectors.

Winter ventilation (preheated) is required to the extent of replacing exhausted air. Summer ventilation is usually supplemented by use of pedestal fans.

Spot coolers are sometimes used in larger installations.

Table 1 . . . . Temperatures and Humidities Applicable to Industrial Air Conditioning—(*Continued*)

Process	Temp. F	R.H.%
FUR		
Drying	110	....
Shock treatment	18 to 20	....
Storage	40 to 50	55 to 65

Shock treatment, for eradication of any insect infestations, requires the lowering of the temperature to 18–20 F for 3 to 4 days, then raising it to 60–70 F for 2 days, then lowering it once again to 18–20 F for 2 days and raising it to the storage temperature.

Furs remain pliable, oxidation is reduced, and color and luster are preserved when stored at 40 to 50 F.

Mold growth is prevalent with humidities above 80 percent, while hair splitting is common where humidity is lower than 55 percent.

Process	Temp. F	R.H.%
GUM		
Manufacturing	77	33
Rolling	68	63
Stripping	72	53
Breaking	74	47
Wrapping	74	58
LEATHER		
Drying:		
Vegetable tanned	70	75
Chrome tanned	120	45
Storage	50–60	40–60

After leather is moistened in preparation for rolling and stretching, it is placed in an atmosphere held at room temperature with a relative humidity of 95 percent.

Leather is usually stored in warehouses without temperature and humidity control. However, it is necessary to keep humidity below 85 percent to avoid mildew.

Air filtration is recommended for toggle machines.

Process	Temp. F	R.H.%
LENSES (OPTICAL)		
Fusing	75	45
Grinding	80	80

The air must be free of dust and temperature held constant.

To acquire desired cleanliness of air a combination of impingement and electrostatic filters are used. Dust collectors are required for grinding operations.

Rooms	Temp. F	R.H.%
LIBRARIES AND MUSEUMS		
Museum	70–80	40–60
Book storage	70–80	40–50

Spray type dehumidifiers used to eliminate SO_2. Water treatment is essential to maintain between 8.5 and 9.0 pH.

Reheat is usually needed for refrigeration cycle due to the low sensible heat load.

In extremely cold weather a lower relative humidity is required to prevent condensation on walls.

Positive high and low limit relative humidity controls are used. Do not locate water or steam piping where leakage can cause damage.

LIBRARIES AND MUSEUMS (*Concluded*)

CHECK FIGURES FOR COOLING ESTIMATES

	Low	Medium	High
Square feet floor area per person	40	60	80
Watts per sq ft of floor area	..	1	2
Room sensible heat, Btu per (hr) (sq ft)	20	35	45
Grand total heat, Btu per (hr) (sq ft)	30	51	75
Sensible heat factor	0.73	0.83	0.90
Cooling load per person, tons	0.12	0.23	0.40
cfm per sq ft of floor area	0.92	1.60	2.10

MALTING (BREWING)

Steeped 24 to 72 hr in 45 to 65 F water.

Germinated six days at 55 to 75 F.

Kilned at temperatures of 120 to 175 F.

MALTING (DISTILLING)

Germinated twenty days at 63 F.

Germination produces total heat of 16,000 Btu/bushel at varying rate depending upon grain and process.

Process	Temp. F	R.H.%
MATCHES		
Manufacture	72 to 74	50
Drying	70 to 75	40
Storage	60 to 62	50

Water evaporated is 18 to 20 lb per million matches simultaneously with the setting of the glue.

The match machine will turn out about 750,000 matches per hour.

MUSHROOMS

Sweating-out period	120 to 140	
Spawn added	60 to 75	nearly sat.
Growing period	48 to 60	moderate
Storage	32 to 35	80 to 85

As spawn starts to grow, it is necessary to abruptly cool the mushroom house by 15 deg in a 12 hr period (approx.). Usually, this is the controlling factor in selection of refrigeration equipment, unless there is portable equipment available for such a purpose.

Ductwork is usually of wood, due to the deterioration of ferrous metals during sweating-out period.

Air filtration is essential in spawn rooms. Viscous filters are preferred, as mold spores are trapped in the viscous film. Odorless oil should be used.

Heat of emission is 4 Btu per (hr) (sq ft of growing surface).

Process	Temp. F	R.H.%
PAINT APPLICATION		
Lacquers:		
Air drying	70–90	60
Baking	180–300	
Oil Paints:		
Air drying	60–90	60
Paint Spraying	75	

Spray booths to have 100 fpm face velocity. Make-up air must be preheated. Ovens must have air removed to keep fumes below explosive concentration. Equipment must be explosion-proof throughout.

Table 1 . . . . Temperatures and Humidities Applicable to Industrial Air Conditioning—(*Continued*)

Room or Process	Temp. F	R.H.%
PHARMACEUTICAL		
Powder storage (prior to mfg.)	70 to 80	30 to 35
Manufactured powder storage and packing areas	75 to 80	15 to 35
Milling room	80	35
Tablet compressing	70 to 80	40
Tablet coating room	80	35
Effervescent tablets and powders	90	15
Hypodermic tablets	75 to 80	30
Colloids	70	30 to 50
Cough syrups	80	40
Glandular products	78 to 80	5 to 10
Ampule manufacturing	80	35
Gelatin capsules	78	40 to 50
Capsule storage	75	35 to 40
Micro-analysis	80	50
Biological manufacturing	80	35
Liver extracts	70 to 80	20 to 30
Serums	74 to 78	50
Animal rooms	75 to 80	40
Small animal rooms	75 to 78	47 to 48

Gelatin capsules require varying relative humidities, depending upon size of capsule. Moisture content should not exceed 0.25 g per cu ft. Various kinds of gelatin require different temperatures.

Penicillin incubation process requires holding temperature within ½ deg F, with temperatures and humidity rigidly controlled during all manufacturing phases.

Ampule filling requires a 20 percent relative humidity when especially fine powders are used.

Uncoated tablet manufacturing requires accurate control of temperature and relative humidity, since low relative humidity causes formation of a hard outer layer, and high relative humidity retards drying at the proper rate.

Liver extracts require a low relative humidity after they are dried. Temperatures higher than 80 F will cause the extracts to deteriorate.

Tablet coating requires the control of the temperatures of all ingredients and the temperature of air introduced to coating pans.

Sterile conditions are essential in many pharmaceutical processes. Provide suitable air exhaust to remove surplus material from tablet compressing machine.

Air filtration is generally required, with positive air filtration in some areas.

ANIMAL ROOMS IN PHARMACEUTICAL LABORATORIES

In the following tabulation each of the items mentioned is equivalent in metabolism to one man:

Quantity	Animal	Weight Each
672	White Mice	21 g
110	Rats	200 g
73	Rats	400 g
70	Guinea Pigs	410 g
21	Rabbits	2.6 kg
16	Cats	3.0 kg
16	Monkeys	3.0 kg
5	Dogs	14.0 kg

Dogs are generally the worst offenders as far as odor generation is concerned.

Decontamination of exhaust air is recommended in populated areas.

For good air quality conditions the following space per animal and total air circulation (all outdoor or decontaminated in recirculation) should be provided:

Animal	cu ft	cfm	Animal	cu ft	cfm
Mice	3	0.5	Hamsters	8	1.5
Rats	4	0.75	Cats	35	10.0
Guinea Pigs	6	1.0	Dogs	150	28.0
Rabbits	10	1.9			

Process	Temp. F	R.H.%
PHOTO STUDIO		
Dressing Room	74	30–40
Drying Room	75–80	50
Printing Room	70	70
Setting Room	72	65
Developing Room	70–75	60
Storage Room	60	45

Heat liberated during printing, enlarging, and drying processes is removed through an independent exhaust system, which also serves the lamp houses and dryer hoods.

Dust control is essential, and absolute filtration is required in some areas.

Process	Temp. F	R.H.%
PHOTO MATERIAL		
Drying	20–125	40–80
Cutting and Packing	65–75	40–60
Film Base and Paper Storage	70–75	40–65
Coated Paper and Film Storage	70–75	40–65
Safety Base Film Storage	60–80	45–60
Nitrate Film Storage	40–50	40–50

Spray water must have algae inhibitor. Positive dust control must be maintained and absolute filtration is essential.

In nitrate film area take proper precautions against fires.

Recirculated air used for film drying is passed through activated carbon filters.

Relative humidity for film storage should never exceed 60 percent with a minimum of 25 percent.

Process	Temp. F	R.H.%
PLASTICS		
Manufacturing areas:		
Thermo setting molding compounds	80	25 to 30
Cellophane wrapping	75 to 80	45 to 65

Absolute filtration is required in some areas.

Collection and removal of dust and fumes is essential.

Process	Temp. F	R.H.%
PLYWOOD		
Hot Pressing (resin)	90	60
Cold Pressing	90	15 to 25
POULTRY RAISING		
Incubator	70–75	30–60
Brooder:		
1st week	70–75	60
2nd week	50–60	60
Battery room:		
Starting	70	70–75
Growing	50–60	50–60
Egg storage	45–60	70–80

Maximum ventilation is required for laying quarters during the summer months, while an attempt is made to maintain a temperature 10 deg above or below outdoor temperature during the winter, to minimize condensation on the exterior walls. This applies to houses not having forced ventilation systems.

Table 1 . . . . Temperatures and Humidities Applicable to Industrial Air Conditioning—(*Continued*)

Process	Temp. F	R.H.%
PRECISION MACHINING		
Spectrographic analysis	75–80	45–50
Gear matching and special assembly room	75–80	35–40
Gasket storage	100	50
Cement and glue storage	65	40
Precision parts machining	75	45–50
Accurate gaging and inspection	75	45–50
Precision gage manufacture and adjustment	68–75	45–50
Precision gear drive assembly	72	42–50
Watch main spring calibration	76	45
Gage room	78	50
Precision parts—honing	75–80	35–45

For product improvement and increased personnel efficiency some manufacturing areas are being provided with summer air conditioning. Where air conditioning is not provided, an ample supply of outdoor air and air motion are relied upon to provide acceptable working conditions.

Low relative humidity is usually maintained to prevent corrosion. Temperature control within a narrow range is more important than temperature maintained. Dust control is required wherever polishing operations occur. Air distribution is important to maintain constant conditions throughout the area.

Process	Temp. F	R.H.%
PRINTING		
Pressroom:		
Multicolor offset lithography	75–80	46–48
Other sheet fed printing	75–80	45–50
Newspaper and other web printing	75–80	50–55
Stock room:		
Multicolor offset lithography	73–80	[a]
Other paper storage	70–80	[b]
Binding, cutting, drying, folding, and gluing	70–80	45–50
Roll storage	73–80	50

[a] 5 to 8% above pressroom
[b] same as pressroom

Lithography requires constant humidity control of entire pressroom with paper conditioned 5 to 8 percent higher relative humidity at start. All printing requires conditioned paper otherwise it will not lie flat, with 40 to 45 percent R.H. low limit to eliminate static electricity; and 60 percent R.H. high limit to prevent swelling of the rolls and slow ink drying. Temperature is not critical, but extremes should be avoided due to softening of the rolls at high temperature and improper ink distribution at low temperatures.

Hoods must be provided for gas dryers, with solvent recovery recommended for all except job shops. Exhaust system with dust collectors incorporated is required for type and plate cleaning areas. Check use of gasoline and other solvents. Normal air cleaning is adequate. Air distribution must prevent drafts on paper in storage or process. Gas flame dryers impose unusual loads.

Air from press and storage rooms should not be recirculated through office areas.

Process	Temp. F	R.H.%
REFRIGERATION EQUIPMENT		
Valve Manufacturing	75	40
Compressor Assembly	70 to 76	30 to 45
Refrigerator Assembly	75	30 to 50
Testing	65 to 82	47

Process	Temp. F	R.H.%
RUBBER DIPPED GOODS		
Manufacture	90	....
Cementing	80	25–30
Dipping surgical articles	75–90	25–30
Storage prior to manufacture	60–75	40–60
Laboratory (ASTM Standard)	73.4	50

Solvents used in manufacturing processes are usually explosive and toxic, requiring positive ventilation.

Volume manufacturers usually install a solvent recovery system.

TEA		
Packaging	65	65

Ideal moisture content is 5 to 6% for quality and weight. Low limit moisture content for quality is 4%.

TEXTILES		
Cotton: Opening	70–75	55–60
Picking	75	55–60
Carding, Winter	75–80	55
Carding, Summer	83	90
Carding, Summer	85	80
Carding, Summer	87	70
Drawing	80	60
Roving	80	60
Ring Spinning		
Conventional	80–85	70
Long Draft	80–85	55
Frame Spinning	80–85	55–60
Spooling and Warping	78–80	65
Weaving	78–80	70–85
Cloth Room	75	65–70
Combing	75	50–65
Linen: Carding	75–80	60
Spinning	75–80	60
Weaving	80	80
Woolens: Pickers	80–85	60
Carding	80–85	65–70
Spinning	80–85	50–60
Dressing	75–80	60
Weaving		
Light goods	80–85	60
Overcoating (32 oz.)	80–85	60–65
Drawing	75	50–60
Worsteds: Carding	80–85	65
Combing	80–85	65–70
Gilling	80–85	65–70
Top Storage	70–85	75–80
Drawing	80–85	65
Cap Spinning	80–85	50–55
Spooling and Winding	75	65–70
Weaving	80	55–70
Finishing	75	60
Silk: Preparatory	80	60–65
Weaving	80	60–70
Dressing	80	60–65
Spinning	80	65–70
Throwing	80	60
Rayon: Spinning	80–90	50–60
Throwing	80	55–60
Weaving		
Regenerated	80	50–65
Acetate	80	55–60
Spun rayon	80	80

Table 1 . . . . Temperatures and Humidities Applicable to Industrial Air Conditioning—(*Concluded*)

Process	Temp. F	R.H.%
TEXTILES (*Continued*)		
Rayon (*Continued*):		
Picking	75–80	50–60
Carding, roving, drawing	80–90	50–60
Knitting		
Viscose or cuprammonium	80–85	65
Acetate	80–85	60
Laboratory (ASTM)	70	65
Rayon synthetic fiber processing:		
Viscose		
Preparatory	80	60
Weaving	80	60
Celanese		
Preparatory	80	70
Weaving	80	70–75
Nylon		
Preparatory	80	50–60
Weaving	80	50–60

Cotton: Relative humidity maintained in ring spinning depends on staple, twist, and whether leather aprons are used for conveying long draft stock. Aprons readily absorb moisture causing cotton to stick when relative humidity is above 55 percent.

With conventional 3 or 4 roll spinning, relative humidity may be as high as 70 percent dependent upon draft, twist, and staple.

Relative humidity carried in cotton weaving depends upon construction of the cloth. When automatic machines are to be tended and the warps are heavily sized, it may be as high as 90 percent.

Woolen: In woolen spinning, the relative humidity maintained for mules is generally 55 percent, with conditions over frame spinning at 55 to 60 percent. Both types of spinning depend on the class of stock spun, also the regain in the roving.

Worsteds: Top storage temperature depends on whether cellar long period conditioning at low temperature or quick conditioning at high temperature is used. Weaving relative humidity depends upon staple, quality, and construction.

Filtration of air is essential.

Rayon manufacture: The steeping room, where sheets of raw material are dipped in caustic soda then broken into a fine matted crumb, is held at 70 F and 55 percent relative humidity. Relative humidity is held down to prevent condensation on cold pipes and jackets.

In churn room, where sodium cellulose is converted into cellulose xanthate, temperatures of 75 to 80 F are maintained while humidity control is not important. Room temperatures are held below 85 F during the summer months.

The crumb is dumped into aging tanks located in a room held at 73 F with no humidity control.

For spinning operations it is desirable to limit the temperature to 90 F with a minimum relative humidity of 70 percent for the summer, while 70 percent and 75 F are desirable for the winter months.

In the storage room, where the material is held for later processing, design conditions are 85 F and 100 percent relative humidity.

The material is washed, desulfurized, bleached, and then washed again. It is then placed in a drier where controlled conditions of 100 deg and 65 percent relative humidity are required to bring the rayon back to the proper regain.

In the coning room, where winding machines place the rayon yarn on cones, temperature is held at a maximum of 80 F. Relative humidity is held at 55 percent.

Process	Temp. F	R.H.%
TOBACCO		
Cigar and cigarette making	70 to 75	55 to 65
Softening	90	85 to 88
Stemming and stripping	75 to 85	70 to 75
Packing and shipping	74 to 76	65
Filler tobacco casing and conditioning	75	75
Filler tobacco storage and preparation	78	70
Wrapper tobacco storage and conditioning	75	75

In preparation for stripping, the tobacco undergoes a softening operation, whereby it is automatically heated, moistened, and then cooled.

Control of moisture regain, and of chemical and biological reactions is required in tobacco processing.

Exact temperature and humidity conditions maintained are usually trade secrets as they affect the finished product.

(*Text continued from p. 681.*)

Moisture Content and Regain

In the manufacture or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco, and foodstuffs, the temperature and relative humidity of the air have a marked influence upon the rate of production and upon the weight, strength, appearance, and general quality of the product. The moisture content of materials having a vegetable or animal origin, and to a lesser extent minerals in certain forms, comes to equilibrium with the moisture of the surrounding air. This moisture content is known as *regain*. Standards of *regain* are fixed in the trade, and are the fundamental basis for the control of certain physical qualities of the material during manufacture.

Manufacturing economy requires that the moisture content be maintained at a level favorable to rapid and satisfactory manipulation, and to a minimum loss of material through breakage. A uniform condition is desirable in order that high-speed machinery may be adjusted permanently for the desired production with a minimum loss from delays, wastage of raw material, and defective product. *Moisture content* refers to free moisture (as in a sponge) and to hygroscopic moisture (which varies with atmospheric conditions). It is usually expressed as a percentage of the total weight of material. *Regain* is more specific and refers only to hygroscopic moisture. It is expressed as a percentage of the *bone-dry* weight of material. For example, if a sample of cloth weighing 100.0 g is dried to a bone-dry weight of 93.0 g, the loss in weight, or 7.0 g, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 g) gives the moisture content of 7 percent. The regain, which is expressed as a percentage of the bone-dry weight, is $(7.0/93.0) \times 100 = 7.5$ percent.

The use of the term *regain* does not imply that the material as a whole has been completely dried out and has re-absorbed moisture.

A basis for calculating the regain of textiles is obtained by drying, under standard conditions, a sample from the lot, and the dry weight thus obtained is used in the calculations to determine the regain.

Table 2 shows the regain or hygroscopic moisture content of several organic and inorganic materials when in equilibrium at a dry-bulb temperature of 75 F and various

Table 2 . . . . Regain of Hygroscopic Materials

Moisture Content Expressed in Percent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

Classification	Material	Description	Relative Humidity—Percent									Authority
			10	20	30	40	50	60	70	80	90	
Natural Textile Fibers	Cotton	Sea island—roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American—cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino—skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw chevennes—skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.3	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry spun—yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila and sisal—rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons	Viscose Nitrocellulose	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cuprammonium											
	Cellulose Acetate		0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
Paper	M. F. Newsprint	Wood pulp—24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	NBS
	H. M. F. Writing	Wood pulp—3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	NBS
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	NBS
	Com. Ledger	75% rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	NBS
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	NBS
Misc. Organic Materials	Leather	Sole oak—tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid tires	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford
Foodstuffs	White Bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Misc. Inorganic Materials	Asbestos Fiber	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica Gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal	Steam activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulfuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

relative humidities. The effect of temperature as compared to the relative humidity is comparatively unimportant, although sudden changes in temperature cause a slight change in regain even when the relative humidity remains stationary. Changes in temperature do, however, affect the rate of absorption or drying, although this property generally varies with the nature of the material, its thickness and density.

When hygroscopic materials absorb moisture from the surrounding air, they deliver to the air sensible heat equivalent to the latent heat released by the moisture to the material. This amount of heat should be included in the load estimate.

Conditioning and Drying

In general, the materials may be exposed to desirable humidities for treatment coincidentally with the manufacture or processing of the materials, or they may be treated separately in special enclosures. This latter treatment may be classified as conditioning or drying. The usual purpose of conditioning or drying is to establish a desired condition of moisture content and to regulate the physical properties of the material. When the final moisture content is lower than the initial one, the term *drying* is applied (See Chapter 53). If the final moisture content is to be higher, the process is termed *conditioning*. In the case of some textile products and tobacco, for example, drying and conditioning may be combined in one process for the dual purpose of removing undesirable moisture, and accurately regulating the final moisture content. Frequently, conditioning or drying is made a continuous process in which the material is conveyed through an elongated compartment by suitable means, and subjected to various controlled atmospheric conditions.

Control of Rate of Chemical Reactions

A typical example of control of the rate of chemical reactions occurs in the manufacture of rayon. The pulp sheets are conditioned, cut to size, and passed through a mercerizing process. It is essential that, during this process, close control of both temperature and relative humidity should be maintained. The temperature controls the rate of reaction directly, while the relative humidity maintains a constant rate of evaporation from the surface of the solution, and maintains a solution of known strength during the mercerizing period.

Another well-known example in this class is the *drying* of varnish which is an oxidizing process dependent upon temperature. High relative humidities have a retarding effect on the rate of oxidization at the surface, and allow the internal gases to escape freely as the chemical oxidizers *cure* the varnish from within. This produces a surface free from bubbles and a homogeneous film throughout. Desirable temperatures for *drying* varnish vary with the type. A relative humidity of 65 percent is beneficial.

Control of Rate of Biochemical Reactions

In the field of biochemical control, industrial air conditioning has been applied to many different and well-known products. All problems involving fermentation are classed under this heading. As biochemistry is a subdivision of chemistry, subject to the same laws, the rate of reaction may be controlled by temperature. An example of this is the dough room of the modern bakery. Yeast develops best at a temperature of 80 F. A relative humidity of 70 percent is maintained to hold the surface of the dough open to allow the carbon dioxide gasses formed by the fermentation to pass through and produce a loaf of bread, when baked, of even, fine texture without large voids.

Control of Rate of Crystallization

The rate of cooling of a saturated solution determines the size of the crystals formed. Both dry- and wet-bulb temperatures are of importance, as the one controls the rate of cooling, while the other, through evaporation, changes the density of the solution.

In the coating pans for pills, gum, and nuts, a heavy sugar solution is added to the tumbling mass. As the water evaporates, each separate piece is covered with crystals of sugar. A smooth, opaque coating is only accomplished by blowing into the kettle the proper amount of air at the right dry- and wet-bulb temperatures. If the cooling and drying are too slow, the coating will be rough and semi-translucent, and the appearance unsightly; if too fast, the coating will chip through to the interior. Only by balancing temperature, relative humidity, and volume of air to the sugar solution, can the proper rate be obtained and a perfect coating assured.

Control of Temperature for Close Machining Tolerances

Where tolerances must be held within 2 or 3 ten-thousandths of an inch, as in the manufacture of precision instruments, tools, and high quality lenses, temperature variations may cause expansion and contraction of material to an extent that will seriously affect the quality of the work. This type of work usually requires close temperature control to assure accuracy and uniformity of the product.

Usually the temperature level with respect to the product is not as important as controlling the temperature within close limits. For this reason, conditions are usually selected within the comfort range.

Control of Dew Point for Protection of Polished Surfaces

In the manufacture of certain metal articles, the presence of fingerprints, tarnish, or *etching* cannot be tolerated in the finished article. If these articles are manufactured under conditions of effective temperatures that will cause the hands to perspire, an unsatisfactory product will result. The salt and acid contained in body perspiration, when deposited on the highly polished article, can show corrosion and rust within a few hours if examined under a microscope.

It is therefore important to maintain temperatures and relative humidities (dew point) low enough to prevent sweating of the hands. In addition, the manufacture of polished surfaces usually requires a better-than-average job of air filtering to avoid abrasion of the surfaces.

Control of Humidity for Reduction of Static Electricity

The presence of static electricity is often detrimental to the satisfactory and economical processing of many light materials, such as textile fibers and paper. It is also extremely dangerous where explosive atmospheres or materials are present. Fortunately, this hazard is minimized by in-

creasing the relative humidity to at least 55 percent, if the material being processed is not damaged thereby.

It must be borne in mind that for successful elimination, the air that actually comes in contact with the material in the machine must be at a relative humidity of 55 percent or more. As some machines consume a great deal of power, which is converted directly into heat, the temperature in the machine may be considerably higher than the temperature adjacent to the machine where the relative humidity is normally measured. In such cases, the relative humidity in the machine will be appreciably lower than that elsewhere in the room, and it may therefore be necessary to maintain a room relative humidity of 65 percent, or even more, to maintain the desired humidity.

Control of Conditions for Material Test Laboratories

Laboratories having controlled conditions of temperature and humidity, are common, not only for the purposes of scientific research, but also for routine testing and for quality production control. A control of temperature and humidity within fairly-close plus or minus limits is usually required. Laboratories designed for scientific research may require control of conditions over a wide range, whereas the routine testing laboratory or quality control laboratory will usually be designed to maintain the ASTM Standard Conditions of 50 percent relative humidity and 23 C (73.4 F) temperature.

CALCULATIONS

The methods for determining the heating and cooling loads for the various industrial processes are similar to those outlined in Chapters 12 and 13. Some factors affecting heating or cooling requirements are given in Table 1. Because of the large number of motors and heat producing units usually found in an industrial application, it is particularly important that operating allowances, for the latent and sensible heat loads, be definitely ascertained and used in the calculations to determine the total design load.

BIBLIOGRAPHY

H. A. Mosher: When is complete air conditioning of the modern factory advisable? (*Heating, Piping and Air Conditioning*, June 1945, p. 305; July 1945, p. 385).

N. N. Wolpert: Air conditioning design conditions for various industries (*Heating and Ventilating*, May 1949, p. 70; June 1949, p. 73; July 1949, p. 79; August 1949, p. 102).

N. N. Wolpert: Air conditioning design data for commercial applications (*Heating and Ventilating*, February 1950, p. 68).

ASRE Data Book Applications 1956-57 (American Society of Refrigerating Engineers).

W. L. Fleisher: Air conditioning in the bakery (ASHVE Transactions, Vol. 37, 1931, p. 141).

W. W. Reece: Air condition the bakery throughout (*Heating, Piping and Air Conditioning*, August 1936, p. 419).

H. R. Gable: The air conditioning of processes in the bread bakery (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, October 1947, p. 107).

A. E. Stacey, Jr: Proper air conditions for the manufacturing of confections (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, October 1937, p. 640).

J. Johnson and W. B. Ogden: *Relation of Air Conditions to Tobacco Curing* (Wisconsin Agricultural Research Bureau, 110:1-48, 1931).

J. C. Siegesmund: Heating, air conditioning and insulation for penicillin production (*Heating, Piping and Air Conditioning*, August 1944, p. 475).

Air Conditioning in Textile Mills (Textile Workers Union of America, C.I.O., *Research Department Technical Report*).

L. L. Lewis: Air conditioning for textile plants making and using synthetic yarns (*Rayon Textile Monthly*, July, August, September 1930).

P. L. Davidson: Refrigeration for textile mill air conditioning (*Heating and Ventilating*, May 1947, p. 57).

C. G. Weber: Bureau of standards studies determine press room conditioning requirements (*Heating, Piping and Air Conditioning*, March 1936, p. 137).

Humidity in the pressroom (*Heating and Ventilating*, May 1932, p. 35).

Dehumidifying with gas in a printing plant (*Heating and Ventilating*, May 1935, p. 31).

R. T. Williams: Air conditioning the newspaper plant (*Heating and Ventilating*, September 1937, p. 63).

Air conditioning in the paper industry (*Heating and Ventilating*, October 1935, p. 23).

C. G. Weber: Air conditioning requirements of multicolor offset printing (*Refrigerating Engineering*, December 1936, p. 6).

C. G. Weber and L. W. Snyder: Reactions of lithographic papers to variations in humidity and temperature (*U. S. Bureau of Standards Journal of Research*, January 1934).

C. G. Weber and M. N. Geib: The treatment of offset papers for optimum register (*U. S. Bureau of Standards Journal of Research*, February 1936).

H. E. Ryerson: Silica gel air conditioning system serves rotogravure printing plant (*Heating, Piping and Air Conditioning*, August 1937, p. 497).

Air conditioning for clothing research laboratory (*Heating and Ventilating*, July 1943, p. 69).

C. S. Leopold: Controlled air supply for supercharger, carburetor and engine testing (*Refrigerating Engineering*, August 1943, p. 85).

R. P. Dewey: Close machine tolerances possible through temperature control (*Heating and Ventilating*, April 1944, p. 60).

F. M. Johnson: Moisture control by liquid absorption offers a useful air-conditioning tool (*Heating, Piping and Air Conditioning*, December 1938, p. 782).

S. C. Rothman: The engineering control of some solvent hazards in war industries (ASHVE Transactions, Vol. 50, 1944, p. 319).

D. C. Lindsay: Air conditioning as applied in theatres and film laboratories (*Society of Motion Picture Engineers Transactions*, Vol. XI, 1927, p. 335).

F. F. Stevenson: Industrial air conditioning (*Heating, Piping and Air Conditioning*, May 1948, p. 100; June 1948, p. 94).

J. F. Salsburg: Industrial air conditioning (*Heating, Piping and Air Conditioning*, July 1948, p. 88; October 1948, p. 93).

C. S. Cave: Industrial air conditioning (*Industry and Power*, July 1945, p. 67; September 1945, p. 57).

L. R. St. Onge: Reducing heat loads in industrial air conditioning (*Refrigerating Engineering*, January 1946, p. 35).

Air, light, and sound controlled in Western Electric's new plant (*Heating, Piping and Air Conditioning*, November 1948, p. 79).

J. T. Browne: New IBM plant addition, Poughkeepsie, N. Y. (*Heating, Piping and Air Conditioning*, January 1949, p. 111).

E. E. Herbacek: Photo studios need conditioning for both processing and comfort (*Heating, Piping and Air Conditioning*, June 1949, p. 85).

A. D. Benjamin: Maintaining constant relative humidity (*Air Conditioning, Heating and Ventilating*, May 1955, p. 83).

C. P. Strickland, Jr: Today's improved lemon storage (*Refrigerating Engineering*, December 1956, p. 43).

G. A. Warner: Laboratory climates automatically controlled (*Refrigerating Engineering*, June 1957, p. 50).

H. E. Rex: Refrigerating equipment for the citrus industry (*Refrigerating Engineering*, November 1957, p. 43).

R. S. Ash: Greenhouse climate control (*Refrigerating Engineering*, February 1958, p. 41).

CHAPTER 51

CONTROL OF THE INDUSTRIAL ENVIRONMENT

Heat Control of Industrial Work Areas, Thermal Standards, Control of Heat Exposures, Radiation Shielding, Roof Ventilators, Local Relief Methods, Physiological Aspects of System Design, Types and Requirements of Systems, Outlet Types and Design, Control of Contaminants

CONTROL of the industrial environment is concerned with the design and application of equipment for providing the necessary conditions within industrial areas for maintaining the efficiency, health, and safety of workers. This chapter includes a comprehensive list of the requirements for heat control and for control of gases and vapors, and dust and fumes.

General ventilation may be relied upon in some cases to control the industrial environment, as outlined in this chapter. Additional information on natural ventilation may be found in Chapter 11. If mechanical ventilation is to be used, Chapter 20, Air Distribution; Chapter 21, Air Duct Design; Chapter 22, Fans; Chapter 24, Air Cleaning; Chapter 40, Evaporative Apparatus for Heat Rejection; Chapter 41, Evaporative Air Cooling and Humidification; and Chapter 43, Automatic Control; will furnish information covering a broad range of industrial design conditions. Chapters 13, 23, and 38 provide data for mechanically cooled systems. Chapter 52, Industrial Exhaust Systems, should be consulted for the design of hoods and their exhaust duct systems. For further information on estimating radiant heat conditions see Chapters 5 and 30.

Information on globe thermometers and other instruments used to measure the industrial environment may be found in Chapter 44, Instruments and Measurements. More complete information will be found in References[1] and[2] at the end of this chapter.

Specialists in the field of industrial hygiene should be consulted in case of doubt concerning the presence of airborne industrial hazards to health. Chapter 7, Air Contaminants; Chapter 6, Physiological Principles; and Chapter 8, Air Conditioning in the Prevention and Treatment of Disease, will be of help in establishing the atmospheric conditions that should be maintained around the worker. Local codes, ordinances, or state labor laws must likewise be observed, particularly with regard to ventilation requirements for hazardous trades. Comfortable conditions, if possible, or tolerable conditions at least, are desirable because they are likely to increase the efficiency and, hence, the output of workers. The majority of air-conditioning and ventilation installations in a typical industrial plant are concerned with the control of sensible, latent, or radiant heat.

HEAT CONTROL IN INDUSTRIAL WORK AREAS

In certain industrial work situations there is considerable release of heat from the process equipment to the environment. It is not economically feasible to stop the escape of all of this process heat or to offset it completely by the usual methods of *comfort* ventilation and air conditioning. In the design and operation of control measures it is frequently necessary to accept some heat exposure in excess of simple comfort requirements.

The engineer must distinguish between the control needs for *hot-dry* industrial areas and *warm-moist* conditions. In the first case, the process gives off only sensible and radiant heat without adding moisture to the air. The heat load on exposed workers is thereby increased but the rate of cooling by evaporation of sweat is not reduced. Heat balance may be maintained although this will be at the expense, perhaps, of excessive sweating. In the warm-moisture situation, the wet process gives off principally latent heat. There may be no significant rise in the heat load on the worker but the increase in moisture content of the air will seriously reduce the heat loss by evaporation of sweat. The warm-moist situation is potentially more hazardous than the hot-dry.

Examples of hot-dry work situations are seen around hot furnaces, forges, metal-extruding and rolling mills, glass-forming machines, and the like. Typical of warm-moist operations are laundries, dye houses, and deep mines where water is extensively used for dust control.

The industrial heat problem varies in magnitude with local climatic conditions. Solar heat gain and an elevated outdoor temperature will increase the heat load at the work place but these contributions may not be very important compared with the locally generated heat of the process itself. The moisture content of the outdoor air, on the other hand, is a most important climatic factor affecting hot-dry work situations and on a moist summer day will seriously restrict evaporative cooling. For the warm-moist job, in contrast, solar heat gain and elevated outdoor temperature are the more important, since, compared with the moisture release on the job, that contributed by the outdoor air will be of little significance.

Thermal Standards for Industrial Work Areas

The work situations considered here are those in which the attainment of simple thermal comfort is not always practicable. The heat stress must be kept below the level of outright hazard, but how much below? On what criteria should the specifications for control be based? How should the standards be varied in recognition of the differing demands and work rates from one kind of job to another? What differences exist between older and younger workers or between workers with differing degrees of physical fitness? To what

extent can periods of heat exposure be offset by alternating periods of recovery? How is this expressed in terms of design of the control measures? These and other questions arise in the analysis of an industrial heat problem and must be considered by the design engineer and others before specifications are fixed. As there is no single set of thermal standards applicable to all work situations, failure to give proper recognition to the many facets of the problem can result in inadequate control measures.

Biophysical Basis for Fixing Thermal Standards

Under conditions of thermal comfort, the rate of internal heat production (metabolism) is just balanced by the rate of heat loss to the environment. This comfortable balance is maintained without active sweating, with optimum skin and deep-body temperatures and without unusual load upon the heart. At the upper limits of physiological tolerance to sustained heat exposure, thermal balance may also be established, but this is accomplished only with active sweating, elevated skin and body temperatures, and accelerated heart rate. The limit is fixed by the maximum permissible degree to which these indexes of physiological strain—sweating, body temperature, and heart rate—can be safely elevated. For practical work situations in industry, the permissible limits of these indexes must be set below the absolute physiological maxima.

The thermal relationship between man and his environment is dependent upon four independently variable thermal characteristics of the environment: air temperature, radiant temperature, moisture content of the air, and air velocity. These may combine in various ways, together with the rate of internal heat production, to create widely different degrees of heat stress. The need is to provide a rational basis for combining them into a single index that will predict the magnitude of the heat stress and will serve also as a basis for fixing permissible limits of exposure.

A start is made with the heat balance equation:

$$M \pm R \pm C = B + E \pm \Delta S \qquad (1)$$

where

M = metabolic rate.
R = rate of heat exchange with environment by radiation.
C = rate of heat exchange with environment by convection.
B = rate of heat loss from body in exhaled air.
E = rate of heat loss from body by evaporation of sweat.
ΔS = rate of change in heat content of body.

Terms on the left side of Equation 1 represent the heat load; those on the right represent the heat loss together with the heat storage or withdrawal from the body. R and C are positive when the environmental temperature is above skin temperature and negative when below.

In practice, B and ΔS are of small magnitude compared with the others and can be neglected. For a condition of heat balance, therefore, Equation 1 may be written:

$$M \pm R \pm C = E_{req} \qquad (2)$$

E_{req} = required rate of evaporation of sweat to maintain heat balance.

M can be measured by standard physiological techniques or estimated for a given work situation by reference to Table 1. On both theoretical and experimental grounds, R and C can be calculated with reasonable accuracy by means of standard equations of heat exchange:

$$R = K_r A_r (t_w - t_s) = 22(t_w - t_s) \qquad (3)$$

$$C = K_c \sqrt{V} A (t_a - t_s) = 2\sqrt{V}(t_a - t_s) \qquad (4)$$

where

K_r = coefficient of radiant heat exchange, Btu per (hour) (square foot) (Fahrenheit degree temperature difference).

K_r varies in value; depending on the mean radiant temperature of the surroundings, t_w.

A_r = radiation area of body, square feet.
A_r differs with body position and is around 0.8 of total body surface for a standing man.

22 = $K_r \times A_r$ for average size man with moderately high mean radiant temperature, t_w.

K_c = coefficient of convective heat exchange, Btu per (hour) (square foot) (unit velocity) (Fahrenheit degree temperature difference).

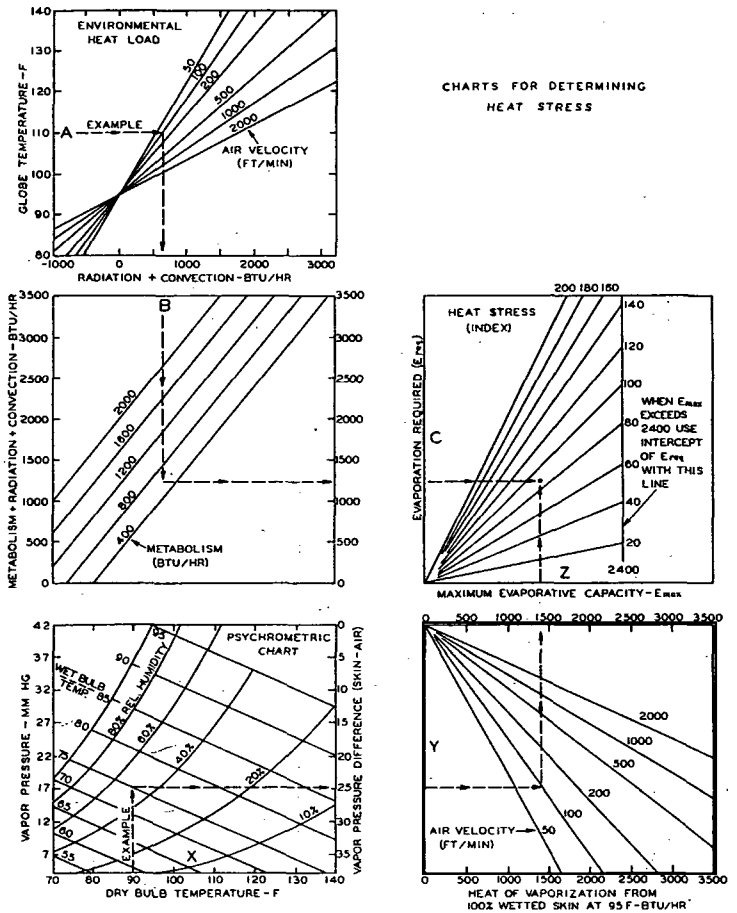
A = surface area of body, square feet.
2 = $K_c \times A$, for average size man.
V = effective velocity of air movement, feet per minute.
t_w = (black-body equivalent) mean radiant temperature of environment, Fahrenheit.
t_a = ambient air temperature, Fahrenheit.
t_s = skin temperature, Fahrenheit.

So long as there is no limitation on evaporation rate, the

Table 1 .... Estimates of Energy Metabolism (M) of Various Types of Activity

(Values apply for a 154 lb man, and do not include rest pauses)

Kind of Work	Activity	M Btu/hr
	Sleeping	250
	Sitting quietly	400
Light Work	Sitting, moderate arm and trunk movements (e.g., desk work, typing)	450–650
	Sitting, moderate arm and leg movements (e.g., playing organ, driving car in traffic)	550–650
	Standing, light work at machine or bench, mostly arms	550–650
Moderate Work	Sitting, heavy arm and leg movements	650–800
	Standing, light work at machine or bench, some walking about	650–750
	Standing, moderate work at machine or bench, some walking about	750–1000
	Walking about, with moderate lifting or pushing	1000–1400
Heavy Work	Intermittent heavy lifting, pushing or pulling (e.g., pick and shovel work)	1500–2000
	Hardest sustained work	2000–2400



CHARTS FOR DETERMINING HEAT STRESS

Example: Determine Heat Stress Index for worker doing light arm work while standing at a bench.

Metabolism 600 Btuh

Environmental conditions:

Globe thermometer temperature	110 deg
Dry-bulb temperature	90 F
Wet-bulb temperature	75 F
Air velocity	100 fpm

Solution: Follow the broken lines from the globe thermometer temperature and from dry-bulb temperature to their intersection on above diagram C to read a heat stress index of 90.

Fig. 1 .... Flow Charts for Determining Heat Stress Index Values

sweat output is automatically limited by the body to that amount which satisfies the required value of E. With restricted evaporation, however, there is excessive sweating; the body becomes fully wetted and a maximum evaporative cooling rate is established which does not increase with further sweating. This maximum evaporative cooling rate E_{max} has been established experimentally and can be calculated by the equation:

$$E_{max} = K_e A V^{0.4}(P_s - P_a) = 10V^{0.4}(P_s - P_a) \quad (5)$$

where

K_e = coefficient of heat exchange by evaporation, Btu per (hour)(square foot)(unit velocity)(millimeter of mercury vapor-pressure difference).

10 = $K_e \times A$ for average size man, fully wetted.

P_s = vapor pressure of water at skin temperature, millimeters of mercury.

P_a = partial pressure of water vapor in ambient air, millimeters of mercury.

A combination of Equations 4 and 5 yields Equation 6 which is a statement of the maximum thermal conditions under which man can maintain heat balance, giving proper recognition to internal heat production M, skin temperature t_s and to all four of the thermal characteristics of the environment:

$$M + 22(t_a - t_s) + 2\sqrt{V}(t_a - t_s) = 10V^{0.4}(P_s - P_a) \quad (6)$$

Equation 6 provides a rational basis on which to construct a heat stress index and to establish thermal standards for different industrial work situations.

Two physiological criteria are important in fixing the limit of sustained heat exposure:

1. The increase in body heat content must not exceed a certain level. Practically, this is reflected in the rise in skin temperature. This temperature may be limited to 95 F.

2. Thermal balance must be accomplished with a rate of sweating not greater than one liter per hour (equivalent to evaporative cooling of 2400 Btu per hr). Studies on young acclimatized subjects have shown that this maximum rate of sweating can be maintained over a workday without undue strain or physiological cost.

The Heat Stress Index (HSI) proposed by Belding and Hatch[1] makes use of these criteria together with the previous biophysical equations. A so-called *standard* young man, in good physical condition and acclimatized to heat, can safely engage in simple physical work over a normal eight-hour day in a particular thermal environment so long as his skin temperature does not exceed 95 F and he maintains thermal balance with a sweat rate not exceeding 1 liter per hr. By definition, this degree of stress has an (HSI) value of 100 and is given by any combination of $M + R + C$ which is balanced by E_{max} within their defined limits. For other situations where E_{req} is greater or less than E_{max} ($\leq$2400 Btu per hr), the value of (HSI) is proportionately different from 100:

$$(HSI) = 100 \times \frac{E_{req}}{E_{max} \leq 2400} \quad (7)$$

By using the globe thermometer temperature (see Chapter 52) of the environment to represent the combined effects of mean radiant temperature and air temperature, the Heat Stress Index chart Fig. 1 has been constructed, to permit quick determination of (HSI) when M, t_s, t_a, t_{wb}, and air velocity are given. The procedure for finding (HSI) will be obvious by following the dash line in Fig. 1.

The Heat Stress Index was developed from research experience with young men in good physical condition and well acclimatized to heat. Table 2 assigns suggested physiological meaning to (HSI) values above and below 100, for older men as well as the standard young man and, also, for various kinds of jobs which differ in their demands upon mental and physical effort. From these it becomes possible to select (HSI) levels appropriate to each work situation.

The Heat Stress Index (HSI) differs from the Effective Temperature (ET) scale in the rational rather than subjective basis of its construction. Practically, the commonly recommended ET = 80 F for heavy work at moderately high air velocity agrees very well with the (HSI) limit of 100 for acclimatized young men. In Fig. 2 are shown the combinations of dry- and wet-bulb temperature (air and wall temperature alike) for three different air velocities and three

Table 2 .... Evaluation of Index of Heat Stress

Index of Heat Stress	Physiological and Hygienic Implications of 8-Hr Exposures to Various Heat Stresses
−20	Mild cold strain. This condition frequently exists in
−10	areas where men recover from exposure to heat.
0	No thermal strain.
+10	Mild to moderate heat strain. Where a job involves
20	higher intellectual functions, dexterity, or alert-
30	ness, subtle to substantial decrements in performance may be expected. In performance of heavy physical work, little decrement expected unless ability of individuals to perform such work under no thermal stress is marginal.
40	Severe heat strain, involving a threat to health unless
50	men are physically fit. Break-in period required
60	for men not previously acclimatized. Some decrement in performance of physical work is to be expected. Medical selection of personnel desirable because these conditions are unsuitable for those with cardiovascular or respiratory impairment or with chronic dermatitis. These working conditions are also unsuitable for activities requiring sustained mental effort.
70	Very severe heat strain. Only a small percentage of the
80	population may be expected to qualify for this work.
90	Personnel should be selected (*a*) by medical examination and (*b*) by trial on the job (after acclimatization). Special measures are needed to assure adequate water and salt intake. Amelioration of working conditions by any feasible means is highly desirable, and may be expected to decrease the health hazard while increasing efficiency on the job. Slight "indisposition" which in most jobs would be insufficient to affect performance may render workers unfit for this exposure.
100	The maximum strain tolerated daily by fit, acclimatized young men.

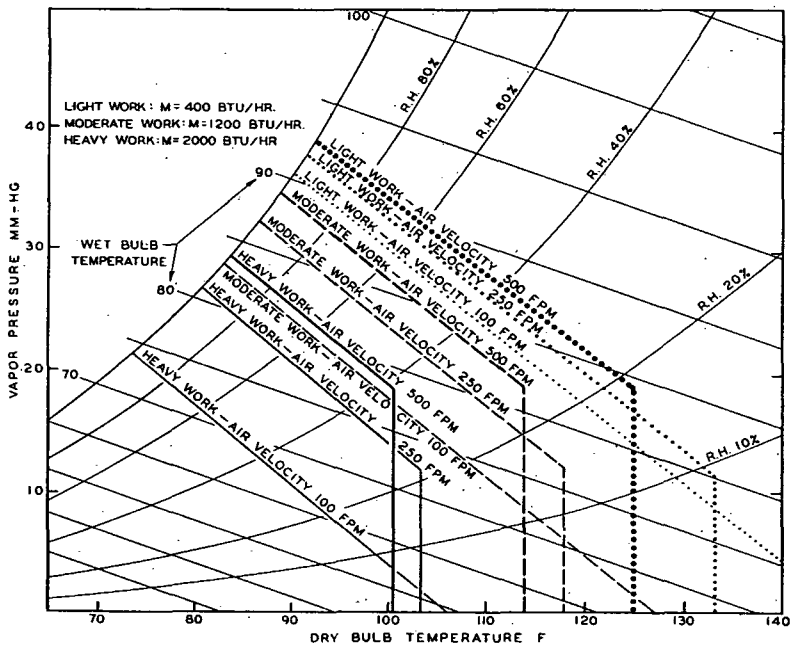


Fig. 2 .... Conditions for Stress 100 at Three Grades of Work and Three Air Velocities

different grades of work, all of which give (HSI) = 100. Points representing ET = 80 F with an air velocity of 300 fpm are also shown for comparison.

CONTROL OF HEAT EXPOSURES

Measures for control of heat exposures will be treated under the following headings: Control at Source, Local Exhaust Ventilation, Radiation Shielding, General Ventilation, Dilution Ventilation, Local Relief Ventilation.

Control at Source

The magnitude of heat exposure can be reduced by insulating hot equipment, locating such equipment most favorably (in zones of good general ventilation within buildings or even outdoors), covering steaming water tanks, providing covered drains for direct removal of hot water, and maintaining tight joints and valves where steam may escape. This method is an obvious one and requires no particular comment other than to emphasize the benefits to be derived from elimination of heat sources wherever possible.

Local Exhaust Ventilation

The natural convection column of heated air rising from a hot process may be captured by means of ventilated enclosures or exhaust hoods and removed with a minimum of dilution by air from the surrounding space when local exhaust ventilation can be used. Chapter 52 provides detailed information on the design of exhaust hoods and duct systems for local exhaust.

Radiation Shielding

In some industries, commonly referred to as the hot industries, there are many hot objects and surfaces such as furnaces, ovens, furnace flues and stacks, boilers, molten material, and hot ingots of metal, castings, or forgings and, in consequence, the major environmental heat load is in the form of radiant heat. Since air temperature has no significant influence on the flow of radiant heat, ventilation is of no help in controlling such exposures. The only effective control is the direct one of decreasing the amount of radiant heat im-

pinging on the exposed worker. This frequently can be accomplished by radiation shielding. Radiant heat exposures can be reduced by lowering the surface temperature of hot equipment by using insulation or water-cooled surfaces or radiation shields.

Hot surfaces emit infra-red waves which reach all objects within visible range. This is true not only if the hot surface is a primary one such as a furnace wall, a high temperature stack, or a hot ingot, but also if it is secondary in nature, that is, one which is reradiating heat after receiving it by direct radiation from a high temperature source. It may well be that all surfaces in a shop or building are hot because of radiant heat received from a few primary sources. Under these circumstances, the ventilation air is heated rapidly as it circulates through the building. The result is a very unsatisfactory thermal environment, all stemming from the few primary heat sources. Consequently if the radiant heat load *from these is reduced there will be a great improvement* in the entire building.

A shield is simply a sheet of material, opaque to the infra-red waves (or essentially so), placed between the hot object such as a furnace, and the cooler surroundings. The closer it can be placed to the hot object, the greater will be the effective coverage. Essential to most shielding is the absence of physical contact and the presence of an air space between the hot object and the shield to avoid heating of the shield by conduction. This is not a vital matter if the shielding material has a low coefficient of emission for infra-red rays as does aluminum, for example. Under such circumstances, very little heat is radiated even if the shield becomes very hot.

Radiation shields in the following 5 forms are very effective:

1. Sheets of reflective metal or insulating board, semi-permanently attached to the hot equipment (such as furnace buck stays) or arranged as semi-portable floorstands.
2. Aluminum foil-faced cloth curtains raised or lowered on spring rolls.
3. Transparent shields, including heat reflective tempered plate glass, reflective metal chain curtains, and close mesh wire screens. These have lower efficiency than opaque shields in items 1 and 2.
4. Water-cooled shields for absorbing the heat.
5. Reflective garments, such as aprons, or in the form of a sandwich in cases of continuous front and back exposures. For continuous wear the apron or sandwich width should be limited to from 14 to 18 in. in order to assure adequate continuous side openings for air circulation and body ventilation. In addition, gauntlets and face shields are particularly applicable to operations such as the pouring and casting of hot metal. Supply houses dealing in safety clothing for industry offer suitable light-weight flame proofed foil-faced cotton drill or denim of excellent reflectivity. For repairs inside hot coke ovens and industrial furnaces, a complete suit is available using forced ventilation from a small blower or a compressed air source. These suits may be made of asbestos cloth faced with a reflective metal in atomized form.

If the shield is a good reflector, it will remain relatively cool in the presence of severe radiant heat. This reflectivity is a surface characteristic and is not dependent on thickness. A thin foil is effective. Tinplate, stainless steel, and ordinary flat or corrugated aluminum sheets are efficient and long-lived. Foil-faced plaster board, though less durable, gives good reflectivity on one side. Since the best radiation shields are effective reflectors of infra-red rays, they must be used intelligently lest a radiant heat load merely be transferred from one place to another. The objective is to have the shield reflect the radiant heat back to the primary source where it may be removed by local exhaust. However, unless the shield completely surrounds the primary source, some of the infra-red energy will be reflected into the cooler surroundings and possibly into an occupied area. It is imperative, therefore, to study well where the reflected heat will go before shielding is installed, to avoid merely getting rid of the problem in one area by transferring it to another.

GENERAL VENTILATION

General ventilation, which involves sweeping of the space occupied by workmen and process heat sources with large quantities of outdoor air, may be used to limit the temperature rise within the space, if the outdoor air itself is not too hot. The method is particularly suitable for cases where the heat sources are spread over the entire area, and for the removal of solar heat from the space.

Dilution Ventilation

The principles of dilution ventilation furnish the basis for the design of the general ventilation system. The amount of air which must be circulated can be estimated as:

$$(\text{cfm}) = \frac{H}{1.08\Delta t} \tag{8}$$

where

H = Heat to be removed from the space, Btu per hour.
Δt = temperature rise of the air, Fahrenheit degrees.

All heat sources such as furnaces, electric motors, ovens, and kilowatts of lighting should be tabulated to give the total heat output in Btu per hour (See Cooling Load Calculations, Chapter 13). The amount of air to be exhausted is a function of the total amount of heat and the permissible air temperature rise which, in practice, may vary from 10 up to 40 deg, depending on many factors such as roof height, nature of work being done, and magnitude of heat release.

Advantage should be taken of the chimney effect of the heated air within the space by introducing the diluting air near the floor of the space and discharging the heated air as near the top of the structure as possible, regardless of whether natural or mechanical exhaust ventilation is used. Since the workmen ordinarily operate near floor level, the entering air temperature should be within acceptable limits as outlined earlier in this section. The leaving air temperature will have its effect on workmen through the radiant heat emitted by the upper parts of the structure which are heated by the air. A higher leaving air temperature can be used with a high than with a low ceiling, since the ceiling contribution to the mean radiant temperature will be less for the taller structure.

Roof Ventilators

Roof ventilators are basically heat escape ports located in the high section of a building and properly enclosed for weather-tightness. Stack-draft effect plus some wind induction are the motive forces for several common designs of continuous ventilators and round ventilators. The latter can be equipped with fan barrel and motor, thus permitting gravity operation, or motorized high capacity operation, at will.

Two other main designs are available: one is the low-type ventilator which consists essentially of a stack fan with a rainhood; the other contains a stack fan with a split butterfly closure that is floated to open position by the discharge air stream and is self-closing. Both employ mimimum enclosures and have little or no gravity capacity.

Various types of roof ventilators may be listed in diminishing order of temperature differential and heat-removal capacity. Next to chimneys and stacks, the continuous ventilation monitor is most effective for removal of substantial and concentrated heat loads. This may be in the form of an elevated length of roof ridge with extended overhang and no rain louvers. Such structures are commonly employed over open hearth furnaces and soaking pit buildings but, in practice they give uncertain performance. A more efficient type is a streamlined, watertight monitor constructed of noncorrosive metals or protected metals. Its capacity increases with wind velocity and it can be readily closed in winter to conserve building heat. Both types have tremendous capacity and are limited only by roof area and proper relation of low-level air inlets.

Next in capacity are the (1) round gravity or windband ventilator, (2) round gravity type with fan and motor added, (3) low-hood powered ventilator, and (4) vertical upblast powered ventilator. Popular modifications of these include the rotary type and the turbine ventilator.

Minimum-capacity ventilators of the gravity type are applicable to warehouses with light heat loads, and to manufacturing areas having high roofs and light loads. Non-mechanical ventilators are desired here because the wide spacing makes electrical wiring and maintenance of fans and motors a matter of concern.

Streamlined continuous ventilators must be able to operate effectively in the absence of mechanical power. While it is difficult to predict performance accurately, it can safely be said that efficient ventilator operation is generally obtained when difference in elevation between the average air inlet level and the roof ventilation is not less than 30 ft and the exit temperature is 25 F deg above the prevailing outdoor temperature. See Chapter 11, Infiltration and Ventilation. Under these conditions and with a wind velocity of 5 mph, the ventilator throat velocity will be about 375 fpm. It will thus remove 10,000 Btu (per hr) (sq ft of throat area):

$$25 \text{ deg} \times 1.08 \times 375 \text{ cfm} = 10{,}000 \text{ Btu}$$

To insure this level of performance it is essential that sufficient low-level openings be provided for the incoming air; otherwise the gravity ventilator becomes starved for air and capacity falls off. The manufacturers recommend 250 fpm to 450 fpm inlet velocity. Lack of adequate inlet area is the most common cause of failure of gravity-type roof ventilators. A positive supply of air to insure direct ventilation around the hot equipment may be necessary within buildings of considerable area where the external wall inlets are remote from the equipment.

A decision between gravity and mechanical ventilators may be difficult to reach. The electrical power required for fan operation may be a deterring factor but this is offset by the advantage of constant ventilation capacity given by fan operation. Mechanical ventilators will also function despite inadequate fresh air inlets. In many cases there is justification for a mechanical supply (make-up air) which will positively supply air to the work zone.

LOCAL RELIEF

General

The methods required for general ventilation of industrial areas have been described. Such ventilation may have to be supplemented, or in some cases replaced by local ventilation in large industrial areas having substantial heat loads, high ceilings, and scattered work stations. Provision must be made for local relief, to provide tolerable working conditions around the several operations where the heat loads are concentrated. The methods indicated to be described for local relief are based on the assumption that exhaust ventilation, radiation shielding, equipment insulation, and possible changes in process design have been fully utilized to minimize the heat loads.

Methods of Providing Relief

Relief in local industrial work areas may be provided by any of the following three methods (or a combination of the three):

1. Provide a complete enclosure around the worker with separate ventilation in order to maintain cooler working conditions. This may be in the form of a control room, small shelter booth, or ventilated crane cab. In effect, this is "localized" general ventilation, differing only in the conditions of air temperature, humidity, and motion required.
2. Surround the worker with a relatively cool atmosphere by a direct supply of air introduced at a low level over a small area of the plant. In such cases we are not concerned with the temperature at higher levels in the space so long as adjacent structures do not become hot and thus return radiant heat into the work space.
3. Direct a high-velocity air stream at the worker to increase the evaporative cooling effect. This method, commonly called *spot cooling* will incorporate varying degrees of Method 2, depending upon the number of employees and the distribution of the work stations.

Method Number 1, in the form of a work station enclosure, is the most desirable because it permits complete control of the environment. The second method is effective in large areas with many work stations, such as machine shops and assembly lines. The last method is used to provide *spot cooling* in large spaces where scattered work stations and localized sources of heat make it highly impracticable to maintain a tolerable environment throughout the building.

Physiological Aspects of System Design

Two different situations in respect to heat load are encountered: (1) where radiant heat sources are not important, and (2) where radiant heat sources are important. These two situations must be treated differently in design.

Where Radiant Heat is Not Important. Where no important sources of radiant heat are located within or close to the work area, *it is only necessary to introduce the relief air* into the work space in such a way as to displace the hot air and thus surround the worker with an acceptable atmosphere, consistent with his own needs.

Where Radiant Heat is Important. Where important sources of radiant heat are present and cannot be entirely controlled by radiation shielding, the air supplied for relief must do more than simply displace the hot air. It must also provide convective cooling to offset the radiant heat load. This requires that the air be supplied at a lower temperature and with sufficient velocity. Contrariwise, if the temperature of the relief air exceeds skin temperature, a convective load is added to the radiative load and these, together with the metabolic heat must be removed by evaporative cooling. As the temperature of the air impinging on the worker rises above the skin temperature, a critical point is approached where too high an air velocity will add to the body heat load and too low a velocity will fail to evaporate the sweat, and thus fail to provide the maximum evaporative cooling effect.[4, 5]

The level of air motion provided at the work station should be low, approaching normal room velocities, or about

50 fpm for light activity in a fixed position, and particularly when the individual is seated at a desk or bench. The maximum velocity around the worker should not exceed 200 fpm for continuous exposure. With a high work level and intermittent exposure (relief stations), velocities of 1000 to 2000 fpm have been used successfully. Great care must be exercised in using such high velocities, however, to avoid the undesirable effects of air at high temperature.

Ordinarily the temperature of the air supplied to the worker for convective cooling should be about 80 F. Evaporative cooling ventilation systems will usually provide this. Preferably, the air should be directed to the front and torso of the body. Impingement on the back of the head, neck, and shoulders should be minimized. The supplied air may require tempering in winter to raise it to an acceptable temperature. However, in areas of high heat load, the temperature of the air supplied during the winter may have to be lower than room temperature to provide adequate relief for the worker.

People vary considerably in their tolerance to air motion, temperature, and humidity, and this tolerance varies with the season. It is essential, therefore, that the air supply outlets for most local relief ventilation systems be adjustable in direction, and permit reduction in outlet velocity. A level of air motion which feels comfortable and refreshing in hot weather may feel disagreeable and drafty in the winter.

Types and Design Requirements of Systems

The supply air can be provided by local man-cooling fans; by outdoor air introduced directly or after dehumidification or cooling (evaporative or mechanical); and by combinations of outdoor and recirculated air either direct or after conditioning, as required.

Local man-cooling fans should be used with caution other than in light heat-load areas where the ambient temperature is below the skin temperature. In particular, where there is an elevated ambient temperature with or without high radiation load, the high velocity may add considerably to the convective heat load and thus seriously increase the demand for sweating and evaporative cooling.

A relief system employing outdoor air is to be preferred over man-coolers and will provide excellent relief in many industrial areas. However, when the outdoor air temperature exceeds the skin temperature, the direct supply of outdoor air is obviously reduced in effectiveness. Such a system is properly used only in geographical areas where the periods of hot weather are of short duration.

In geographic areas where outdoor humidities do not exceed 50 percent relative humidity at high daytime temperatures an evaporative cooling system offers greater relief for the worker in that the discharge air temperature can be lowered (within five degrees of the wet-bulb temperature) to obtain adequate convective body cooling. The sensible heat gain by the discharge air (through mixing) in a properly designed system will be sufficient to reduce the relative humidity (and vapor pressure) of the supply air and thus permit the necessary evaporative loss from the worker's skin. Generally the supply of evaporatively cooled air should not exceed 20 to 50 percent of the total ventilation through the building. So limited, the relative humidity will be kept low enough to avoid distress to the workers and rusting of equipment under most weather conditions in the United States. For information on the design of evaporative cooling systems, see References 5 and 6 and Chapter 41.

An air supply with mechanical refrigeration offers the greatest relief. Such systems are expensive in first cost and this has retarded their use. However, air supply with mechanical refrigeration is finding greatly increased use in precision work and testing areas, in areas requiring constant ambient conditions for product uniformity or control, and where the increased efficiency of the worker is reflected in a reasonable return on the investment.

It is essential that the outdoor air, (with or without cooling) be brought as directly as feasible to the proximity of the work station. Impingement upon hot equipment or mixing with hot ambient air should be minimized. Supply ducts passing through hot areas should be insulated. The use of aluminum ducts will minimize heat gain from radiant sources. For large work areas provided with a high rate of ventilation the local relief can best be obtained by locating the outlets as close to the floor as possible, consistent with the work and structural interferences.

Design of Outlets for Local Relief

In the design of outlets for local relief it is of great importance to consider the following: location; discharge velocity, discharge volume, and control of direction. The influence of these items will be discussed in the paragraphs which follow.

Location. The outlets should be kept as close to the worker as possible in order to minimize mixing with the warmer air in the space. In most areas the outlets can and should be brought down to the 7-ft level. Outlets at floor level can be used with success in many cases.

Discharge Velocity. The discharge velocity may be as high as necessary to obtain the desired velocity at the work station, as outlined above. Outlet velocities of 3000 to 4000 fpm may be necessary for remotely located outlets. Velocities of 1000 to 2000 fpm are the most frequently used for low outlets (at the 7-ft level). When the supply air is cooled, the velocity through an outlet directly at or over the worker must be kept low (around 50 fpm). It should be kept in mind that these recommended velocities are for conditions of maximum heat load. For more moderate weather and ambient conditions the workers will desire to reduce the velocities. Outlet dampers for velocity control (in the direction of the worker) should always be provided. The control must be designed so as not to reduce the ventilation for the space.

Discharge Volume. The outlet volume required will vary widely, depending upon whether the system is designed to provide highly localized *spot cooling* or is to provide general ventilation throughout a sizeable work area. Generally 1500 to 2000 cfm per station will be adequate for moderate loads, and 3000 cfm per station for higher loads such as at hot metal furnace stations. With remote outlets large air volumes are required to insure adequate relief because of the mixing of the supply air with the warmer surrounding air through which it is projected. The air stream from a large outlet will maintain an appreciable core of air at the original supply air temperature for a considerable distance from the outlet. Small outlets and slot outlets have small cores which are rapidly dissipated through induction. In small enclosures or semi-enclosures (shields against radiant heat, for instance) perforated panel supply outlets are very effective for mechanically cooled installations, because of their low induction characteristics. For further information on outlet design and the throw and induction characteristics, see Chapter 20.

Control of Direction. With very few exceptions, directional outlets are necessary in order to maintain ventilation or make-up air to the space even when the workers do not wish to have the air directed into their work zones. In this way, the air can be directed down in the summer when it is needed, and up in the winter for ventilation, heating, and make-up purposes. It will be possible, too, to direct the air toward one or another work station as needed. From the production standpoint, directional control may be necessary to direct the relief ventilation so as not to disturb the product or upset the performance of local exhaust hoods.

Types of Outlets for Local Relief

A few of the different types of outlets are shown in Fig. 3. In most cases, volume control is highly desirable. Directionalizing and damper control should be designed for easy adjustment from the floor by chain, sash cord, or pole. Where

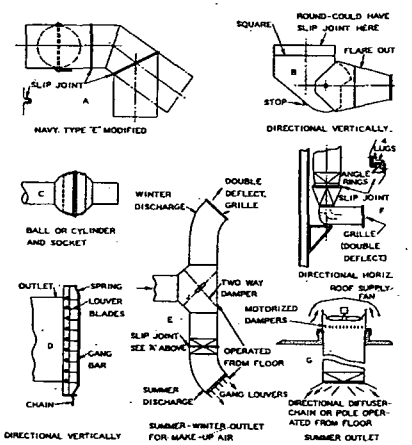


Fig. 3 .... Directional Outlets for Spot Cooling

a constant total volume of supply make-up air must be maintained, an outlet similar to *E* will permit volume adjustment to the *work zone* without reducing the total air supply to the building area. A double deflection grille is available with gang-operated, horizontal face bars (similar to *D*) to accomplish the desired objective by directing the air down to the work zone or toward the ceiling. Such a grille could be installed on outlet *F*. Outlets *A*, *B*, and *C* could be used without dampers for these applications. An inexpensive means of supplying summer relief air is shown in *G*. Easily-adjusted diffusers, which will provide a flat or down-blast air discharge pattern, are available.

Where overhead installations are not possible because of structural interferences or other difficulty, outlets near the floor may be used. These may be in the form of grilles located in pilasters, control panels, tables, equipment, or in the floor itself. Such outlets have been quite successful in welding and foundry areas and at the work locations in kitchens.

CONTROL OF ENVIRONMENTAL GASES AND VAPORS, DUSTS, AND FUMES

In general, the systems for the control of these atmospheric contaminants in industrial plants will be of three types:

1. *Local exhaust systems* will be indicated where the contamination originates at concentrated areas and is characterized by low or imperceptible air motion, or where the contaminant is a dust, mist, or fume requiring a capture velocity exceeding 25 fpm. Design of this type system is discussed in Chapter 52, and will not be further treated in this chapter.

2. A system employing the *dilution* method will usually be indicated where the contamination originates at scattered points dispersed generally throughout the area.

3. Combination of local *exhaust* and *dilution* methods is often economical, since well designed exhaust hoods or openings, removing from the space that portion of the contamination load which is susceptible to such treatment, will often reduce greatly the air volumes required for dilution purposes. The choice of the type of system should be made on the basis of economic comparisons.

Design of Dilution Systems

The first step in the design of a system employing the dilution method is to determine as exactly as possible the nature and extent of the contaminating load. This will often be difficult, and may require construction of pilot production models. Often, however, the required data will be available from production records, showing the weight or volume rate of loss of the contaminating agent to the atmosphere, or it may be estimated from parallel operations in other plants, or by applying experienced engineering judgment. However obtained, the determination of the nature and magnitude of the contaminating load is an indispensable step in the proper design of the corrective system. Designs based on number of air changes per hour, or other rule-of-thumb methods, are hopelessly inadequate, and lead either to unsuccessful operation or to excessive and unnecessarily high cost of installation.

1. *Gases and Vapors.* Once having established the nature and magnitude of the contamination load, it is rarely necessary to completely remove contaminating agents from the atmosphere. For cases involving diffusible vapor or gas contaminants, maximum allowable concentrations (MAC) of commonly encountered gases and vapors have been established, and these data are tabulated in Chapter 7. From these data, and the previously established rate of addition of the contaminant to the space, the volume of air required to dilute the addition to a tolerable level can be calculated by the equation:

$$Q_s = \frac{Q_c \times 10^6}{(\text{MAC}) - (\text{SAC})} \qquad (9)$$

where

Q_s = quantity of air circulated, cubic feet per minute.
Q_c = rate of generation of contaminant, cubic feet per minute.
(MAC) = maximum allowable concentration, ppm by volume.
(SAC) = concentration in supply air, ppm by volume.

The rate of generation of the contaminating vapor will often be available as a *weight or volume* of liquid evaporated into the space per unit time. These may be converted to the units of Equation 9 by applying the principle that a pound-mol of a

gas or vapor will occupy approximately 359 cu ft at standard pressure and temperature. Thus,

$$\text{cfm (vapor)} = \frac{W}{M_w} \times 359 \times \frac{t + 460}{492} \qquad (10)$$

where

W = rate of generation of contaminant, pounds of liquid solvent per minute.
M_w = molecular weight.
t = air temperature, Fahrenheit.

A special case occurs where local concentrations of solvent vapors *at the breathing zone*, resulting from concentrated sources of contamination, are intolerably higher than the average design concentration when using dilution methods. Data are available for calculations, but involve many assumptions regarding boundary conditions, such as convection area and random air movement in the vicinity.

2. *Dusts and Fumes.* Maximum allowable concentration of various dusts, fumes, and mists are also tabulated in Chapter 7. However, the dilution method as a means of treating particulate contaminating agents should be used with care, since the allowable air movement in spaces will ordinarily be lower than the capture velocity required for such particles. Exhausting at the source (see Chapter 52) will generally be the recommended treatment for these particulate contaminants.

REFERENCES

[1] *Encyclopedia of Instrumentation for Industrial Hygiene* (Publications Distribution Service, University of Michigan).

[2] *Industrial Ventilation* (American Conference of Governmental Industrial Hygienists).

[3] H. S. Belding and T. F. Hatch: Index for evaluating heat stress in terms of resulting physiological strains (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 213).

[4] G. F. Haines, Jr. and T. F. Hatch: Industrial heat exposure-evaluation (*Heating and Ventilating*, November 1952, p. 93).

[5] B. R. Small: Heat relief in industry (*Iron and Steel Engineers Magazine*, April 1952).

[6] Evaporative cooling—a symposium (ASHAE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, August 1955, p. 141).

CHAPTER 52

INDUSTRIAL EXHAUST SYSTEMS

Elements of Exhaust Systems; Hoods or Enclosures, Capture Velocities, Air Volume, Exterior Hoods, Special Exhaust Requirements, Exhaust of Hot Processes, Induced Air Flow, Duct System Design, Calculations, Construction Specifications, Materials, Details, Air Flow Producing Equipment, Air Cleaning Equipment, Make-up Air, Maintenance of Performance, Materials for Corrosion Resistance

IN industrial plants, some type of exhaust system designed to collect and remove dusts, fumes, mists, vapors, and gases is installed to protect health and safety of industrial personnel, promote worker efficiency, salvage usable material, or improve plant housekeeping. This chapter will not include consideration of systems similar in design, but used to convey heavy loadings (100 or more grains per cu ft) of materials. Definitions of various air contaminants, their particle sizes, maximum allowable concentrations, and upper and lower explosion limits are included in Chapter 7, Air Contaminants.

Exhaust systems are extensively used for control of contaminants from:

1. Mechanical cutting and abrading operations including abrasive blasting and rock cutting.
2. Fuel-burning and exhaust gas producing operations.
3. Molten materials handling operations.
4. Welding, burning, and soldering operations.
5. Fiber handling operations.
6. Volatile and gaseous material handling operations.
7. Chemical processes.

Local exhaust systems should be considered whenever possible rather than general ventilation methods which allow contaminants to be dispersed within the workroom. The former provide more positive control, as well as a great reduction in exhaust volume handled, thereby reducing the cost of air cleaning equipment required.

ELEMENTS OF EXHAUST SYSTEMS

An exhaust system consists of (1) hoods or enclosures at sources of air contamination, (2) branch and main ducts through which an air stream transports the contaminant to air cleaning devices or to the atmosphere, (3) air moving equipment to produce the required air flow into hoods or enclosures, and (4) air cleaning equipment when required. See Chapter 24, Part II, for discussion of types, applications, and principles of operation.

HOODS OR ENCLOSURES [1-10]

The most effective hood or enclosure is one that will require the minimum exhaust volume for effective contaminant control. The design must therefore be based upon a knowledge of the process or operation for which control must be obtained. The more complete the enclosure, the more economical and effective will be the installation.

Many designers give first consideration to a hood completely enclosing the operation and then provide necessary access and working openings. The familiar hoods, such as booths, sidedraft or downdraft hoods (with or without side shields) have been developed from this complete enclosure concept. Openings in hoods are kept to a minimum size and are placed away from the natural path of the contaminant travel when possible. Doors should be provided for inspection and maintenance when needed.

Capture Velocities and Air Volume Exhausted

Only after the hood design has been determined can the exhaust volume requirements be calculated. With enclosures, volumes are calculated from the known open area of the hood and the selected capture or indraft velocity sufficient to prevent outward escape.

Usual capture velocities for typical operations are listed in Table 1 and refer in the case of remote hoods to the air movement required at the zone of air contaminant generation. Required capture velocities for any operation will vary with the magnitude of the air volume handled, with uncontrolled air movement in the area, and often with the location of the process or operation and size of the workroom. Large remote hoods exhausting large air volumes will provide effective control at lower maintained capture velocities than will small remote hoods handling lower exhaust volumes. A hood at one end of a small narrow room with air supply at the opposite end will provide control with a lower capture velocity than that required from the same hood in a large room where no

Table 1 .... Minimum Air Velocities Required at Point of Origin to Capture Contaminant Effectively

Condition of Generation of Contaminant	Minimum Capture Velocity, Fpm	Process
Released without noticeable movement	50–100	Evaporation of vapors, exhaust from pickling, washing, degreasing, plating, welding, etc.
Released with low velocity	100–200	Paint spraying in booth; inspection, sorting, weighing, packaging, low speed (less than 200 fpm) conveyor transfer points, blending, mixing, barrel filling.
Active generation	200–500	Foundry shakeout, high speed (over 200 fpm) conveyor transfer points, crushers, screens.
Released with great force	500–2000	Grinding, tumbling mills, abrasive cleaning.

perceptible air flow will exist except in the immediate area of the hood.

Exterior Hoods

Where enclosure of the process is impracticable, the air flow pattern in front of the hood must be such that capture velocities required to convey the contaminant to the hood opening will be maintained in the area of contaminant generation.

The method for determining, approximately, the quantity of air that must be exhausted from an unobstructed hood, without flanges, to produce design capture velocities at the point or origin, is given in Equation 1:

$$Q = V_X(10X^2 + A) \qquad (1)$$

where

Q = quantity of air exhausted, cubic feet per minute.
V_X = air velocity at X-distance in feet from the hood and and on the center line of the hood, feet per minute.
X = distance along the hood center line, from the face of the hood to the point where the air velocity is V_X feet per minute, feet.
A = area of the hood opening, square feet.

Fig. 1 shows lines of equal velocities (velocity contours) for a rectangular hood opening with a side ratio of one-half. The velocities are expressed as percentages of the velocity at the opening. Studies have established the principle of *similarity of contours* which states that the positions of the velocity contours for any hood (when the contours are expressed in terms of the average velocity at the hood opening) are purely functions of the shape of the hood. Extensive studies[9, 11, 12] have revealed variations in values of such velocity contours for long narrow slots and for hoods with one or more planes shielded against air flow.

For smaller hoods, flanges which are usually 3 to 6 in. wide surrounding the hood opening usually will improve

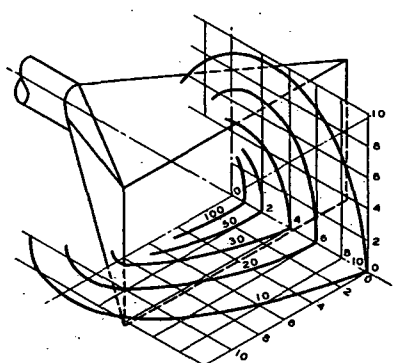


Fig. 1 . . . . Velocity Contours for Rectangular Opening with a Side Ratio of One-Half. Contours are Expressed as Percentages of the Velocity at the Opening

Table 2 . . . . Summary of Exhaust Rates Required with Common Types of Exterior Hoods Applied to Cold Processes Only[13]

X-Distance Inches	*Simple Opening, with or without Flange or Taper, but No Plane Flanking Openings*[a]	*Simple Rectangular Opening Flanked by Plane Parallel to Axis*[b]
	Exhaust Rate Cfm	Exhaust Rate Cfm
2	25–75	25–50
4	50–100	50–75
6	100–200	75–150
8	200–400	150–300
10	300–600	200–400
12	400–800	300–600
15	600–1200	400–800
18	900–1500	600–900
21	1200–1800	700–1000
24	1500–2200	800–1200
30	2500–3500	1200–1800
36	3000–4500	1500–2200
42	4000–6000	2000–3000
48	5000–7000	2500–3500

X-Distance	*Single slots flanked by parallel plane on which contamination process occurs* For *X*-distance (i.e., width of plane) always less than ½ length of slot[c]
Less than 2 ft	Exhaust rate 125 to 150 cfm per sq ft of plane area.
More than 2 ft	Exhaust rate 75 to 125 cfm per sq ft of plane area.

[a] Refers to simple hood opening that performs in accordance with Equation 1.
[b] Refers to hood having a complete flanking plane parallel to axis which prevents flow of air from half of normal air supply zone, and performs in accordance with the equation, $Q = V_X(5X^2 + A)$. Example: rectangular hood opening resting on bench top. Includes single slots, so flanked, for X-distance greater than ½ slot length.
[c] If X-distance is greater than ½ slot length, consider it as a single opening flanked by a plane (see upper right column). With exhaust rate divided between 2 slots, an additional safety factor is provided.

From *Plant and Process Ventilation*, by W. C. L. Hemeon (Industrial Press, New York).

the air flow in front of the hood and will reduce the air volume required to provide desired capture velocities by as much as 25 percent.

The exhaust volume calculated for an exterior hood by different designers may vary greatly due to their selection of different empirical design velocities. Usually capture velocity design values are in the 50 to 200 fpm range.

The exhaust rates calculated for large exterior hoods may become needlessly large if the capture velocity is selected blindly. Whether on not an individual particle or a small wisp of gas or smoke in motion away from the hood is captured before it escapes from the zone of influence created by the hood is determined not only by the specific air velocity (design capture velocity) at the immediate point of contaminant release but also by the depth of the moving air curtain it must traverse before it gets beyond the influence of the hood. The capturing force of the air flow in front of any exhaust hood is the summation of all the separate velocities beyond (away from the hood) the point of contaminant release multiplied by the distance through which each velocity acts; i.e., force

times distance, the force being a function of the velocity. For large hoods, or more accurately for large exhaust rates, the depth of the zone of influence is much greater than for small exhaust rates and, also for the former, the distance between successive velocity contours is greater. Further effectiveness for larger volumes can be obtained where the air supply source produces a *sweeping* flow toward the zone of contaminant release.

Table 2 presents design data by Hemeon[13] applicable to exterior hoods employed for exhausting contaminants originating in a *cold process* only. These values are based on Equation 1 wherein varying values of V_X have been employed according to a scale that takes account of velocity depth factors discussed in the preceding paragraph, and where X-distance is taken as the distance from the hood face to the point where any high velocity air currents engendered by the contaminating process itself have expended their energy. In addition, the ranges of exhaust volumes which have been found satisfactory in practice for many operations are reported in Table 3. These data provide a means of checking the calculated rates for specific applications.

SPECIAL EXHAUST REQUIREMENTS

It is important to note that certain operations may require exhaust volumes in excess of the quantities based on design data from Table 1 and Equation 1. Typical reasons for increased ventilation rates include:

1. Induced air flow caused by the thermal or stack effect from sources of extreme heat.

2. Induced air flow caused by falling granular material in large quantities through considerable height, or by internal rotating parts such as some types of crushers, knives, or macerators. (See discussion in later section Induced Air Flow and Table 4.)

3. Exhaust volumes insufficient to dilute mixtures of combustible vapor and air to less than 20 percent of the lower explosive limit of the combustible.

4. Room air currents caused by cross drafts, spot cooling, motion of machinery or operators. Cross drafts are of great significance in the exhaust of hot processes with high canopy hoods.

5. Design volumes, especially in the case of excess heat control and solvent vapors, may be based on dilution systems where exhaust volumes are selected to keep contaminant concentrations below design levels. See Reference 13, Chapter 7.

6. Local or State regulations may specify larger exhaust volumes for specific operations.

Exhaust volume requirements for many specific operations have been listed in Tables 3, 5, 6, and 7.

Exhaust of Hot Processes

In designing local exhaust hoods for hot processes, it is necessary to estimate the rate of delivery of hot air to the hood by the convection column. The exhaust capacity should exceed this by an amount sufficient to create a velocity in the excess air flowing into the envelope surrounding the convection column that will prevent escape of the heated air at the edges of the hood. Unless distances from the heat source to the receiving hood are small (possibly under 3 ft), the quantity of heated air entering the hood is increased substantially by the induction and turbulent mixing of large quantities of room air as the distance above the heat source increases. Unless the exhaust hood can handle this total volume, spillage of contaminated mixture will occur.

The exhaust rate for satisfactory control of hot processes can be kept at a minimum by enclosing the hot processes as completely as possible and exhausting from the top of the enclosure.

Low canopy hoods rank next to enclosures in economy of air flow. High canopy hoods must handle greatly increased amounts of air over that in the heated stream itself, in order to receive and dispose of the air entrained by the convection column from far below. Where lateral exhaust ventilation must be employed, the air flow toward the hood has to overcome the tendency of the heated air to rise and also must sweep it into the hood. Consequently, the required exhaust air flow is very much larger than for hoods that simply capture the upward flow of heated air. A combination of an exhaust hood with a positive jet of air blown across the top of the equipment and directed into the hood may be required above large pieces of equipment.

Hemeon has suggested[13] design Equations 2 to 6 which follow for various types of hoods.

Equation 2 is suggested for estimating the flow of heated air rising from the top of a hot body:

$$q_s = 29\sqrt[3]{H_s A_p^2 l_h} \qquad (2)$$

where

q_s = air flow rate at upper limits of hot body, cubic feet per minute.
A_p = cross-sectional area of air stream at upper limits of hot body, square feet.
l_h = height of hot body, feet.
H_s = convectional heat transfer rate, Btu per minute.

The area A_p may be approximated from the dimensions of the hot body. For horizontal rods, the width of the stream is practically the diameter of the rod. For vertical planes, the air stream appears to thicken at an angle of 4 to 5 deg with the plane. For horizontal planes, the area of the air stream may be taken as equal to the area of the plane itself.

In the case of horizontal plates, in the absence of better experimental information, l_h may be taken as equal to the horizontal diameter. Chapter 5 furnishes information which may be used in estimating the heat transfer rate H_s.

The flow rate, q_s, of heated air entering the hood after turbulent mixing and dilution have taken place may be approximated for low canopy hoods above horizontal surfaces and where no heat from steam is involved, as

$$q_s = 5.4\, A_s \sqrt[3]{l_h\, (\Delta t)^4} \qquad (3)$$

where

A_s = surface area of hot body, square feet.
Δt = the temperature difference, hot body to room air, Fahrenheit degrees.

Similarly, where the heat is furnished by steam from a tank of hot water,

$$q_s = 290\, A_s \sqrt[3]{l_h\, G} \qquad (4)$$

where

G = the rate of steam formation, pounds per (square foot water surface) (minute).

(*Continued on p. 707*)

Table 3 .... A Guide to Ventilation Rates for Typical Industrial Equipment

State or Local Regulations Should be Consulted and Followed Where Higher Ventilation Rates are Specified

Operation	Ventilation		Usual Transport Velocity Fpm	Remarks and References
	Type of Hood	Air Flow		
Abrasive blast rooms (sand, grit, or shot)	Tight enclosure with air inlets (usually in roof)	60–100 fpm downdraft (long rooms of tunnel proportions 100 fpm crossdraft)	3500	Ref. 8. Many codes specify minimum downdraft of 80 fpm
Abrasive blast cabinets	Tight enclosure with access openings	20 air changes per minute but not less than 500 fpm through all openings	3500	Ref. 14
Asbestos: Carding Spool winding	Enclosure Local Hoods	800 cfm per machine 50 cfm per spool	3000 3000	Ref. 2, 16, 17. See references for details and other operations
Bagging: Open Bag Top	Booth or enclosure (provide spillage hopper)	Paper bags—100 cfm per sq ft open area Cloth bags—200 cfm per sq ft open area	3500 3500	Ref. 18
Barrels—Drums (filling or removing material by scoop)	Local Hood Booth	100 cfm sq ft of container cross-section 100 fpm at face	3500 3500	Ref. 15, 19. Hood with 1-in. slot extending 120 to 180 deg container. Does not confine spillage. Confine spillage—also recommended for container upsetting.
Belt conveyors	Hood at transfer point	Belt speeds less than 200 fpm-350 cfm per foot of belt width, but not less than 150 fpm through open area. Belt speeds over 200 fpm—500 cfm per foot of belt width but not less than 200 *fpm through open area*	3500	Ref. 18, 20
Bins (closed top)	Connect to bin top away from feed point	150–200 fpm through open area at feed points	3500	Ref. 19, 20
Bucket elevators	Tight casing required	10 cfm per sq ft of elevator casing cross-section	3500	Ref. 19
Brick cutting and sizing (abrasive cut-off wheel used dry)	Local Hood Booth with saw at face of booth	500 cfm 150 fpm at face	3500 3500	Ref. 7, 19. For portable operations. Control not as effective as booth
Ceramics Dry pan Dry press Aerographing Fettling, brushing, sagger filling, and unload	 Enclosure Local at die Local at die At supply bin Booth Downdraft or side hood	 200 fpm through all openings 500 cfm 500 cfm 500 cfm 100 fpm (face) 100–150 cfm per sq ft of plan area of dust producing operation	 3500 3500 3500 3500 ... 3500	 Ref. 19, 21, 22 Automatic feed *Manual feed* Manual feed
Cooling Tunnels (foundry molds)	Enclosure	75–100 cfm per running foot of enclosure		Ref. 20
Crushers and grinders	Enclosure	200 fpm *through openings*	3500	*Ref. 18, 23*
Furnaces Stationary melting pots for nonferrous Tilting or rocking melting for non-ferrous Electric Arc for Steel	 Enclosure Canopy Hood attached to roof ring	 100–200 fpm at hood opening 3000–6000 cfm 2500 cfm per ton charged	 1500–2000 1500–2000 2500–3500	Ref. 11, 15, 22, 24, 25 Use higher ventilation rate for toxic fumes Ventilation rate can be materially reduced with more complete enclosures General ventilation of melting room may be substituted but involves greatly increased ventilation rate

Table 3 .... A Guide to Ventilation Rates for Typical Industrial Equipment (*Continued*)

State or Local Regulations Should be Consulted and Followed Where Higher Ventilation Rates are Specified

Operation	Ventilation		Usual Transport Velocity Fpm	Remarks and References
	Type of Hood	Air Flow		
Furnaces Forge (hand)	 Canopy	 200 fpm at face	 1500	 Ref. 24
Garage (tail pipe at servicing location)	Local hood slipped over tail pipe	100 cfm thru 3-in. flexible duct for autos up to 200 hp 200 cfm thru 4-in. flexible duct for trucks and autos above 200 hp 400 cfm thru 4½-in. flexible duct for Diesel engines	2000	Ref. 15
Granite cutting and finishing Pneumatic hand tools *Surfacing machine*	 Local hood Local hood	 500 cfm 500 cfm for tools up to 2⅜-in. diameter 1000 cfm for 2⅜- to 2⅞-in. diameter	 5500–6000 5500–6000	Ref. 26, 27 Typical hood 3 x 8 in. opening with 3-in. flange Hood surrounds tool
Grinders Polishers, buffers, etc. Portable Swing frame	 Standard wheel hood Downdraft bench Booth Booth	 See Table 5 Bench type, 2–400 cfm per sq ft of exhaust grille but not less than 150 cfm per sq ft of plan working area. 100 fpm at face 100–200 fpm indraft through opening in booth face	 3500 3500 3000	Ref. 22, 28 Ref. 15, 19, 22, 29 Recommended for larger parts (Usual cfm per grinder 2000–4000)
Kitchen range	Canopy	100 fpm at hood face	1500–1800	Provide drip gutter on all interior vertical walls to catch condensed grease. Ref. 30
Laboratory hood (provide with door)	Booth type	50–100 fpm		Ref. 24, 31. Air supply location very critical
Metalizing	Local hood Booth	200 fpm at hood face 125–200 fpm at booth face	3500 3000	Ref. 15. Not recommended for toxic materials Use higher ventilation rates for toxic materials
Mixers	Enclosure	100–200 fpm through feed and inspection openings	3000–3500	Ref. 19, 22
Motion picture projector (carbon arc)	Enclosure	12–100 cfm (for fume and gas removal) directly exhausted from projector housing		Ref. 31. 1000–1500 cfm or 20–30 air changes per hour for heat removal
Pharmaceuticals Blenders Coating pans Centrifuges Hammer mills Oscillators Shakers Mixers Process kettles and tanks	 Fully enclosed Hood Enclosure Local hood Hood Enclosure	 100 to 200 fpm through opening 120 cfm exh 24-in. dia pan 60 cfm supply direct to pan *Differential 60 cfm* 250–300 fpm through opening 200 fmp but not less than 50 cfm Not less than 75 cfm per sq ft of plan area 100–250 fpm through opening or manhole	 2500–3500 3000 1500–2000 2500–3500 1500–2000	 Exhaust required only when filling or emptying Flexible exhaust hood connection extended into pans. *Feed opening covered to* greatest possible extent Centrifuge to be provided with cover for 50–75 percent of opening Hood with slot at each end of mixer Ref. 33. Use higher ventilation rate when contents are being heated

Table 3 . . . . A Guide to Ventilation Rates for Typical Industrial Equipment (*Continued*)

State or Local Regulations Should be Consulted and Followed Where Higher Ventilation Rates are Specified

Operation	Ventilation: Type of Hood	Ventilation: Air Flow	Usual Transport Velocity Fpm	Remarks and References
Pouring hoods Foundry	Side hood	200 to 300 cfm per linear ft of hood with slot velocities of 1500 fpm. Exhaust take-off every 8 to 10 ft	3500	Ref. 20
Rock drilling Dry drilling (rock)	Special trap (see references)	60 cfm—vertical (downward) work 200 cfm—horizontal work	3500	Ref. 26, 27. May vary with size and speed of drill. Wet drilling offers alternate control methods
Rubber calender rolls	Canopy—side panels	75–100 fpm indraft	3500	Ref. 24, 31
Quartz fusing	Booth on bench	150–200 fpm at face		Ref. 34, 35
Screens Vibrating Flat deck	Enclosure	150–200 fpm indraft through hood openings but not less than 25–50 cfm per sq ft of screen area	3500	Ref. 15, 19
Shakeouts Foundry	Enclosure	200 fpm through all openings in enclosure, but not less than 200 cfm per sq ft of grate area	3500	Ref. 20, 22
Spray coating	Booth—operator inside Booth—operator outside Booth—downdraft	100–200 fpm at booth cross-section 150–200 fpm at booth cross-section 100–200 fpm downdraft	1500–2000 1500–2000 1500–2000	Ref. 15 Use higher ventilation rate for small booths 4 sq ft or less
Tanks, open surface	See Table 6			Ref. 36, 37, 38, 39
Tumbling mills Hollow trunnion type	Exhaust connection by manufacturer	Use branch diameter same size as exhaust outlet. For round mills branch dia should be ⅙ dia of mill; for square mills branch dia should be 1 in. plus ⅙ side dimension of mill	3500–5000	Ref. 14
Tumbling mills, drums, cages, barrels	Enclosure	400 fpm through openings but not less than 75 cfm per sq ft plan area	3500	Where equipment is enclosed and dust tight during rotation, enclosure may not be needed if feed and discharge operations can be otherwise controlled
Welding	Local hood with flange Downdraft bench Booth	6 in. from arc—150 cfm 6–9 in. from arc—275 cfm 8–10 in. from arc—425 cfm 10–12 in. from arc—600 cfm 150–250 cfm per sq ft grille area 100 fpm at booth face	2000–4000 2000 2000	Ref. 40
Woodworking	See Table 7			Ref. 22, 41
Miscellaneous Packaging, machines, granulators, enclosed dust producing units Packaging, weighing, container filling, inspection	Complete enclosure Booth Downdraft	100–400 fpm indraft through inspection or working openings, but not less than 25 cfm per sq ft of enclosed plan area 50–150 cfm per sq ft of open face area 75–150 cfm per sq ft of dust producing plan area	3000 3000 3500	Ref. 19

For high canopy hoods, Equation 5 may be used:

$$q_s = 7.4\, l_s^{5/3}\, H_c^{1/3} \qquad (5)$$

where

l_s = effective height, feet.

The effective height l_s may be taken as the actual vertical distance from hood to hot surface plus twice the width of the hot surface.

Where it is necessary to have openings at the top of a hood that is filled with heated air, leakage of the hood contents through these openings may be prevented by using sufficient ventilation to obtain a face velocity V as calculated from the following equation:

$$V = C \sqrt[3]{\frac{l_s H_s}{A_f}} \qquad (6)$$

where

l_s = height of the air column, feet.
A_f = area of the (sharp-edged) openings, square feet.
H_s = sensible heat released to air stream, Btu per minute.
C = a coefficient depending on the excess of temperature inside the hood above room temperature with values as follows:

Temperature excess (F deg)	0–200	200–400	400–600	600–800
Value of C	20	18	16	14

The American Society of Heating and Air-Conditioning Engineers has recognized the need for design criteria for determining exhaust ventilation requirements for hot processes in industry. Research studies on this subject have been in progress since 1952, and the results to date have been published.[42]

Induced Air Flow

Where quantities of individual particles are projected through an air space by gravity or by process forces, volumes of room air in proportion to the momentum of the particles are set in motion with resulting intermixing and flow in the same direction as the particles. Exhausted volumes from hoods or enclosures must be sufficient to include this induced air flow if control is to be effective. Induced air flow should be evaluated from high speed rotating machines including pulverizers, from material handling systems employing high speed belts or involving large tonnages of falling granular material, and from escaping compressed air jets from pneumatic tools. The energy represented by material of various particle sizes and falling heights has been computed by Hemeon.[13] The theoretical equivalent induced air flow resulting from unenclosed air streams is summarized in Table 4.

Where falling streams occur largely within an enclosure such as a storage bin, it should be recognized that much of the air set in motion is recirculated within the bin. Exhaust volumes from the enclosure need only include the extra air induced before the falling material enters the enclosure although the recirculating induced air can cause localized positive pressures and outward leakage if the enclosure is not of airtight construction.

The amount of air induced by falling material is much less if the space through which the material falls is enclosed effectively than if not enclosed, and can be reduced to a minimum by enclosing as well as possible the openings at the feed point of the material.

Table 4 . . . . Induced-Air-Flow Equivalent of the Energy in Falling, Unenclosed Streams of Particles

(For a solids flow rate R of 1 lb per sec and a specific gravity s = 1 referred to water)

Falling Distance, z	Stream Area, sq ft	Air Flow Equivalent, Cfm, to be multiplied by $(R/z)^{1/3}$: Particle Size, Millimeters: 1	2	5	10	20	50	100
3 ft	½	450	350	220	180	150	80	65
	1	750	550	350	300	250	150	100
	2	1200	850	550	450	350	200	170
	4	1900	1400	850	700	600	350	250
	8	3000	2200	1400	1100	900	500	400
	15	4600	3300	2000	1700	1400	800	650
	25	6400	4600	3000	2500	2000	1000	900
	50	10000	7500	4700	4000	3200	1800	1400
6 ft	½			350	300	250	200	150
	1			600	500	350	300	200
	2			900	700	600	400	350
	4			1500	1200	900	700	500
	8			2300	1800	1500	1000	900
	15			3500	2800	2200	1500	1300
	25			5000	4000	3200	2300	1800
	50			8000	6000	5000	3700	3000
12 ft	½			560	450	350	270	220
	1			900	700	600	420	350
	2			1500	1100	900	700	550
	4			2300	1800	1500	1000	850
	8			3600	2900	2300	1700	1300
	15			5300	4200	3500	2600	2100
	25			7800	6000	5000	3600	2900
	50			—	10000	8000	6000	4700
20 ft	½			800	650	500	370	300
	1			1300	1000	800	600	500
	2			2000	1600	1300	960	760
	4			3300	2600	2000	1500	1200
	8			5000	4100	3200	2400	1900
	15			8000	6400	5000	3700	2900
	25			—	9000	7000	5000	4000
30 ft	½			1000	850	650	500	400
	1			1700	1400	1100	800	600
	2			2700	2100	1700	1300	1000
	4			4400	3500	2700	2000	1600
	8			6800	5400	4200	3200	2500
	15			10000	8200	6600	4800	4000
	25			—	—	9000	6800	5400

From *Plant and Process Ventilation* by W. C. L. Hemeon, The Industrial Press New York (1955).

DUCT SYSTEM DESIGN

The duct system will consist of branch ducts connected to a main duct that will convey the air from the hoods to the exhaust fan and the air cleaning equipment, if used. Round ducts should be used wherever possible. Their gage size and construction differ from air supply practice due to the rougher usage encountered, and in the case of dusts, to the abrasive effect. (See later section on Construction Specifications for Local Exhaust Systems.) Usual conveying velocities are shown in Table 8. Where solid contaminants are handled, recommended velocities must be maintained throughout the system to prevent material from settling in the ducts and obstructing

Table 5 . . . . Exhaust Requirements[28] for Grinding, Polishing, Buffing, Scratch Brushing Abrasive Cut-Off Wheels, Grinding and Polishing Belts

(Air Volume: 4500 fpm in all branches.) (2-in. water suction at average hood.)

Grinding or Cut-off Wheels				Buffing, Polishing, and Scratch Brushing			
Wheel (dia), In.	Max. Width[a] In.	Branch (dia) In.	Cfm	Wheel (dia), In.	Max Width[a] In.	Branch (dia), In.	Cfm
–9	1¼	3	225				
10–16	2	4	390	–9	2	3½	300
17–19	3	4½	500	10–16	3	4½	500
20–24	4	5	610	17–19	4	5	610
25–30	5	6	880	20–24	5	5½	740
31–36	6	7	1200	25–30	6	6½	1040

Horizontal Single Spindle Disc Grinders			Horizontal Double Spindle Disc Grinders			Vertical Single Spindle Disc Grinders		
Wheel (dia) In.	Branch (dia) In.	Cfm	Wheel (dia) In.	Branch (dia) In.	Cfm	Wheel (dia) In.	Branch (dia) In.	Cfm
–12	3	225	–19	1–5	610	–20	1–4½	500
13–19	4	390	20–25	1–6	880	21–30	2–4	780
20–30	5	610	26–30	1–7	1200	31–53	2–6	1760
31–36	6	880	31–53	2–6	1760	54–72	2–8	3120
			54–72	4–8	6240			

GRINDING OR POLISHING BELTS:
To 3 in. wide—3 in. branch, add ½ in. to branch diameter for each 2 in. or fraction increase in belt width over 3 in.

[a] Increase branch diameter 0.3 in., but not less than ½ in., for each 1 in. increase in width over dimension noted.

the air flow. For conveying of solid particles, velocities shown in Table 8 include a reasonable factor of safety and, consequently, design velocities may therefore be adjusted as required to permit use of commercial pipe sizes that are available generally in ½-in. diameter steps through 6-in. diameter, 1-in. steps through 16-in. diameter or larger, and 2-in. diameter steps for large diameters. For contaminants other than solids, conveying velocities are based on a consideration of the higher resistance and lower first cost of smaller diameter ducts against higher first cost and lower resistance of larger diameter ducts.

Main ducts connecting two or more branches are sized by the same procedure as for branches. The main should be designed for the total exhaust volume to be handled through all branch ducts.

Where exhaust systems handling solids are to provide for a substantial increase of future capacity, required conveying velocities can be maintained by:

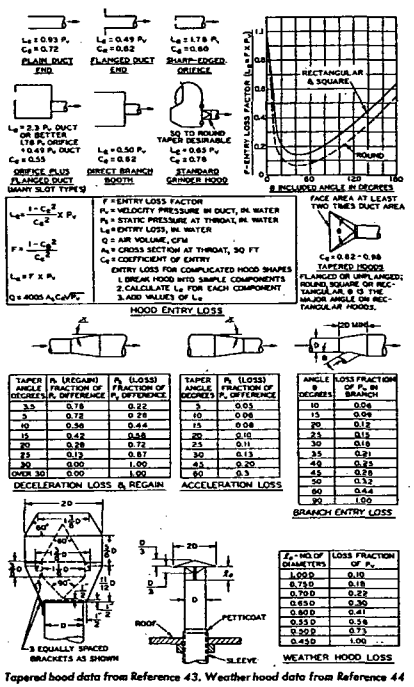
1. Providing open end stub branches in the main through which air will be admitted into system until future connection is made. The volume admitted into the main can be adjusted to required *air flow rate (cfm) by use of a blast gate or orifice* plate in such stub branch.

2. Arranging the system layout so that exhaust from future points can be picked up by a separate main that will run directly to the fan or air cleaning equipment inlet.

CALCULATION OF SYSTEM PRESSURE LOSS

The pressure loss of the system includes entrance loss[44] as air is accelerated through the hood to a branch duct connection, resistance losses of ducts,[44] elbows, and junctions, and acceleration or deceleration losses from velocity changes within the system. Chapter 21 includes data on duct resistance and on resistance of elbows, and provides sample calculations. In exhaust systems, either the equal friction method (illustrated in Chapter 21) or the use of blast gates in the branches to equalize pressure loss of all runs is employed.[9, 22, 23, 29]

Data on losses for hood entrance, branch entry, transitions, and weather hoods are given in Fig. 2. The static pressure in branch ducts close to the hood connection is frequently referred to as *hood suction*. It is the sum of the branch velocity pressure and the hood entrance loss.



Tapered hood data from Reference 43. Weather hood data from Reference 44

Fig. 2 . . . . Exhaust System Design Data[15]

CONSTRUCTION SPECIFICATIONS FOR LOCAL EXHAUST SYSTEMS

Correct design and competent installation of sheet steel ducts and hoods, are necessary for the proper functioning of any exhaust system. The following specifications are those recommended in the *Industrial Ventilation Manual*.[18]

General

All exhaust systems shall be constructed with the materials recommended herewith and shall be installed in a permanent and workmanlike manner. The interior of all ducts shall be smooth and free from obstructions; with joints either welded, flanged, or soldered airtight.

Materials

1. Ducts shall be constructed of block iron welded or of galvanized sheet steel riveted and soldered unless the presence of corrosive gases, vapors, and mists or other conditions make use of such material impracticable. Galvanized construction is not recommended for temperatures exceeding 400 F. Welding of black iron of 18 gage and lighter is not recommended for field fabrication.

2. For average exhaust systems on non-corrosive applications the following metal thickness shall be supplied:

Diameter of Straight Ducts (Inches)	U. S. Standard Gage for Steel Duct		
	Class I	Class II	Class III
Up to 8	24	22	20
Over 8 to 18	22	20	18
Over 18 to 30	20	18	16
Over 30	18	16	14

Table 6 . . . . Ventilation Rates for Open Surface Tanks[26, 27]

Process	Minimum Ventilation Rate Cfm Per Sq Ft Hood Opening: Enclosing Hood: One Open Side	Enclosing Hood: Two Open Sides	Canopy Hood: Three Open Sides	Canopy Hood: Four Open Sides	Minimum Ventilation Rate Cfm Per Sq Ft Tank Area Lateral Exhaust (Note 1) W/L = Tank Width / Tank Length RATIO: W/L 0–0.24 A	W/L 0–0.24 B	W/L 0.25–0.49 A	W/L 0.25–0.49 B	W/L 0.50–1.0 A	W/L 0.50–1.0 B
Plating												
Chromium (Chromic Acid Mist)	75	100	125	175	125	175	150	200	175	225		
Arsenic (Arsine)	65	90	100	150	90	130	110	150	130	170		
Hydrogen Cyanide	75	100	125	175	125	175	150	200	175	225		
Cadmium	75	100	125	175	125	175	150	200	175	225		
Anodizing	75	100	125	175	125	175	150	200	175	225		
Metal Cleaning (Pickling)												
Cold Acid	65	90	100	150	90	130	110	150	130	170		
Hot Acid	75	100	125	175	125	175	150	200	175	225		
Nitric and Sulfuric Acid	75	100	125	175	125	175	150	200	175	225		
Nitric and Hydrofluoric Acid	75	100	125	175	125	175	150	200	175	225		
Metal Cleaning (Degreasing) See Note 2 and Ref. 38 and 39												
Metal Cleaning (Caustic or Electrolytic)												
Not Boiling	65	90	100	150	90	130	110	150	130	170		
Boiling	75	100	125	175	125	175	150	200	175	225		
Bright Dip (Nitric Acid)	75	100	125	175	125	175	150	200	175	225		
Stripping												
Concentrated Nitric Acid	75	100	125	175	125	175	150	200	175	225		
Stripping												
Conc. Nitric and Sulfuric Acid	75	100	125	175	125	175	150	200	175	225		
Salt Baths (Molten Salt)	50	75	75	125	60	90	75	100	90	110		
Salt Solution (Parkerize, Bonderize, etc.)												
Not Boiling	90	90	100	150	90	130	110	150	130	170		
Boiling	75	100	125	175	125	175	150	200	175	225		
Hot Water (if vent. desired)												
Not Boiling	50	75	75	125	60	90	75	100	90	110		
Boiling	75	100	125	175	125	175	150	200	175	225		

Note 1—Column *A* refers to tank with hood along one side or two parallel sides when one hood is against a wall or a baffle running length of tank and as high as tank is wide; also to tanks with exhaust manifold along center line with $\frac{W}{2}$ becoming tank width in *W/L* Ratio.

Column *B* refers to free standing tank with hood along one side or two parallel sides.

Note 2—Complete control of the vapors and mist from degreasing operations requires the same ventilation rates as recommended for pickling. However, the solvents employed in degreasing operations are relatively volatile, and the solvent loss caused by evaporation increases rapidly as the exhaust rate at the tank increases. For this reason perfect control is usually sacrificed in favor of lower solvent evaporation rates. Where solvent loss does not present an important cost or operating problem, the exhaust rates given for pickling should be adhered to, but where solvent loss must be kept at a minimum, an exhaust rate of 50 cfm per sq ft of tank area is commonly employed. Where this rate is used, control will be adequate only if the tank is located in an area free of drafts and if the degreasing operations are carried out in accordance with a rigid schedule.[38]

Class I. Includes non-abrasive applications such as paint spray, woodworking, pharmaceutical and food product, and discharge ducts from dust collectors.

Class II. Includes applications for non-abrasive material in high concentration (low-pressure pneumatic conveying); moderately abrasive material; and highly abrasive materials in light concentrations. Typical applications are conveying of chemicals and wood dust; exhaust of foundry shakeouts and sand handling systems, grain dusts; coal crushing and screening; exhaust of grit blast cabinets; and grinding, buffing, and polishing.

Class III. Includes applications for all highly abrasive materials in moderate to heavy concentrations and moderately abrasive materials in heavy concentrations such as low-pressure conveying of tobacco; exhaust systems from sand and grit blasting and abrasive cleaning operations, from rock and ore screening and crushing, from dryers and kilns, and fly ash removal from boiler stacks.

Where aluminum duct is indicated, the following equivalent B & S gage thickness of sheets should be used:

Steel—U. S. Standard Gage	26	24	22	20	18	16	14
Aluminum—B & S Gage	24	22	20	18	16	14	12

3. For exhaust systems on corrosive applications, consideration should be given to non-corrosive materials or coatings.

(Continued on p. 711)

Table 7 .... Exhaust Requirements for Woodworking Operations[41]

(Air Velocity: 4000 fpm in all branches unless noted otherwise)

Self Feed Table Rip Saw
Use 1–5 in. branch at bottom pulley; 1–4 in. branch at saw head.

Large Self Feed Rip Saws Other Than Table Saws
Use 1–6 in. branch at bottom; 1–5 in. branch at top.

Swing Saws
Up to 20 in. use 1–4 in. branch; over 20 in. use 1–4½ in. branch.

Circular Saws

Up to 12 in. dia.	1–4 in. branch
Over 12 in. dia.	1–5 in. branch
Dado head saws	1–5 in. branch

Gang Circular Saws
Ventilation rate depends on effectiveness of hood and maximum number and diameter of saws.

Disc Sanders

Up to 12 in.	1–4 in. branch
12 in. to 18 in. dia.	1–4½ in. branch
18 in. to 26 in. dia.	1–5 in. branch
26 in. to 32 in. dia.	2–4 in. branch
32 in. to 38 in. dia.	1–4 in. branch and 1–5 in. branch
38 in. to 48 in. dia.	1–5 in. branch and 2–4 in. branch

Multiple Drum Sanders (over the table)

Up to 31 in. wide	5 in. branch (see note)
31 in. to 49 in. wide	6 in. branch (see note)
49 in. to 67 in. wide	7 in. branch (see note)
Over 67 in. wide	8 in. branch (see note)

Note: One branch of size indicated required for each drum. Fewer branch pipes may be used as long as total branch pipe cross-sectional area is not reduced. Some manufacturers use 1 extra branch of equal size on feed side. 1–4 in. branch may be required for exhausting bed rubber cushions or mechanically operated brush

Single Drum and Spindle Type Sanders

Up to 50 sq in. sanding surface	1–3 in. branch
50 to 200 sq in. sanding surface	1–4 in. branch
200 to 400 sq in. sanding surface	1–5 in. branch
400 to 700 sq in. sanding surface	1–6 in. branch
700 to 1400 sq in. sanding surface	1–7 in. branch
1400 to 2400 sq in. sanding surface	1–8 in. branch

Horizontal Belt Sander or Edge Sander (see note)

	Drive Pulley	Idler Pulley
Up to 6 in. wide	1–4½ in. branch	1–4 in. branch
6 in. to 9 in. wide	1–5 in. branch	1–4 in. branch
9 in. to 14 in. wide	1–6 in. branch	1–4 in. branch
Over 14 in. wide	1–7 in. branch	1–5 in. branch

Note: Where more than one belt is used, the pipe sizes specified shall be required for each belt. Where a common hood is used for more than one belt, a single pipe of an area not less than the sum of the areas of the pipes specified may be used.

Where belt is reversible, pipe at idler pulley shall be the size specified for the drive pulley.

Vertical Belt Sanders (rear belt and both pulleys enclosed)

Up to 6 in. wide	4½ in. branch on bottom
6 in. to 9 in. wide	5 in. branch on bottom
9 in. to 14 in. wide	6 in. branch on bottom
Over 14 in. wide	7 in. branch on bottom

Band Saws and Resaws

	Bottom	Top
Up to 2 in. wide	1–4 in. branch	1–4 in. branch
2 in. to 3 in. wide	1–5 in. branch	1–4 in. branch
3 in. to 4 in. wide	1–6 in. branch	1–5 in. branch
4 in. to 5 in. wide	1–7 in. branch	1–5 in. branch
Over 5 in. wide	1–8 in. branch	1–5 in. branch

Jointers

Up to 6 in. knives	1–4 in. branch
6 in. to 12 in. knives	1–5 in. branch
12 in. to 20 in. knives	1–6 in. branch
Over 20 in. knives	1–7 in. branch

Single Planners or Surfacers

Up to 20 in. max. working width	1–6 in. branch
20 in. to 26 in. max. working width	1–7 in. branch
26 in. to 32 in. max. working width	1–8 in. branch
32 in. to 38 in. max. working width	1–9 in. branch
Over 38 in. max. working width	1–10 in. branch

Double Planers or Surfacers

	Upper head	Lower
Up to 20 in. max. working width	1–6 in. branch	1–5 in. branch
20 in. to 26 in. max. working width	1–7 in. branch	1–6 in. branch
26 in. to 32 in. max. working width	1–8 in. branch	1–7 in. branch
32 in. to 38 in. max. working width	1–9 in. branch	1–8 in. branch
Over 38 in. max. working width	1–10 in. branch	1–8 in. branch

Molders, Matchers, Sizers (see note)

	Matcher head	Top	Bottom
Up to 4 in.	2–4 in. branch	1–5 in. branch	1–6 in. branch
4 to 6 in.	2–5½ in. branch	1–6 in. branch	1–6 in. branch
6 to 8 in.	2–6 in. branch	1–7 in. branch	1–6 in. branch
8 to 18 in.	2–6 in. branch	1–8 in. branch	1–8 in. branch

Note: Where profiler is used, provide additional pipe not less than that specified for top head.

Wood Shapers
1–4½ to 10 in. branch for each spindle according to size and character of work.

Tenoner
For cut-off saws: Up to 12 in. dia. 1–4 in. branch
Over 12 in. dia. 1–5 in. branch
Use 1–5 in. branch minimum for each top and bottom tenon, cope, and dado head.

Automatic Lathes
Use 1–3 in. to 1–10 in. branch according to work length.

Hogs
Air volume based on 50 cu ft per lb of refuse conveyed. Use 4500 fpm minimum conveying velocity.

Floor Sweeps
Use 6 in. branch for fine dust to 8 in. branch for coarse material.
Mouth at floor, 10 × 4 in. to 12 × 5 in.

Note: Area of floor sweep pipe need not be included in computing main pipe area.

Miscellaneous Equipment
Pulley pockets and chain mortises-3 in. branch; dovetail and lock corner machines, dowel machines, duplex molding sanders, forming lathes, panel raisers (each head), ploughs, rail shears, routers, sash stickers (each head) 4 in. branch; pulley stiles-5 in. branch; glue jointer-6 in. branch.

Table 8 .... Approximate Conveying Velocities

Material Conveyed	Design Velocity Fpm
Vapors, gases, fumes, very fine dust	2,000
Fine dry dusts	3,000
Average industrial dusts	3,500
Coarse particles	3,500–4,500
Large particles, heavy loads, moist materials	4,500 and over

(*Continued from p. 709*)

4. Elbows and angles shall be a minimum of two gages heavier than straight lengths of equal diameter.

5. Hoods shall be a minimum of two gages heavier than straight section of connecting branches.

6. Where flexible piping is necessary, a non-collapsible type of flexible piping shall be used and its length shall be kept at a minimum.

Construction

1. Longitudinal joints of ducts shall be lapped, and riveted or spot-welded on 3-in. centers, maximum. Double-lock seams may be used on Class I application only.

2. Girth joints of ducts shall be made with lap in direction of air flow with 1-in. lap for diameters up to 19 in. and 1¼-in. lap for diameters over 19 in.

3. Elbows and angles shall have an inside or throat radius of two pipe diameters whenever possible. Large radii are recommended for heavy concentrations of highly abrasive dusts. Construct elbows 6 in. or less in diameter of at least five sections, over 6 in. diameter of seven sections. Prefabricated elbows of smooth construction may be used. Angles shall be pieced proportionately.

4. Hoods must be free of sharp edges or burrs and shall be reinforced to provide necessary stiffness.

System Details

1. Connect duct to fan inlet with split sleeve drawband one pipe diameter long but not less than 12 in.

2. Transitions in mains and sub-mains are to be tapered; a taper 5-in. long for each 1-in. change in diameter is recommended when possible.

3. All branches shall enter the main at the large end of the transition at an angle not to exceed 45 deg (30 deg is preferred). Connect branches only to top or sides of main, with no two branches entering diametrically opposite.

4. Provide dead end caps within 6 in. from last branch of all mains and sub-mains.

5. Provide access openings or cleanouts every 10 ft and near each elbow, angle, or duct junction in horizontal sections except for non-corrosive gases and vapors containing no particulate matter.

6. Support ducts sufficiently to prevent placing of any load on connected equipment and to carry weight of system if plugged with material. Use maximum supporting interval 12 ft for 8 in. or smaller ducts and 20 ft for larger ducts.

7. Provide 6 in. minimum clearance between ducts and ceiling, wall, or floor.

8. Where blast gates are used for adjustment of system, place them near connection of branch to main. Provide means of locking after adjustments have been made. Butterfly-type dampers shall not be permitted.

9. Fire dampers, explosion vents, etc., should be installed in accordance with NFPA Codes or local fire ordinances.

10. Rectangular ducts can be used only when clearances prevent the use of round construction. Rectangular ducts must be as nearly square as possible. Weight of metal, lap, and other construction details are to be the equal of round duct construction whose diameter equals the longest side.

11. Where State or local laws conflict with the specifications given, the more stringent regulation shall be followed. Any other deviation must be approved before installation.

AIR FLOW PRODUCING EQUIPMENT

The principal types of air moving equipment are centrifugal fans, axial flow fans, and venturi ejectors. Chapter 22 includes information on fan types, arrangements, and application. Gravity stacks have application for some systems handling higher air temperatures or steam where no air cleaning equipment is installed.

Centrifugal fans are most frequently used for the bulk of exhaust system applications due to the system pressures involved. Where air cleaning equipment is included, fans are usually located on the clean air side. The paddle wheel or modified paddle wheel designs are heavily constructed and have few blades to make them more suitable if wear, corrosion, or accumulations are a factor. The higher efficiency backward-curved blade designs find application where relatively clean, non-corrosive air is handled. Forward-curved blade designs have limited application due to the number, shape, and metal thickness of the short curved blades.

Belt driven fans are recommended by many designers because a change in fan speed is often needed where future changes of the system may be involved, where added pressure losses from air cleaning equipment improvements are anticipated, or where the fan may be used at some later date for another application.

The axial-flow fan is used for systems having low-pressure losses. Propeller or disc designs develop pressures under 1 in., the vane-axial designs develop higher pressures but seldom are used where pressures exceed 3 in.

The venturi ejector[46] is an inefficient method of air movement, but has the advantage of causing air flow without having the exhaust air pass through the air flow producing equipment. It minimizes the explosion or corrosion potentialities in certain types of system.

Fans, motors, and drives should be located so that safe and easy access for periodic inspection, servicing, and maintenance is possible.

AIR CLEANING EQUIPMENT

As discussed in detail in Chapter 24, air cleaning equipment should be considered in all systems where it can prevent property damage, neighborhood pollution, or reentry of polluted air to the working space, or where it can salvage usable material, reduce fire and explosion hazards, or make possible some recirculation of air to the working spaces. The present growing emphasis on air pollution control makes it desirable to remove all contaminants to the greatest practical degree based on reasonable cost and maintenance.

MAKE-UP AIR REQUIREMENTS

A correctly designed exhaust system may be ineffective during periods when windows and doors are closed. This condition should be anticipated and prevented by making provision for an adequate air supply. Poor performance of exhaust hoods is caused by: absence of air supply systems; cold drafts due to high indraft velocities through cracks and openings in building construction especially at windows and doors; reverse air flow through low-pressure systems of roof ventilators or general ventilation systems; downflow through heater vent stacks preventing exhaust of flue gases to the outdoors. Systems for conditioning make-up air do not necessarily increase heating requirements for the space, as cold air due to infiltration

Table 9 .... Resistance to Corrosion of Materials Used for Hoods and Ducts[46]

Process	See Footnote 1	Galvanized Iron	Black Iron	Stainless Steel	Monel	Lead Lining	Aluminum	Hard Rubber	Natural Rubber	Pressed Wood	Neoprene	Terne Plate	Buna S	Koroseal	Paint on Iron Galv or Black (Asphalt)	Hot, 180–200 F on Galv Iron (Asphalt)	3-Course Felted on Metal or Wood (Asphalt)	Painted Pressed Wood (Asphalt)	Paint (Phenolic Resin)	Bonded Plywood or Redwood Baked, or Solid Cast Resin (Phenolic Resin)		Asbestos Cement Board or Tube	Vinylite Resin Type Paint	Vitrified Clay Pipe with Mastic Joints
Acid, Chromic (Plating and Anodising)		EP	EP	S		ES				SP	S	S			P				SF	P	EF			E
Acid, Hydrochloric		EP	P	E	S	F		E	E4		E	F	E	E	EF	S		S		F	EP	F	E4	E
Acid, Hydrofluoric		P			S				S						S						E			F
Acid, Nitric		P		E		E					E				FP	F					F		S	E
Acid, Oxalic (Anodising)					E	S	E								S									E
Acid, Sulfuric		P	P	S5		E6		E	E7	P	E	S			EP	S	F		F	ES	E	E5	EP	E
Acid Dipping and Pickling (Horiz Ducts)															P	F	E							E
Alkaline Cleaner (Caustic 140–180 F)		EP	EP	E						S	E				ES			S			S	E		E
Alkaline Cleaning Solutions	2	E	E	E							E				E									E
Alkaline Plating (Cu, Cd, Ag)		EP	EP							E	S				ES			S			S	S	S	E
Plating and Anodising (Horizontal Ducts)															P	S	E							E
Plating, Bright Nickel		SP	SP								S				E			S					S	E
Plating and Dipping (Vertical Ducts)		P													F					E				E
Tanning, Dressing, Dyeing, Bleaching Sol		S8							S9															E
Water, Boiling and Steam		EP	SP	E	E						E				ES	E						E	S	E
Decarbonising Agents	3	E																						E
Degreasing Compounds: Organic & Mineral Sol		E	E																					E
Enamels and Lacquers (Coating & Dipping)		E	S																					
Fabric Impregnation		S																						
Salt, Supersaturated Solution 300–400 F		S	S																					E
Wire Insulation		S																						

Classification	*Approximate Service Life*
E—Excellent	Over 5 years
S—Satisfactory	2 to 5 years
F—Fair	½ to 2 years
P—Poor	6 months
Combinations (EP)	Indefinite rating

Notes: 1. Ratings are based upon collated field service reports. Indefinite ratings are given where reports are inconsistent due to varying operating conditions (such as dilution, exposure to weather, etc.). Blank spaces denote insufficient information, not poor service. 2. Mild soaps, detergents, water soluble oils, 140–180 F. 3. Creosote oil, kerosine, ethylene dichloride. 4. To 20 percent: E; 50-70 percent, 150 F: S. 5. To 50 percent, plating: E; 20 percent anodising, agitated 80 F: S. 6. To 50 percent plating: S; over 50 percent pickling, 20 percent anodizing: E. 7.5–22 percent, 135–215 F pickling: S; plating, pickling, anodising: E. 8. Except acid. 9. Acid.

from the outdoors must be heated somehow to maintain comfortable temperatures in the area. The cost will depend on the ratio of exhaust air to that required for ventilation of the space occupied.

MAINTENANCE OF PERFORMANCE

Periodic inspection and checking of exhaust systems are necessary if control is to be maintained at the effective level of the original installation. Continued effectiveness depends on maintained design air volume flowing through the exhaust hoods. A checking procedure, therefore, must include some data to indicate at least relative air flow through the hoods. The static pressure or hood suction measurement will prove useful for such checking if data are available on air volumes and pressures at the time the system was installed. Testing and recording of such data for each new installation are of the utmost importance.

While hood suction readings have rightfully been discarded as a means of measuring air flow, they do offer a quick and accurate method of measuring relative air flow. If the hood suction is known while an exhaust system is functioning properly, its continued effectiveness can be assured so long as the hood suction is not reduced from its original value.

Difficulty from plugging may be encountered where heavy dust loads or moist air are encountered. Unless the hood design is altered or there are accumulations in the hood or branch pipe between hood and point of hood suction reading, the air volume exhausted from a hood cannot change without a change in hood suction reading.

The Pitot tube can be used for routine check purposes instead of the static pressure method but requires care in reading velocity pressures in an exact position with the tube paralleling the flow of air. The hood suction method of checking can, however, be more readily delegated to an assistant having no technical training. U-gages have been standard plant equipment long enough to eliminate any feeling of uncertainty in their use.

Since pressure readings vary as the square of the velocity or volume of flow, a slight change in flow is magnified by a comparison of gage readings. Normally, a reduction of volume or velocity of 10 to 15 percent will not be sufficient to reduce the effectiveness of the exhaust system. This range is equivalent to a reduction in static pressure readings of 19 to 30 percent.

A marked reduction in hood suction can often be traced to one or more of the following items:

1. Reduced performance by the exhaust fan caused by reduced speed due to belt slippage, wear on rotor or casing, or an accumulation of material in the rotor or casing obstructing the air flow.

2. Incorrect direction of exhauster rotation.

3. Reduced performance caused by defects in the exhaust piping, such as accumulations of material in branch or main ducts due to insufficient conveying velocities, condensation of oil or water vapors on duct walls, adhesive characteristics of material exhausted.

4. Leakage losses caused by loose cleanout doors, broken joints, holes worn in duct (most frequently in elbows), or poor connection to the exhauster inlet.

5. Losses in suction due to exhaust openings added to the system or due to a change of setting of blast gates in branch lines.

6. Increased pressure loss through the dust collector due to lack of maintenance, improper operation, wear, etc.

Table 10 .... Resistance to Corrosion of Materials Used for Exhaust Fans[46]

Process	Steel	Aluminum	Galvanized Steel	Cast Iron	Brass	Lead	Asphalt Paint	Vinyl Resin Paint	Red Lead Paint	Hot Asphalt	Phenolic Resin Baked Coating	Hard Rubber Lining	Soft Rubber	Neoprene	Everdur	Koroloc	Pressed Wood	AISI 316	Acid Bronze
Acid Dipping, Pickling, and Stripping	EP	S		E		E	SP		E		S	E	ES	E	E	F			P
Alkaline Cleaning	EF	S	E	E			EP				S	ES		E	E	F		E	
Anodizing	SF	S					E	E	E		E	E		E					
Coating—Enamel and Lacquer	E	E			E						S			E					
Decarbonizing	ES				E		P					E			E	F			
Degreasing and Metal Cleaning	ES		ES		E														
Hardening and Quenching	ES		ES																
Plating	EP	S		E			ES	S			ES	E	E	E					
Washing and Rinsing	EP	E		E			SP			S		E		E			F		

Classification	*Approximate Service Life*
E—Excellent	Over 5 years
S—Satisfactory	2 to 5 years
F—Fair	6 months to 2 years
P—Poor	6 months
Combinations	Variable performance

Note: Ratings are based upon collated field service reports. Forward curved fans are *not* recommended for open-tank ventilation service. Cleanout doors and bottom drains are recommended on all fans on open tank operations. The deposits which build up on fan wheels and casings should be removed by regular cleaning. Repaint where any coating has been broken and removed.

MATERIALS REQUIRED FOR CORROSION RESISTANCE

In many cases, exhaust systems including hoods, ducts, air flow producing equipment, and air cleaning equipment will require protective coatings if carbon steel is used, or other materials of construction will be required. Need for such materials will be determined by (*a*) the corrosion rates on interior or exterior duct surfaces, (*b*) protection against product contamination, and (*c*) explosion hazards.

Construction for *corrosion protection* is the most difficult due to the complex factors that influence rate of corrosion. It is seldom possible to predict the concentration, composition, and dry-bulb and dew-point temperatures that will exist in actual operations. A guide for selecting corrosive resistant materials based on a survey of actual experiences has been reproduced in Tables 9 and 10.[46]

REFERENCES

The material in this chapter is based largely on the recommendations of the American Conference of Governmental Industrial Hygienists as published in the *Industrial Ventilation Manual*, 1958 Edition.

[1] J. J. Bloomfield and J. M. DallaValle: *The Determination and Control of Industrial Dust* (*U. S. Public Health Service Bulletin* 217, 1935).

[2] J. M. DallaValle: *Exhaust Hoods* (Industrial Press, New York, 1952).

[3] T. Hatch: Design of exhaust hoods for dust control systems (*Journal of Industrial Hygiene and Toxicology*, Vol. 18, 1936, p. 595).

[4] W. C. L. Hemeon: Air dilution in industrial ventilation (*Heating and Ventilating*, February 1941).

[5] B. F. Postman: Practical application of industrial exhaust ventilation for the control of occupation exposures (*American Journal of Public Health*, Vol. 30, 1940, p. 149).

[6] W. N. Witheridge: *Principles of Industrial Process Ventilation* (University of Michigan, Inservice Training Course, October 1945).

[1] P. Drinker and T. Hatch: *Industrial Dusts* (McGraw-Hill Book Co., New York, 1936).

[2] J. L. Alden: *Design of Industrial Exhaust Systems* (Industrial Press, New York, 1948).

[3] W. N. Witheridge: Ventilation (*Industrial Hygiene and Toxicology*, Vol. I, Chapter 10, ed. F. A. Patty, Interscience Publishers, New York, 1948).

[4] A. C. Stern et al: Transport velocities for industrial dusts (*American Industrial Hygiene Association Quarterly*, December 1948).

[5] A. D. Brandt: *Industrial Health Engineering* (John Wiley and Sons, New York, 1947).

[6] L. Silverman: Velocity characteristics of narrow exhaust slots (*Journal of Industrial Hygiene and Toxicology*, November 1942, p. 267).

[7] W. C. L. Hemeon: *Plant and Process Ventilation* (The Industrial Press, New York, 1955).

[8] *Code of Recommended Good Practice for Metal Cleaning Sanitation* (American Foundrymen's Association).

[9] *Industrial Ventilation Manual* (American Conference of Governmental Industrial Hygienists, 1958, 5th ed.).

[10] W. C. Dressen et al: *A Study of Asbestosis in the Asbestos Textile Industry* (*U. S. Public Health Service Bulletin* No. 241, 1938).

[11] R. T. Page and J. J. Bloomfield: *A Study of Dust Control Methods in an Asbestos Fabricating Plant* (*U. S. Public Health Reports*, Reprint No. 1883, November 1937).

[12] *Rules Relating to the Control of Silica Dust in Stone Crushing Operations* (New York State Department of Labor, *Industrial Code Rule* No. 34, July 1942).

[13] J. M. Kane: Design of exhaust systems (*Heating and Ventilating*, November 1945, p. 68).

[14] J. M. Kane: Foundry ventilation (*The Foundry*, February, March 1946).

[15] W. M. Oddie: Pottery dusts: their collection and removal (*Pottery Gazette*, Vol. 53, 1928, p. 1280).

[16] J. M. Kane: Scrapbook of exhaust hood design (*Heating and Ventilating*, July, August, November 1950 and February, March, April, June 1951).

[17] R. T. Pring et al: Design of exhaust ventilation for solid material handling (*Industrial and Engineering Chemistry*, November 1949).

[18] *Hartzell Blowers: Engineering Data & Installation* (*Hartsell Propeller Fan Company Bulletin* 1001).

[19] J. M. Kane: The application of local exhaust ventilation to electric melting furnaces (*American Foundrymen's Association Transactions*, Vol. 52, 1945, p. 1351).

[20] T. Hatch et al: Control of silicosis hazard in the hard rock industries. II. An investigation of the Kelley dust trap for use with pneumatic rock drills of the jackhammer type (*Journal of Industrial Hygiene*, February 1932, p. 69).

[21] P. S. Hay: Modified design of Hay dust trap (*Journal of Industrial Hygiene*, January 1930, p. 28).

[22] *American Standard for Grinding, Polishing and Buffing Equipment Sanitation* (American Standards Association Z 43-1941).

[23] J. M. Kane: Swing frame grinder dust control (*The Foundry*, August 1944).

[24] J. M. DallaValle: Design of kitchen range hoods (*Heating and Ventilating*, August 1953, p. 95).

[25] *What We Make* (*B. F. Sturtevant Co. Catalog* No. 500).

[26] P. Drinker and J. R. Snell: Ventilation of motion picture booths (*Journal of Industrial Hygiene and Toxicology*, April 1938, p. 321).

[27] P. J. Marschall: How to justify an industrial air conditioning investment (*Heating, Piping and Air Conditioning*, February 1952, p. 71).

[28] E. C. Riley et al: How to design exhaust hoods for quartz-fusing operations (*Heating and Ventilating*, April 1940, p. 23).

[29] E. C. Riley and J. M. DallaValle: A study of quartz-fusing operations with reference to measurement and control of silica fumes (*U. S. Public Health Reports*, 1939, Vol. 54, p. 532).

[30] A. C. Stern: Ventilation of open tanks (*Industrial Hygiene Quarterly*, September 1950).

[31] I. Kingsley: Tabulation of atmospheric contaminants released by open surface tanks (New York State Department of Labor, Division of Industrial Hygiene, *Engineering Unit Plates* 161, 162, 163).

[32] K. M. Morse and L. Goldberg: Chlorinated solvent exposures at degreasing operations (*Industrial Medicine*, October 1943, p. 706).

[33] W. N. Witheridge and H. T. Walworth: Ventilation of a trichlorethylene degreaser (*Journal of Industrial Hygiene and Toxicology*, May 1940, p. 175).

[34] *American Standard for Safety in Electric and Gas Welding and Cutting Operations* (American Standards Association, A 49.1).

[35] *Rules Relating to the Removal of Dust, Gases, and Fumes* (New York State Department of Labor, *Industrial Code Rule* No. 12, January 1931).

[36] T. F. Hatch and D. Barron-Gorozco: Air flow in free convection over heated bodies (ASHAE TRANSACTIONS, Vol. 63, 1957, p. 275).

[37] A. D. Brandt and R. J. Steffy: Energy losses at suction hoods (ASHVE TRANSACTIONS, Vol. 52, 1946, p. 205).

[38] A. Nutting: Resistance test on pipe (*Mechanical Engineering*, May 1938).

[39] G. E. McElroy: *Design of Injector for Low Pressure Air Flow* (*U. S. Bureau of Mines Technical Paper* 678).

[40] *American Standard Safety Code for Ventilation and Operation of Open Surface Tanks* (American Standards Association, Z 9.1–1951).

CHAPTER 53

INDUSTRIAL DRYING SYSTEMS

Drying Terminology, Mechanism of Drying, Internal and External Conditions, Periods of Drying, Equilibrium Moisture Content, Application of Hygrometry to Drying, Determination of Drying Time, Commercial Drying Time, Dryer Calculations, Drying Methods and Equipment; Radiant, Conduction, and Convection Drying; Agricultural Drying, Solution of Drying Problem

THE term *drying*, in a broad sense, encompasses the removal of water, and occasionally other liquids, from gases, liquids, or solids. However, the common usage of the word confines the meaning principally to the removal of water or solvent from solids by thermal means. *Dehumidification* is the term that is commonly assigned to the drying of gases. This is usually accomplished by condensation or adsorption by various drying agents, and is treated in Chapter 42. *Distillation*, and more particularly *fractional distillation*, is associated with the drying of liquids.

It is usually more economical to employ, whenever possible, mechanical means of separating as much water as is practicable from the solid materials before undertaking drying or dehydration steps. These mechanical methods such as filtration, screening, pressing, centrifuging, or settling usually require much less power, and frequently less capital outlay, thereby making the operation cheaper in terms of cost per pound of water removed.

DRYING TERMINOLOGY[1]

The generally accepted definitions of terms used in drying technology follow:

Adiabatic drying occurs in a dryer when all of the sensible heat given up by the air in cooling is used to evaporate water from the wet stock.

Batch drying is that type of drying operation in which the material is fed to and discharged from the drying chamber in batches at definite intervals of time.

Bone-dry basis indicates the moisture content of a wet solid as pounds of water per pound of bone-dry solid. The advantage of using this basis is that the absolute amount of moisture loss is obtained simply by subtracting the moisture contents before and after drying. (See definition of *Wet Basis*.)

Bound moisture refers to liquid (held by a solid) which exerts a vapor pressure less than that of the pure liquid at the same temperature. Liquid may become *bound* by retention in small capillaries, by solution in cell or fiber walls, by homogeneous solution throughout the solid, and by chemical or physical adsorption on solid surfaces. Bound moisture can be removed from a solid only under specific conditions of humidity in the external surroundings.

Capillary flow refers to the flow of liquid through the interstices and over the surface of a solid. It is caused by liquid-solid molecular attraction.

Commercial dry basis expresses the moisture content of a product as pounds of water per pound of solid as it leaves the dryer, i.e., per pound of commercially dry solid.

Commercial drying time is the residence time of the material being dried in a commercial dryer.

Constant-rate period is that drying period during which the rate of water removal per unit of drying surface is constant.

Continuous drying is that type of drying operation in which the material to be dried is fed and discharged continuously to and from the dryer. The operating conditions are independent of time.

Critical moisture content is that obtaining when the constant-rate period ends and the falling-rate period begins.

Equilibrium moisture content is that to which a given material can be dried under specific conditions of air temperature and humidity.

Evaporative load is the total amount of water evaporated per hour in the dryer.

Falling-rate period is that drying period during which the instantaneous drying rate continually decreases.

Fiber-saturation point is the moisture content of cellular materials (wood, etc.) at which the cell walls are completely saturated while the cavities are liquid-free. It may be defined as the equilibrium moisture content as the humidity of the surrounding atmosphere approaches saturation.

Final moisture content is the percentage by weight of moisture remaining in the solid at the end of the drying operation.

Free moisture content is that liquid content which is removable at a given temperature and humidity. Free moisture may include both bound and unbound moisture.

Fuel economy is the number of units of fuel (cubic feet of gas, pounds of steam, gallons of oil, etc.) required to remove 1 lb of water from the solid being dried.

Humidity denotes the amount of water vapor actually present in a gas, and is generally expressed as weight of vapor per unit weight of any gas.

Hygroscopic material is material that may contain bound moisture.

Initial moisture content is the percentage by weight of moisture in the solid at the start of the drying operation.

Initial moisture distribution refers to the moisture distribution throughout a solid when drying begins.

Moisture gradient refers to the internal distribution of water in a solid at a given moment in the drying process, the nature of which depends on the characteristics of the solid involved.

Non-hygroscopic material is material that can contain no bound moisture.

Test drying time is the drying time of a relatively small and representative sample of the material which is dried in a laboratory dryer under conditions simulating commercial operation.

Through-circulation drying is that method of drying in which the air flow is directed through a permeable bed of solids.

Unbound moisture in a hygroscopic material is that moisture in excess of the equilibrium moisture content corresponding to saturation humidity. All water in a non-hygroscopic material is unbound water.

Wet basis expresses the moisture in a material as a percentage of the weight of the *wet* solid. This basis is less satisfactory than the dry-weight basis on which the percentage change of moisture is constant for all moisture contents.

MECHANISM OF DRYING[1]

When a solid dries, two fundamental processes are involved: (1) the transfer of heat to evaporate the liquid, and (2) the

transfer of mass as vapor and internal liquid. These two processes occur simultaneously, and the factors governing the rate of each process determine the rate of drying.

In any commercial drying problem, a principal objective is to supply the required heat in the most efficient manner. Consequently, heat transfer may occur by convection, conduction, or radiation, or by any combination of these mechanisms. The various types of industrial dryers may be shown to differ fundamentally with respect to the method used for transferring heat to the solid. In general, heat must flow first to the outer surface of the solid and then into the interior. An important exception is drying with high frequency electrical currents where heat is *generated within the solid*, producing a higher temperature at the interior than at the surface, and consequently, causing heat to flow from inside the solid to the outer surfaces.

Mass transfer in drying occurs as liquid or vapor flow, or both, within the solid, and as vapor flow from the external wet surfaces. The nature of liquid-concentration gradients in solids during drying depends on the mechanism of internal liquid flow, and this mechanism, in turn, depends to a large extent upon the physical and chemical characteristics of the solid being dried.

Internal and External Conditions

A study of how a solid dries may be based on the *internal* mechanism of liquid flow, or on the effect of the *external conditions* of temperature, humidity, air flow, state of subdivision, etc., on the drying rate of the solid. The former procedure involves a fundamental study of the liquid flow conditions within a solid during drying. The latter procedure, although less fundamental, is more generally used because the effects are easier to establish and the results have greater immediate application in dryer design and operation.

Internal Mechanism of Liquid Flow. Internal liquid flow may occur by several mechanisms, depending on the structure of the solid. Several mechanisms of flow are as follows:

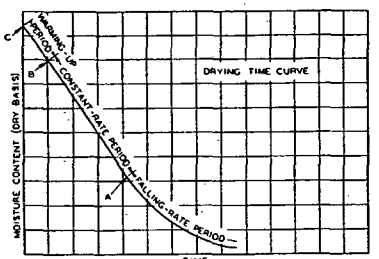
1. *Diffusion* in continuous, homogeneous solids.
2. *Capillary flow* in granular and porous solids.
3. Flow caused by *shrinkage* and *pressure* gradients.
4. Flow caused by a *vaporization-condensation* sequence.
5. Flow caused by *gravity*.
6. Flow caused by an electrical *potential*, *electro-osmosis*.
7. Flow caused by temperature gradients, *thermal diffusion*.

Although more than one of these mechanisms of flow may be effective at one time, only one predominates as a rule at a given time in a solid during drying. However, a different mechanism may predominate at a different time in the cycle. The mechanism of moisture flow is usually established experimentally from a study of moisture gradients.

External Variables. The principal external variables involved in any drying problem are: *temperature, humidity, air flow, state of subdivision* of the solid, *agitation* of the solid, *method of supporting the solid*, and the *contact* between hot surfaces and wet solid. All these variables do not necessarily occur simultaneously in one problem.

Periods of Drying[1]

A typical drying time curve for a wet solid is shown in Fig. 1. This curve is a plot of the moisture content at any time in a solid undergoing drying. It is the usual method of presenting experimental drying data. Although Fig. 1 shows that the moisture content is subject to a continuous variation with time, a more precise illustration of the nature of this variation can be obtained by differentiating the curve and plotting the drying rate in pounds of water per (hour) (pound of dry material) against the moisture content in pounds of water per pound of dry material as shown in Fig. 2. The *rate curve* shows that the drying process is not a smooth, continuous one in which a single mechanism controls throughout.



From Reference 2

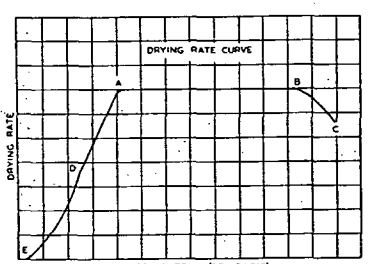
Fig. 1 . . . . Moisture Content w vs. Drying Time θ

Section BA on each curve represents a *constant-rate period*. In Fig. 1, it is shown by a straight line of constant slope $dw/d\theta$, which becomes a horizontal line on the rate curve in Fig. 2.

The curved portion of Fig. 1 is termed the falling-rate period, and, as shown in Fig. 2, it is typified by a continuously changing rate. Point A, where the constant rate ends and the drying rate begins to decrease, is termed the *critical moisture content*.

The portion of the curve designated by CB in Figs. 1 and 2 represents a warming-up period, and it may, or may not, be a significant item depending on the total time involved.

Constant-Rate Period. Drying during the constant-rate period is equivalent to evaporation from a free-water surface on the surface of the solid. The rate of drying in this period is determined by the rate of diffusion of water vapor through an air film at the wet surface of the solid. A constant rate of evaporation on the surface of the solid maintains the surface at a



From Reference 2

Fig. 2 . . . . Rate of Drying $\frac{dw}{d\theta}$ vs. Moisture Content w

constant temperature, which, in the absence of other heat effects, is very nearly the wet-bulb temperature of the air. If heat flows to the surface of evaporation by radiation and conduction, or both, in addition to convection, the surface temperature will be constant at some value between the air temperature and the wet-bulb temperature. This higher temperature in turn produces a higher constant rate of evaporation.

In those dryers in which heat is transferred to a wet solid by conduction through hot surfaces, and heat transfer by convection is not a factor, the wet surfaces approach the boiling point temperature rather than a wet-bulb temperature.

When all the heat for evaporation in the constant-rate period is supplied by a hot gas, a dynamic equilibrium is established between the rate of heat transfer to the material and the rate of vapor removal from the surface. This equilibrium between heat and mass transfer rates can be expressed as follows:

$$\frac{dw}{d\theta} = \frac{h_t A \Delta t}{H} = k_g A \Delta p \qquad (1)$$

where

$\frac{dw}{d\theta}$ = drying rate, pounds of water per (hour) (pound of bone-dry material).
h_t = total heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
A = area of heat transfer and evaporation, square feet per pound of bone-dry material.
H = enthalpy of evaporation at t_s, Btu per pound.
k_g = mass transfer coefficient, pounds per (hour) (square foot) (atmosphere).
$\Delta t = (t_a - t_s)$ = temperature difference between air and surface of evaporation, Fahrenheit degrees.
t_a = air temperature, Fahrenheit.
t_s = temperature of surface of evaporation, Fahrenheit.
$\Delta p = (p_s - p_a)$ = vapor-pressure difference, atmospheres.
p_s = vapor pressure of water at t_s, atmospheres.
p_a = partial pressure of water vapor in air, atmospheres.

When $h_t = h_c$, the coefficient of heat transfer by convection only, then t_s under equilibrium conditions becomes t_w, the wet-bulb temperature of the air, and p_s is the vapor pressure at this temperature. If heat is also supplied by radiation, then h_t is the sum $(h_c + h_r)$ where h_r is the radiation coefficient and h_c is the convection coefficient, and t_s becomes higher than the wet-bulb temperature. A similar result occurs when heat reaches the surface of evaporation by convection and conduction.

Effect of Air Velocity. The principal effect of air velocity is on h_c and k_g, since the rate of transfer of heat and mass in the constant-rate period depends mainly on the rate of diffusion of heat and vapor through the air film at the surface of the solid, and air velocity is the chief factor affecting the thickness of this film. The influence of direction of air flow on the heat transfer coefficient h_c and on the corresponding drying rate $\frac{dw}{d\theta}$ *is shown in Table 1.*

Effect of Temperature or Humidity. Temperature or humidity enters the drying rate equation as a driving force across the air film. The wet-bulb depression, which is the difference between the dry-bulb temperature and the wet-bulb temperature, is directly proportional to the drying rate in this period.

Table 1 . . . . Convection Heat Transfer (h_c) and Rates of Drying $\left(\frac{dw}{d\theta}\right)$ Coefficients for Constant-Rate Period

Direction of Air Flow	h_c	$\frac{dw}{d\theta}$
Parallel to plane surfaces[3]	$0.0128G^{0.8}$	$\frac{0.0128G^{0.8}A}{H}(t_a - t_w)$
Perpendicular to plane surfaces[3]	$0.37G^{0.37}$	$\frac{0.37G^{0.37}A}{H}(t_a - t_w)$
Through circulation[4] (for Reynolds number > 300)	$\frac{0.37G^{0.33}c_s}{D_p^{0.41}}$	$\frac{0.37c_sG^{0.33}\Delta t_m}{p_sHD_p^{0.41}}$

Letter Symbols for Table 1
(not previously defined)

h_c = convection heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
G = mass velocity of dry air, pounds per (hour) (square foot).
t_w = wet-bulb temperature of drying air, Fahrenheit.
a = drying area, square feet per cubic foot of bed volume.
ρ_s = bulk density of dry granular bed, pounds per cubic foot.
D_p = average diameter of particle, feet.
Δt_m = logarithmic mean difference between air temperature entering and leaving the bed and the wet-bulb temperature, Fahrenheit.
c_s = humid heat, Btu per (pound of dry air) (Fahrenheit degree).

When dealing with heat transfer coefficients the wet-bulb depression is the driving force, whereas with mass transfer coefficients the driving force is expressed in terms of humidity or vapor-pressure differential.

Fig. 3 permits a ready estimate of the constant drying rate for various air temperatures and humidities. The chart is based on the difference between the dry-bulb and wet-bulb temperatures of the entering stream of air, and on an air velocity of 300 fpm. It may be assumed satisfactory for tray drying of any material in the constant-rate drying period. It does not apply to rotary or through-circulation drying. A curve for correcting the air velocity is incorporated in Fig. 3. This curve is based on the variation of drying rate with the 0.8 power of the velocity.

Evaporation from Liquid Drops. For the important problem of spray drying, evaporation rates of liquid drops must be estimated. Below a value of Reynolds number (D_pG/μ) of 10 for spherical particles, the heat transfer coefficient across the gas film surrounding the drop is given by

$$h = \frac{2k_f}{D_p} \qquad (2)$$

where

h = film heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
k_f = thermal conductivity of gas film, Btu per (hour) (square foot) (Fahrenheit degree per foot).

Equation 2 is applicable when the Reynolds number for liquid drops is less than 10. Drop diameters are almost always less than 500 microns, and are usually in the range of 20 to 150 microns.

The rate of evaporation of drops may be expressed in terms of heat transfer or mass transfer. In terms of heat transfer, the evaporation rate is given by the equation:

$$\frac{dw}{d\theta} = \frac{2\pi k_f D_p}{H}(t_a - t_s) \qquad (3)$$

where

$\frac{dw}{d\theta}$ = evaporation rate, pounds per hour.

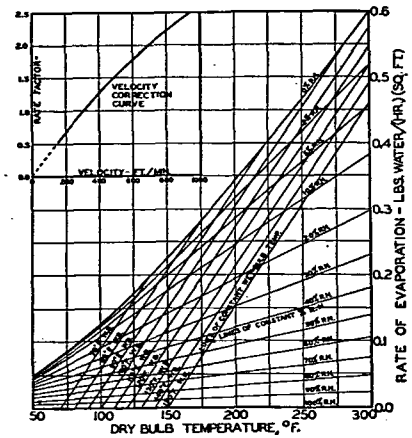


Fig. 3 .... Rate of Evaporation Chart[2]

Drying at Air Temperatures above the Boiling Point of the Liquid. When the temperature of the drying air is maintained above the boiling point of the liquid being evaporated, or when superheated vapors are used for drying, the usual equations for mass transfer, expressing rate of evaporation as a function of the vapor-pressure difference, lose significance, since large errors are introduced in the expression for vapor-pressure driving force due to its apparently small value. Such cases can be treated conveniently on a basis of heat transfer, since a temperature difference must always exist in order that drying may proceed.

Constant-Rate Period When Heat Transfer Depends on Conduction and Radiation. In indirect drying, where heat transfer and drying do not depend on the flow of heated gases, the drying rate depends either on heat conduction through retaining walls to wet material in contact with such surfaces, or on radiation, or both. This applies to drum dryers, agitated pan dryers, indirect continuous sheet dryers, steam tube rotary dryers, vacuum rotary and vacuum tray dryers, and infra-red dryers.

A principal difference between indirect drying and direct drying is that, with the former, the material is usually at a higher temperature than the surrounding air, so that heat is actually transferred to the air instead of from the air.

The Falling-Rate Period. In the discussion of the periods of drying, it was shown that the drying process is discontinuous, consisting of a period of a constant rate of evaporation and a period in which the rate continuously decreases. (See Fig. 2.) This latter period is usually designated as the falling-rate period. It begins when the constant-rate period ends at the critical moisture content. If the critical moisture content is less than the required final moisture content, the constant-rate period will constitute the whole of the drying process. On the other hand, if the initial moisture content is less than the critical moisture content, as in the case of some slow-drying materials, such as soap and wood, then no constant rate will appear, and the whole of the drying process will be in the falling-rate period. This period, in the most general case, can be divided into two zones which may be termed (1) the zone of unsaturated surface drying, and (2) the zone where internal liquid flow controls.

The zone of unsaturated surface drying follows immediately after the critical point and results from a progressively decreasing wetted surface. With the surface no longer completely wetted, dry portions of the solid protrude into the air film, so that the rate of evaporation per unit of total surface is reduced. The effective wetted surface in this zone is frequently a linear function of the water content, so that the curve representing rate of drying vs. water content of the solid is straight in this region, as shown by line AD in Fig. 2. The mechanism of drying is essentially the same as during the constant-rate period.

The zone where internal liquid flow is in control is usually the second zone of the falling-rate period. In this phase the rate of internal liquid movement by one or more of the controlling mechanisms considered previously, such as diffusion, capillarity, etc., will determine the drying rate.

When diffusion does control in the falling-rate period, it obeys the same fundamental laws of diffusion as those applying to the diffusion of heat. Thus for the case where the surface is dry or at the equilibrium moisture content, and the solid has a uniform initial moisture distribution, the following rate equation holds for relatively large values of time θ, and when $(w - w_e)/(w_o - w_e) < 0.6$

$$\frac{dw}{d\theta} = -\frac{\pi^2 d}{4L^2}(w - w_e) \qquad (4)$$

where

- w = moisture content on dry basis, at any time θ, pounds of water per pound.
- w_e = moisture content at equilibrium with external conditions, pounds of water per pound of dry material.
- w_o = initial moisture content at start of diffusional period, pounds of water per pound dry material.
- d = the liquid diffusivity, square feet per hour.
- L = one-half material thickness, feet.

Equation 4 is restricted to a slab-shaped solid, the length of which is large compared with the thickness.

For some materials the drying time in the falling-rate period varies directly with the thickness. When this occurs the falling

Table 2 .... Approximate Classification of Materials Most Likely to Obey Equations 4 and 5

Materials Obeying Equation 4
1. Single-phase solid systems such as soap, gelatin, glue.
2. Wood and similar solids below the fiber saturation point.
3. Last stages of drying starches, textiles, paper, clay, hydrophilic solids, and other materials when bound water is being removed.
Materials Obeying Equation 5
1. Coarse granular solids, such as sand, paint pigments, minerals, etc.
2. Materials in which moisture flow occurs at concentrations above the equilibrium moisture content at atmospheric saturation, or above the fiber saturation point.

rate can be expressed with *fair* accuracy by the following equation:

$$\left(\frac{dw}{d\theta}\right)_f = -\frac{(dw/d\theta)_c}{(w_c - w_e)}(w - w_e) \qquad (5)$$

where

- $(dw/d\theta)_c$ = the constant drying rate, pounds per (hour) (pound dry material).
- $\left(\frac{dw}{d\theta}\right)_f$ = falling rate, pounds of water per (hour) (pound of dry material).
- w_c = the critical moisture content, pounds per pound dry material.

The appropriate expression for $\left(\frac{dw}{d\theta}\right)_c$ obtained from Table 1, may then be substituted in Equation 5.

Table 2 gives an approximate classification of materials that are most likely to obey Equations 4 and 5.

Equilibrium Moisture Content

In the drying of solids it is important to distinguish between hygroscopic and non-hygroscopic materials. A hygroscopic material is one which retains a definite percentage of moisture under definite conditions of air humidity. This bound moisture is in a state of equilibrium with the water vapor in the surrounding air. A decrease in the water vapor content will decrease the amount of equilibrium bound water. Water so retained by a solid in equilibrium with the humidity of the surrounding air, is designated as the *equilibrium moisture content.* Such moisture may be held as adsorbed surface films or condensed in fine capillary structures at reduced vapor pressure.

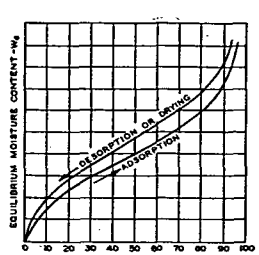


Fig. 4 .... Typical Equilibrium Moisture Content Curves[1]

The equilibrium moisture content varies with the temperature and humidity of the surrounding air. Consequently, any correlation of equilibrium moisture content should take these two factors into account. However, at low temperatures, e.g., 60 to 120 F, a plot of equilibrium moisture content *vs.* percent relative humidity, expressed as 100 (p/p_s), is essentially independent of temperature. Such a plot usually results in a curve of double curvature with a point of inflection (see Fig. 4).

APPLICATION OF HYGROMETRY TO DRYING

In analysing any particular drying problem it is sometimes advisable to consider the thermodynamic changes involving the drying medium, since in most cases the drying medium is an air-water-vapor mixture. The thermodynamic process can be traced on a psychrometric chart.

The psychrometric chart, Fig. 5, suggested by Grosvenor,[5] is suitable for most drying calculations. The adiabatic cooling lines on the chart indicate the paths along which the changes in the thermodynamic properties of the air occur in an adiabatic dryer.

The theoretical ability of air to pick up moisture corresponds to the difference between the final saturation content at the wet-bulb temperature (approximately the adiabatic cooling temperature or thermodynamic wet-bulb temperature) and the initial moisture content at the supply air dew point. The maximum possible pickup is never achieved in practical dryers because on the basis of good design, this would be undesirable. Experience has shown that there is an optimum pickup which is a function of the independent drying variables and is less than the maximum theoretical value as obtained from the psychrometric chart.

It should be noted that in practice the process will not necessarily follow exactly the adiabatic cooling line since there may be a transfer of heat to the tray or conveyor itself.

Use of the psychrometric chart for analyzing drying problems is illustrated by *Example 1.*

Example 1: Assume a dryer having a capacity of 100 lb of gelatin per hour at 11.1 percent bone-dry basis. The initial moisture content is 228 percent bone-dry basis and the final moisture content is to be 32 percent bone-dry basis. The rate of production of bone-dry gelatin is 90.5 lb per hr. Supply air is available at 120 F dry-bulb, 85 F wet-bulb, with make-up air at 80 F dry-bulb and 65 F wet-bulb. Air is exhausted from the dryer at 100 F dry-bulb and 84.5 F wet-bulb.

Find (1) the required amount of make-up and exhaust air, and (2) the percentage of recirculated air.

Solution: Refer to psychrometric chart, Fig. 5, and obtain the humidity ratio of the make-up air and of the exhaust air. In order to maintain a steady-state condition in the dryer the water which is evaporated from the material must be carried away by the exhaust air. Therefore, the difference between the humidity ratio of the exhaust air and that of the make-up air (known as pickup) is equal to the water evaporated from the material divided by the pounds per hour of dry air in the exhaust.

Step 1: The moisture in exhaust air at 100 F dry-bulb and 84.5 F wet-bulb is found to be 0.022 lb per pound of dry air. The moisture in make-up air at 80 F dry-bulb and 65 F wet-

bulb is found to be 0.010 lb per pound of dry air. Therefore the moisture pickup is 0.022–0.010, or 0.012 lb per pound of dry air.

The amount of water evaporated in the dryer = 90.5 (2.28 – 0.32) = 177 lb per hr.

The weight of dry air, make-up and exhaust, required to remove the water of evaporation = $\frac{177}{0.012}$ = 14,750 lb of dry air per hour.

Step 2: Let x = percentage of recirculated air and $100 - x$ = percentage of make-up air.

Then $\frac{x}{100}$ (humidity ratio of exhaust and recirculated air) + $\frac{100 - x}{100}$ (humidity ratio of make-up air) = humidity ratio of supply air. Hence, $\frac{x}{100}$ (0.022) + $\frac{(100 - x)}{100}$ (0.010) = 0.018 at 120 F dry-bulb, 85 F wet-bulb, or x = 66.7 percent recirculated air and therefore the required make-up air = 33.3 percent.

DETERMINATION OF DRYING TIME

In most drying problems it is necessary to obtain the drying time of a particular material. Three methods are listed below in order of preference.

1. Conduct tests in a laboratory dryer simulating conditions in the commercial machine, or obtain performance data directly from the commercial machine.
2. If the specific material is not available, obtain drying data on similar material by either of the above methods. This is subject to the experience and judgment of the investigator.
3. Estimate drying time from the theoretical equations which have been considered under *Mechanism of Drying*.

In designing commercial equipment the usual procedure is to conduct tests in a laboratory dryer in which commercial operating conditions are simulated. A representative sample in the physical form of the material as used in the laboratory tests should be identical to that which will be encountered in commercial operation. Otherwise the test results may not be an accurate indication of the drying characteristics of the commercial material.

In many cases where it is impracticable to run laboratory tests, commercial drying data are derived from the experience of the equipment manufacturer. This should not be overlooked since it can be an important source of data which would otherwise have to be obtained by less reliable methods.

Method 3, estimating drying time from the theoretical equations, should be used only as a last resort because it yields only approximate values.

Commercial Drying Time

The first step in selecting a commercial dryer is to apply the estimated drying time to determine the size of the commercial machine necessary for a given capacity. If the drying time has been derived from laboratory tests it is necessary to consider the following factors:

1. In a laboratory dryer considerable drying may have been *effected* by radiation and conduction of heat to the material, whereas in a commercial dryer these factors are usually negligible.
2. Humidity conditions in a commercial dryer may be higher than in a laboratory dryer. For drying operations with controlled humidity, this factor can be eliminated by duplicating the commercial humidity condition in the laboratory dryer.
3. Operating conditions usually cannot be kept as uniform in a commercial dryer as in a laboratory dryer.
4. Because of the relatively small sample used, the test material may not be representative of the material handled commercially.

In consideration of the preceding factors 1 to 4, it should

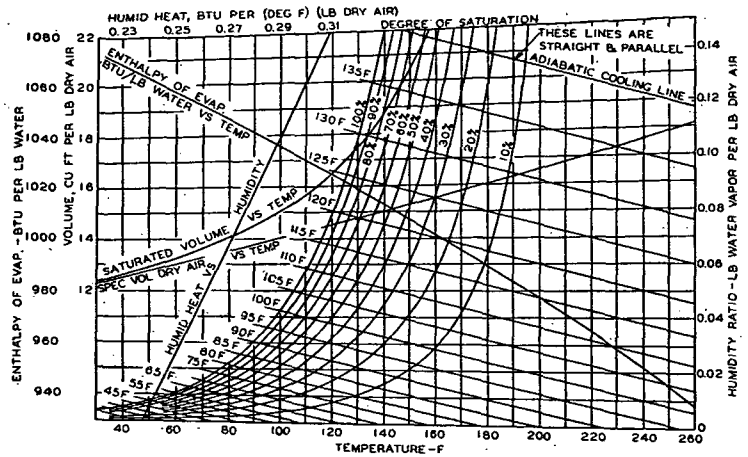


Fig. 5 .... Psychrometric Chart for Drying Calculations[5]

be stressed that the test drying time must be corrected to suit commercial conditions. This is a matter of experience and judgment on the part of the designer.

DRYER CALCULATIONS

To determine preliminary cost estimates for a commercial dryer it is usually necessary to make the following calculations:

Circulating air. The amount of circulating or supply air required is established by the optimum air velocity with reference to the material. This may be obtained from laboratory tests or previous experience, keeping in mind that there is also an optimum moisture pickup for the air, as explained in the section Application of Hygrometry to Drying.

Make-up and Exhaust. The amount of make-up and exhaust air required to maintain steady-state conditions within the dryer is discussed under the section Application of Hygrometry to Drying. In any continuously operating dryer, the relation between moisture content of the material and quantity of make-up air is given by Equation 6

$$G_T(W_2 - W_1) = M(w_1 - w_2) \text{ where } W_2 \text{ is constant.} \quad (6)$$

where

G_T = dry air supplied as make-up air to the dryer, pounds per hour.
M = stock dried in a continuous dryer, pounds per hour.
W_1 = humidity ratio of entering air, pounds of water vapor per pound of dry air.
W_2 = humidity ratio of leaving air, pounds of water vapor per pound of dry air.
w_1 = moisture content of entering material dry basis, pounds of water per pound.
w_2 = moisture content of leaving material dry basis, pounds of water per pound.

In batch type dryers, the drying operation is given by Equation 7

$$G(W_2 - W_1) = M_1 \frac{dw}{d\theta} \quad (7)$$

where

M_1 = weight of material charged in a discontinuous dryer, pounds per batch.
$\frac{dw}{d\theta}$ = the instantaneous rate of evaporation corresponding to w. W_2 is a variable during a portion of the cycle.

In actual practice the quantity of make-up air supplied is kept constant and is based on the average evaporation rate. Equation 7 then becomes identical with Equation 6 where $M = \frac{M_1}{\theta}$. Under this condition the humidity in the batch dryer will vary from a maximum to a minimum during the drying cycle; whereas, in the continuous dryer the humidity at any given point is constant with constant load.

Heat Balance. In order to estimate the fuel requirements of a dryer it is usually necessary to make a heat balance, consisting of the following:

1. Radiation and convection losses from dryer.
2. Heating of the commercial dry material to the leaving temperature (usually estimated).
3. Vaporization of the water being removed from the material (usually considered to take place at the wet-bulb temperature).
4. Heating of the vapor from the wet-bulb temperature in the dryer to the exhaust temperature.
5. Heating of the total water in the material from the entering temperature to the wet-bulb temperature in the dryer.
6. Heating of the make-up air from its initial temperature to the exhaust temperature.

The energy absorbed by items 1 to 6 must be supplied by the fuel. The proper selection and design of the heating equipment is an important part of the overall design of the dryer.

DRYING METHODS AND EQUIPMENT

Drying systems are sometimes classified according to the method of heat transfer that is employed, since the entire problem of drying resolves itself into individual problems of heat transfer and the thermodynamics of air and water vapor. The methods of heat transfer are radiation, conduction, and convection. Many types of dryers have been built on these principles for different purposes.

Drying systems can also be classified, according to the method of product handling, as batch operation, semi-continuous, and continuous.

The method of supporting or transporting the material to be dried, efficiency of operation, cost of investment, and the method of applying heat to the material are to be considered in selecting the equipment. Usually the final selection is a compromise between mechanical design, heat efficiency, quality, and product loss.

A systematic procedure is often justified when selecting a dryer for a given process. A general procedure will consist of:

1. Survey of suitable dryers
2. Preliminary cost estimates of various types
   a. Initial investment
   b. Operating cost
3. Drying tests should be conducted in prototype or laboratory units of the most promising equipment available. Sometimes a pilot plant is justified.
4. Summary of tests to evaluate quality and samples of the dried products.

Some items may overshadow the operating cost, or the investment cost, such as:

1. Product quality, which should not be sacrificed.
2. Dusting, solvent, or other product losses.
3. Space limitation.
4. Bulk density of product which may affect packaging cost.

A comprehensive bibliography for special reading will be found in the article by S. J. Friedman.[6]

Radiant Drying

In artificial systems, *radiating surfaces heated by steam*, electricity, or other means, afford a good method of heat distribution and control. Radiant heating sets up convection currents, and consequently, in low temperature dryers, only about one-third to one-half of the total heat for evaporation is actually supplied to the material by radiation. At high temperatures the radiation output increases rapidly, according to the *fourth-power law*. The total radiation may be computed by the equations and tables given in Chapter 5. In general, fins and irregular surfaces do not increase radiation, hence, the area to be used in calculations is the area of a smooth-surface envelope enclosing the radiating elements.

A certain amount of air circulation is required through a radiant dryer in order to carry off the vapor.

Radiant heat from infra-red lamps has been accepted by

certain industries as practicable for their specific problems. An example of successful application is found in the drying of laoquers.

Conduction Drying

Drying rolls or drums, Fig. 6,[7] flat surfaces, open kettles, and immersion heaters are examples of the direct-contact method. Intimate contact of the material with the heating surface is important, and in some cases agitation is desirable to increase the uniformity of heating or to prevent overheating.

A rather interesting method of conduction drying was put into practical use during the war for the drying of blood

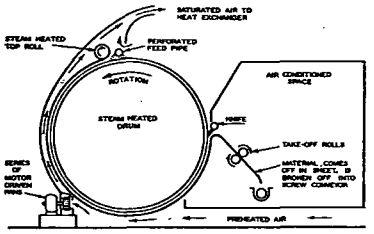


Fig. 6 .... Drum Dryer[7]

plasma, and has since been expanded to other fields such as the preservation of bacteria and other micro-organisms. This has come to be known as freeze drying or drying by sublimation. The material to be dried is first frozen and then placed in a high vacuum chamber connected to extremely low temperature condensers. The water is removed by vaporizing from the solid directly to the gas without ever becoming liquid.

Convection Drying (Direct Dryers)

A typical convection dryer is composed of an air-conditioning system, a material handling system, and the dryer housing. In normal practice it is desirable to recirculate as much as 95 percent of the air, from the standpoint of heat economy. There is an optimum point for the percentage of recirculation, depending on the temperature and evaporation rate.

A limited amount of convection drying takes place in almost any dryer such as those described in the preceding paragraphs. However, to be classified as a convection dryer the principal source of heat is the heated air or other gases circulated in the dryer. There are a number of mechanical means of accomplishing this circulation of air or gases, each of which has some particular virtue. Brief descriptions of some important types of convection dryers follow:

Rotary Dryers. These dryers are cylindrical drums which cascade the material being dried through the air stream. (See Fig. 7.) The dryers may be heated directly or indirectly, and the air circulation may be parallel or counterflow. A variation is the rotating-louver type dryer, which introduces the air beneath the flights thus securing very intimate contact.

Cabinet and Compartment Dryers. These are generally considered batch dryers wherein each charge is dried to completion before removal.[7] A wide range includes types from the heated loft with only natural convection, and usually poor and non-uniform drying, to the self-contained units with forced draft and properly designed baffles which give positive results. It is also possible to evacuate some of the systems for low temperature drying of delicate or hygroscopic materials. These dryers are usually loaded with material spread in trays to increase the exposed surface. The trays are loaded directly into the dryer or may be stacked on trucks which are wheeled in. (See Fig. 8.)

Tunnel Dryers. Tunnel dryers are a modification of the compartment dryer, and as a rule are continuous or semi-continuous in operation. Heated air or combustion gas is usually circulated by means of fans, although a few natural draft units are still in use. The material is handled on trays or racks on trucks, and moves through the dryer either intermittently or continuously. The air flow may be parallel, counterflow, or a combination of the two, obtained by center exhaust. Further, the air flow may be across the surface of the trays or up or down through the bed, or in any combination of directions. By reheating the air in this type of dryer or recirculating it, a high degree of saturation is achieved before exhausting the air. This reduces the waste of sensible heat.

A variation of this type dryer is the strictly continuous type having one or more mesh belts which travel through the dryer carrying the product, such as Fig. 9. Innumerable combinations of temperature, humidity, air direction, and velocity are possible. The labor requirement is low on such a dryer, as it can be loaded and unloaded mechanically. There is the disadvantage of hot air leaks at the entrance and exit, although these can be minimized by means of baffles or inclined ends where the material enters and leaves from the bottom.

Spray Dryers. In recent years the spray dryer has become important for the drying of liquids in many fields, especially in the food industry.

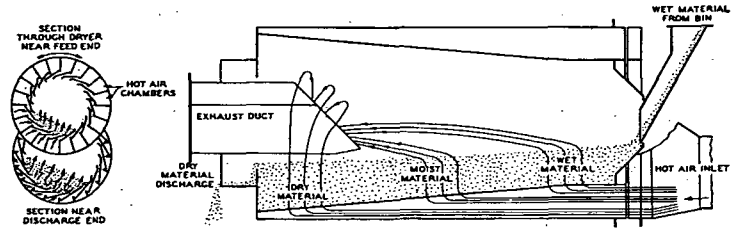


Fig. 7 .... Cross Section and Longitudinal Section Through Circulation Dryer[7]

The liquid is atomized by means of pressure nozzles, air jets, or centrifugal bowls into the air stream of a tower or chamber. Inlet air temperatures may run from 250–300 F up as high as 1200 F. Drying times are very short because of the minute particle size. Particles as small as 5 to 10 microns are formed in spray dryers. The dry powder is separated from the air by cyclone separators which are sometimes followed by cloth bags or scrubbing towers.

Because of the high inlet temperatures and the relatively large volume of air required, the efficiency of the spray dryer is relatively low and, consequently, it is seldom used for dilute solutions (less than 30 percent solid). Fig. 10 shows a typical arrangement for a spray drying system.[7]

A common and important feature of all spray processing is the direct conversion of the spray liquid to a granular product

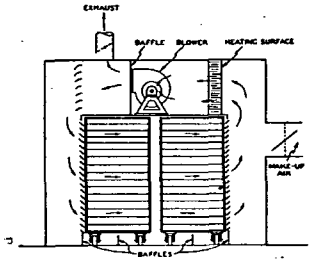


Fig. 8 .... Compartment Dryer, Showing Trucks, with Air Circulation[7]

suitable for packaging without grinding or other intermediate handling. Another aspect is the unusually high rate of drying attained. In a well designed system 15 to 30 seconds is a fair time for the passage of the sprayed particle through the drying zone; the particle temperature need not rise materially above the wet-bulb temperature of the drying air. This makes the process particularly adaptable to the drying of heat-sensitive material, some of its most important applications being the drying of milk, eggs, potato flour, soap, and blood.[8]

Tobacco Curing[9]

Considerable progress has been made recently in developing improved tobacco curing systems, through the application of heating equipment for maintaining uniform temperatures. Controlled heat and air circulation shortens the curing period and improves the quality of tobacco.

Suitable barn ventilation effected by bottom and ridge venting reduces the time and cost of curing. Orientation of barn with respect to the prevailing wind is helpful.

Tobacco Re-drying

In modern plants, tobacco is re-dried in continuous, conveyor type machines under controlled temperature and humidity conditions. One type of machine re-dries leaf tobacco in hands. This machine consists of a drying section, cooling section, and ordering section. As the hands are carried through the drying section on sticks, a high velocity of air is directed down through the heads, and a low velocity of

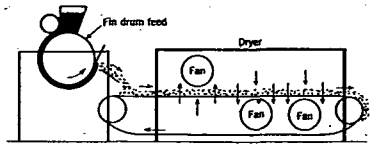


Fig. 9 .... Section of Continuous Dryer, Blow-Through Type

air passes over the body of the tobacco resulting in fast and uniform re-drying.

In the cooling section, the tobacco is quickly cooled and conveyed through the ordering section, where a controlled amount of moisture is added in this section which is maintained at almost saturated humidity ratio, so that the leaves can be packed for storage without the danger of breakage.

Agricultural Drying

Information on drying of agricultural products, such as milk, apples, flowers, tung nut, lumber, fruit juices, broom corn, grain, hay, peanuts, tobacco, and other farm crops, may be found in current issues of the *Agricultural Index*.[10]

SOLUTION OF TYPICAL DRYING PROBLEM

Since there are so many types of dryers that may be used, and so many special conditions surrounding each particular problem, it is usually recommended that those having experience with the dryer to be used be consulted. The following example, however, will serve as a guide for typical dryer calculations.

Example 2: Magnesium hydroxide is to be dried from 82 percent moisture on bone-dry basis to 4 percent moisture content on bone-dry basis. The production rate is to be 3000 lb per hr on 4 percent bone-dry basis. Previous experience indicates that a continuous single-conveyor through-circulation dryer with a fin-drum feed as illustrated in Fig. 9 should be used. The dryer is to be heated with steam at 50 psig. The optimum circulating air temperature is 160 F which is not limited by the existing steam pressure.

Step 1: Laboratory tests or previous experience indicate that the material enters the dryer at a temperature of 60 F, with an initial moisture content of 82 percent bone-dry basis. The test drying time is 25 min and the final moisture content is 4 percent bone-dry basis. The temperature of make-up air is 70 F dry-bulb and 60 F wet-bulb. The temperature of circulating air is 160 F dry-bulb and 100 F wet-bulb. The air velocity down through the preformed bed is 250 fpm, and the static pressure drop through the bed is 0.4 in. of water. The dryer

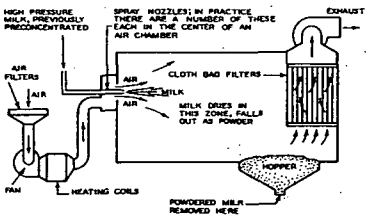


Fig. 10 .... Spray Dryer of the Pressure-Spray Rotary Type[7]

bed is to be loaded with 6.820 lb per sq ft of bone-dry material. Depth of the bed is to be 4 in.

Step 2: Previous experience indicates that the commercial drying time is 70 percent greater than the test time obtained in the particular laboratory setup used.

Therefore, the commercial drying time = $1.7 \times 25 = 42.5$ min.

Step 3: In order to dry the desired 3000 lb per hr of material the holding capacity of the dryer is $3000 \times \frac{42.5}{60} = 2125$ lb at 4 percent bone-dry basis.

The required conveyor area is $\frac{2125}{6.820} = 312$ sq ft. Assuming that a perforated plate conveyor with an 8 foot effective width is used, the length of the drying zone is $\frac{312}{8} = 39$ ft.

Step 4: The amount of water entering the dryer is $\frac{3000}{1.04} \times \frac{82}{100} = 2370$ lb per hr, while the amount of water leaving the dryer is $\frac{3000}{1.04} \times \frac{4}{100} = 115$ lb per hr. Thus, the evaporation rate in the dryer is $2370 - 115 = 2255$ lb per hr.

Step 5: Since the air circulation is perpendicular to the perforated plate conveyor, the total quantity of air that must be circulated equals the air velocity (based on the face area) multiplied by the conveyor area. Thus,

$$\text{Supply air} = 250 \times 312 = 78{,}000 \text{ cfm}$$

From Fig. 5 the humidity ratio of the supply air at 160 F dry-bulb, 100 F wet-bulb is 0.0285 lb per lb dry air. The specific volume of the supply air is 16.33 cu ft of moist air per lb of dry air, from Table 2 and Equation 27 of Chapter 3.

The quantity of dry air circulated is

$$\frac{78{,}000 \times 60}{16.33} = 286{,}500 \text{ lb per hr.}$$

Step 6: The amount of moisture pickup is $\frac{2255}{286{,}500} = 0.0079$ lb per lb of dry air. The humidity ratio of the exhaust air is $0.0079 + 0.0285 = 0.0364$ lb per lb dry air.

Substitute in Equation 6 and solve for G, the mass velocity of dry air, to determine the required quantity of make-up air. The humidity ratio of the make-up air is 0.0086 lb per lb dry air, from Fig. 5.

$$G\,(0.0364 - 0.0086) = \left(\frac{3000}{1.04}\right)\left(\frac{82 - 4}{100}\right)$$

$$G = 81{,}000 \text{ lb dry air per hr.}$$

Therefore,

$$\text{Make-up air} = \frac{81{,}000}{286{,}500} = 28.2 \text{ percent.}$$

$$\text{Recirculated air} = 71.8 \text{ percent.}$$

Step 7: Heat Balance

Sensible heat of material $= M(t_{m2} - t_{m1})c_m$
$= \frac{3000}{1.04}(100 - 60)\,0.3$
$= 34{,}600$ Btu per hr

Sensible heat of water $= Mw_1(t_w - t_{m1})c_w$
$= 2370\,(100 - 60)\,1.0$
$= 94{,}800$ Btu per hr

Latent heat of evaporation $= M(w_1 - w_2)H$
$= 2255 \times 1037$
$= 2{,}338{,}400$ Btu per hr

Sensible heat of vapor $= M(w_1 - w_2)\,(t_2 - t_w)c_v$
$= 2255\,(160 - 100)\,0.45$
$= 60{,}900$ Btu per hr

Required heat for material $= 2{,}528{,}700$ Btu per hr

The temperature drop $(t_2 - t_1)$ through the bed is

$$\frac{\text{Required heat}}{\text{Supply air, lb per hr} \times c_a} = \frac{2{,}528{,}700}{286{,}500 \times 0.24} = 37 \text{ deg}$$

Therefore, the exhaust air temperature is $160 - 37 = 123$ F

Required heat for make-up air $= G_T(t_2 - t_1)c_a$
$= 81{,}000\,(123 - 70)\,0.24$
$= 1{,}030{,}300$ Btu per hr.

The total heat required for material and make-up air is

$$2{,}528{,}700 + 1{,}030{,}300 = 3{,}559{,}000 \text{ Btu per hr.}$$

Additional heat must be provided for the radiation and convection losses, which may be calculated from the known construction of the dryer surfaces.

LETTER SYMBOLS USED IN CHAPTER 53

A = area of heat transfer and evaporation, square feet per pound of material.
a = drying area, square feet per cubic foot of bed volume.
c_a = specific heat of air, Btu per (pound) (Fahrenheit degree).
c_m = specific heat of material, Btu per (pound) (Fahrenheit degree).
c_s = humid heat, Btu per (pound of dry air) (Fahrenheit degree).
c_v = specific heat of water vapor, Btu per (pound) (Fahrenheit degree).
c_w = specific heat of water, Btu per (pound) (Fahrenheit degree).
d = diffusivity of the liquid or vapor, square feet per hour.
D_p = average diameter of particle, feet.
G = mass velocity of dry air, pounds per (hour) (square foot).
G_T = dry air supplied as make-up air to the dryer, pounds per hour.
H = enthalpy of evaporation of water at wet-bulb temperature, Btu per pound.
h = film heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
h_c = coefficient of heat transfer by convection, Btu per (hour) (square foot) (Fahrenheit degree).
h_r = coefficient of heat transfer by radiation, Btu per (hour) (square foot) (Fahrenheit degree).
h_t = total gas film heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
k_f = gas film thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot).
k_g = mass transfer coefficient, pounds per (hour) (square foot) (atmosphere).
L = one-half material thickness, feet.
M = weight of stock dried in a continuous dryer, pounds per hour.
M_1 = weight of stock charged in a discontinuous dryer, pounds per batch.
$\Delta p = p_s - p_s$ = vapor-pressure difference, atmospheres.
p_a = partial pressure of water vapor in air, atmospheres.
p_s = vapor pressure of water at t_s, atmospheres.
t_a = air or gas temperature, Fahrenheit.
t_s = temperature of particle, solid, or surface of evaporation, Fahrenheit.
t_w = wet-bulb temperature of drying air, Fahrenheit.
t_1 = make-up air temperature, Fahrenheit.
t_2 = temperature of air entering stock, Fahrenheit.
t_3 = exhaust air temperature, Fahrenheit.
t_{m1} = entering stock temperature, Fahrenheit.
t_{m2} = leaving stock temperature, Fahrenheit.
$\Delta t = (t_a - t_s)$ = temperature difference between air and surface of evaporation, Fahrenheit.
Δt_m = logarithmic mean between temperature entering and leaving the bed, and the wet-bulb temperature, Fahrenheit.
W_1 = humidity ratio of entering air, pounds of water vapor per pound of dry air.
W_2 = humidity ratio of leaving air, pounds of water vapor per pound of dry air.
w = moisture content on dry basis at any time θ, pounds of water per pound.
w_c = critical moisture content, pounds water per pound dry material.
w_d = water content, dry basis, of the drop as it enters the drying chamber, pounds per pound of dry solid.
w_e = moisture content at equilibrium with external conditions, pounds per pound dry material.
w_o = initial moisture or moisture content at start of diffusional period, pounds per pound dry material.
$\frac{dw}{d\theta}$ = drying rate, pounds of water per (hour) (pound dry material).
$\left(\frac{dw}{d\theta}\right)_c$ = constant drying rate, pounds per (hour) (pound dry dry material).
$\left(\frac{dw}{d\theta}\right)_f$ = falling rate, pounds water per (hour) (pound of dry stock).
θ = time, hours.
ρ_s = bulk density of dry granular bed, density of dry particle, pounds per cubic foot.

REFERENCES

[1] W. R. Marshall, Jr. and S. J. Friedman: Drying (*Perry's Chemical Engineers' Handbook*, McGraw-Hill Co., New York, 1950, 3rd ed.). Indicated material supplied by W. R. Marshall, Jr. and S. J. Friedman, authors of the Section on Drying in the Third Edition of the *Chemical Engineers' Handbook*. Permission to use this material has been kindly granted to THE GUIDE by the Editor, John H. Perry, and by McGraw-Hill Book Company, publishers of the *Chemical Engineers' Handbook*.

[2] W. R. Marshall, Jr: The drying of foods (*Heating, Piping and Air Conditioning*, September to December 1942, also November and December 1943).

[3] C. B. Shepherd, C. Hadlock, and R. C. Brewer: Drying materials in trays (*Industrial and Engineering Chemistry*, April 1938).

[4] B. W. Gamson, G. Thodos, and O. A. Hougen: Heat, mass and momentum transfer in the flow of gases through granular solids (*American Institute of Chemical Engineers Transactions*, 1943).

[5] W. M. Grosvenor: Calculations for dryer design (*American Institute of Chemical Engineers Transactions*, Vol. 1, 1908, p. 184).

[6] S. J. Friedman: Steps in the selection of drying equipment (*Heating and Ventilating*, February 1951, p. 95).

[7] What the air conditioning engineer should know about drying (*Heating and Ventilating*, December 1942).

[8] B. B. Fogler and R. V. Kleinschmidt: Spray drying (*Industrial and Engineering Chemistry*, December 1938).

[9] *Tobacco Curing* (*Virginia Agricultural Experiment Station Technical Bulletin* 116, January 1951).

[10] *Agricultural Index* (H. B. Wilson Co., New York).

BIBLIOGRAPHY

W. H. Carrier: The temperature of evaporation (ASHVE TRANSACTIONS, Vol. 24, 1918, p. 25).

J. A. Goff and S. Gratch: Thermodynamic properties of moist air (ASHVE TRANSACTIONS, Vol. 51, 1945, p. 125).

C. F. Prutton and C. O. Miller: Factors influencing the performance of rotary dryers (*American Institute of Chemical Engineers Transactions*, February, August 1942).

W. R. Marshall and O. A. Hougen: Drying of solids by through circulation (*American Institute of Chemical Engineers Transactions*, 1942).

A. Weisselberg: Factors that influence dryer performance (*Chemical and Metallurgical Engineering*, August 1932).

O. A. Hougen: Typical dryer calculations (*Chemical and Metallurgical Engineering*, January, March 1940).

W. K. Lewis, W. H. Carrier, A. E. Stacey, Jr., R. S. Fleming, R. G. Mets, G. B. Ridley, C. O. Lavett, and D. J. Van Marle: Symposium on drying (*Industrial and Engineering Chemistry*, May 1921, p. 427).

S. J. Friedman and W. R. Marshall, Jr.: Studies in rotary drying, I and II (*Chemical Engineering Progress*, 1949).

Symposium on drying (*Industrial and Engineering Chemistry*, 1938).

H. D. Tiemann: *Principles of Drying Lumber and Humidity Diagram* (*Forest Service Bulletin* 104, 1912).

T. K. Sherwood: *The Drying of Solids* (*Massachusetts Institute of Technology Bulletin* Nos. 237, 247, 258).

T. P. Brown: Radiant energy drying with heat lamps (*Metal Industry*, December 1939, p. 607).

E. W. Flosdorf: Drying by sublimation (*Food Industry*, January 1945, p. 607).

F. H. Slade: Evaporative drying system (*Food Manufacturing*, March 1943, p. 70).

I. R. Berkness: High frequency methods in gluing and drying wood (*Wood Products*, Vol. 45, 1940, p. 12).

T. K. Sherwood: Development of the unit operations of chemical engineering: drying (*Chemical and Metallurgical Engineering*, Vol. 42, p. 214).

D. W. Biocheno: The spray dryer—its possibilities in industry (*Food Manufacturing*, June 1944, p. 195).

L. E. Stout, K. J. Caplan, and W. G. Baird: Mechanism and rate of drying by near-infra-red radiation (*American Institute of Chemical Engineers Transactions*, Vol. 41, 1945, p. 283).

W. B. Van Arsdel: Some engineering problems of the new vegetable dehydration industry (ASHVE TRANSACTIONS, Vol. 49, 1943, p. 49).

V. W. Sherman: Electronic dehydration of foods (*Electronics*, Vol. 17, 1944, p. 94).

Air Conditioning and Engineering (American Blower Co., 1935).

Drying in Industrial Plants (J. O. Ross Company).

Badger and McCabe: *Elements of Chemical Engineering* (McGraw-Hill Co., New York, 1931).

Fan Engineering (Buffalo Forge Co.).

M. Hirsch: *Die Trockentechnik* (Julius Springer, Berlin, 1932).

Drying (*Kent's Mechanical Engineers Handbook*).

W. H. Carrier: Drying (*Mark's Mechanical Engineers Handbook*).

A. Koehler and R. Thelen: *Kiln Drying of Lumber* (McGraw-Hill Co., New York, 1926).

H. D. Tiemann: *Kiln Drying of Lumber* (J. B. Lippincott, New York, 1920).

O. A. Hougen, H. J. McCauley, and W. R. Marshall: Limitations of diffusion equations in drying (*American Institute of Chemical Engineers Transactions*, Vol. 1, 1940, p. 183).

V. P. Victor: An introduction to convection drying and drying calculations (*Heating and Ventilating*, December 1944, p. 67).

R. R. Bennett and S. N. Hawks: *Tobacco Curing* (*University of North Carolina Extension Circular* 332, January 1949).

L. S. O'Bannon: *Principles of Burley Tobacco Barn Operation* (*University of Kentucky Bulletin* 501, May 1947).

E. M. Matthews, M. H. McVickar, and R. B. Davis, Jr.: *Fuels for Curing Tobacco* (*Virginia Agricultural Experiment Station Bulletin* No. 396, May 1946).

N. C. Teter and E. G. Moss: *Heating Tobacco Barns with Stokers* (*North Carolina Agricultural Experiment Station Bulletin* No. 352, May 1945).

Irvin Lavine and O. D. Sutherland: Revised psychrometric chart assists high temperature design (*Chemical and Metallurgical Engineering*, April 1928, p. 226).

C. R. Wilkie and O. A. Hougen: Mass transfer in the flow of gases through granular solids extended to low modified Reynolds numbers (*American Institute of Chemical Engineers Transactions*, Vol. 41, 945, p. 445).

W. R. Marshall: *Current Status of the Theory and Practice of Drying* (University of Wisconsin, *Engineering Experiment Station Reprint* No. 265).

CHAPTER 54

OWNING AND OPERATING COSTS

Fixed Charges: Amortization, Interest, Taxes, Insurance, Rent; Maintenance Costs; Labor for Operation; Energy and Water Costs: Operating Refrigerating Equipment, Condenser Water, Heating

THE total cost for the use of heating, ventilating, and air-conditioning systems may be divided into two classifications. The first of these is the relatively fixed and unvarying expense of ownership, and the second is the variable and somewhat controllable expenditure for actual operation of the equipment. Owners and prospective purchasers of this equipment are particularly concerned with both since the actual expenditure is generally predicated upon the possible return on the investment resulting from increased patronage, greater efficiency on the part of the employees, meeting of competition, or the improvement and maintenance of quality in a manufactured product. These costs may be grouped under four headings: (1) *Fixed Charges*, (2) *Maintenance Costs*, (3) *Labor for Operation*, and (4) *Energy and Water Costs*.

FIXED CHARGES

Fixed charges are the annual expenses arising from the ownership of the installation, including use of the owner's money, and protection of the equipment in the form of insurance. Such costs are usually unchanged from year to year regardless of whether the equipment is in or out of service. Fixed charges may be grouped under five headings: (1) *Amortization*, (2) *Interest*, (3) *Taxes*, (4) *Insurance*, and (5) *Rent*.

Amortization

As air-conditioning equipment becomes older, its ability or capacity to perform present and future service is reduced. Provision must be made for the owner to recover each year a portion of the initial value of the equipment as an expense to be charged against revenue, or to amortize the cost. This decrease in the value of property is usually determined by an accountant or engineer using some theoretical method and will depend upon (1) the total first cost, and (2) the amortization period.

The total first cost of an installation is the actual capital expenditure required to buy and install the air-conditioning, heating, or ventilating system ready for operation. It represents the first cost of the equipment combined with additional costs incurred because of the installation of the system.

The first cost of air-conditioning, heating, or ventilating systems includes the following:

1. Heat producing equipment including boilers, burners, controls, etc.
2. Heat distributing equipment including direct radiation, piping, etc.
3. Air handling equipment including fans, air heaters, air conditioners, filters, controls, etc.
4. Air distribution system including ducts, outlets, grilles, etc.
5. Refrigerating equipment including piping, pumps, etc.
6. Water conservation devices including towers, evaporative condensers, etc.
7. Insulation of pipes, ducts, and equipment.
8. Building alterations, furring-in ducts and pipes, structural work, etc.

To estimate the first cost of any system prior to installation, the best procedure is to determine the heating and cooling load, and then after thorough engineering study, select the type of system. The installed cost of systems will vary widely, depending upon the type of equipment selected, and the design of the distribution system, equipment and labor costs in the locality, demands for the particular installation, etc. A reasonably approximate cost may be determined for a selected design in a given locality from the sum of the estimated unit costs of the component parts of the system. A reasonably precise estimate of the cost of the components may be obtained from cost records of recent installations of a comparable design, or from quotations submitted by manufacturers and contractors.

Approximate costs are given in Table 1 for mechanical air-handling systems including heating and cooling coils for distributing 1½ cfm per square foot of floor area and refrigeration equipment based upon a requirement of one ton for every 333 sq ft of floor area. Different types of service demand may change the floor area per ton of refrigeration and air volume per square foot between wide limits and, consequently, the table should be used with caution.

Other first costs may be incurred because of the installation of the air-conditioning or heating and ventilating system. These will include such items as electrical work, plumbing, miscellaneous piping, building alterations, cutting, patching, furring-in of ducts or pipes, foundations, structural supports, remodeling or redecorating after installation, consulting engineer's fees, licenses, and permits. As these costs vary widely no approximations are practicable. Therefore, each case must be considered individually. If a quotation can be secured from a manufacturer or contractor covering the complete job, it will usually include the items that have been mentioned.

The length of the amortization period to be used depends upon (1) the type and remaining life of the building or

Table 1 .... Cost Dollars per Ton for a Complete System, Including Heating and Cooling Coils, Fan, Ducts, Refrigeration Equipment, Temperature Controls, etc.*

1	2	3	4	5	6	7	8	9	10	11	12	13
			Unit Costs		Total Cost[a]			Unit Costs[b]				
Refrigeration Tons	Floor Area Sq Ft	Air Distributed Cu Ft	Refrigeration Equipment per Ton	Ducts, Fans, Heating Coils, etc. per Cfm	Refrigeration Equipment	Air Handling Equipment	Total	Per Ton	Per Sq Ft Floor	Per Cfm	Evaporative Condensers Per Ton[c]	Cooling Towers Per Ton[a]
25	8,333	12,500	250	1.05	6,250	13,100	19,350	775.00	2.32	1.55	190.00	
50	16,666	25,000	224	0.98	11,200	24,500	35,700	730.00	2.15	1.43	155.00	
75	25,000	37,500	212	0.95	15,900	35,700	51,600	690.00	2.07	1.38	120.00	
100	33,332	50,000	204	0.92	20,400	46,000	66,400	664.00	1.99	1.33	90.00	52.00
150	50,000	75,000	195	0.91	29,200	68,200	97,400	650.00	1.95	1.30	60.00	50.00
200	66,664	100,000	192	0.90	38,200	90,000	128,200	641.00	1.93	1.28		48.00
250	83,333	125,000	187	0.88	46,800	110,000	156,800	628.00	1.88	1.25		46.00
300	100,000	150,000	182	0.86	54,600	129,000	183,600	612.00	1.84	1.23		44.00
400	133,328	200,000	165	0.84	66,000	168,000	234,000	585.00	1.76	1.17		42.00
500	166,666	250,000	158	0.82	79,000	205,000	284,000	568.00	1.71	1.14		40.00

* These cost figures (based on 1954 prices which may be adjusted by using current building cost index) cover only the average type of comfort air conditioning in open spaces as in department stores, or for industrial air conditioning involving large areas. Where a multiplicity of small spaces must be individually controlled, as in office buildings, hotels, and apartment houses, these costs do not apply.

[a] Includes only final water, drain, and electrical connections within the equipment room.

[b] Columns 9, 10, and 11 represent costs where well water or city water is available for condensing purposes. Where conservation of water is required, add column 12 or 13 to column 9 to obtain total cost per ton.

[c] Does not include cost of structural reinforcing if required, to support the weight of cooling towers or evaporative condensers.

space for which the system is to be used; (2) the type of equipment to be employed as a part of the system; (3) the character of the business; and (4) the lease or ownership conditions.

Depreciation, due to deterioration or obsolescence, must be considered in arriving at the amortization period. Maintenance and deterioration usually have the effect of offsetting one or the other. If a long depreciation period is to be used, then the item for maintenance, repair, and the replacement of wearing parts must be greater than for a short depreciation period.

In determining the length of the amortization period, the owner's accounting practices will have considerable bearing upon the number of years used in the calculation. For taxation purposes, this period depends upon the use and the service to which the equipment is applied. At the present time indications are that varying interpretations on depreciation will be made by the Bureau of Internal Revenue. While most air-conditioning equipment may be considered as having a normal useful life of 20 years for depreciation purposes, no final value for depreciation should be determined without consultation with the owner's tax consultant or perhaps referral to Schedule F which is issued by the Bureau of Internal Revenue and summarized in Table 2.

Interest

The interest charged to the cost of ownership may be based upon the average interest rate for the period during which the first cost of the equipment will be amortized. While some accountants do not include interest in the annual fixed charges and consider it a negligible item, the money invested in air-conditioning, heating, or ventilating equipment, must be either borrowed or diverted from the owner's funds and converted to the purpose of making the installation. Whether it is borrowed or taken from surplus funds, the money that might have been earned as interest is properly chargeable to the operation of the system.

Table 2 .... Probable Useful Life of Equipment*

	Life in Years
1. Heat Producing Equipment	
(a) Boilers	20
(b) Stokers and burners	20
2. Heat Distributing Equipment	
(a) Piping—copper	same as bldg.
(b) Piping—iron	20
(c) Radiation—concealed	25
(d) Radiation—direct	25
(e) Valves and specialties	10
3. Air Handling Equipment	
(a) Filters—automatic	20
(b) Heating and cooling coils	20
(c) Spray humidifiers and dehumidifiers	10
(d) Fans	15
(e) Air-conditioning units	10
(f) Motors	20
(g) Electrical starting equipment	20
(h) Pneumatic control systems	15
(i) Electric control systems	15
4. Air Distributing Equipment	
(a) Ductwork	same as bldg.
(b) Outlets, grilles	15
(c) Duct insulation	20
5. Refrigerating Equipment	
(a) Centrifugal refrigerating machines	20
(b) Reciprocating refrigerating machines	20
(c) Motors and starters	20
(d) Piping—copper	20
(e) Piping—steel	20
(f) Pumps	20
6. Water Saving Devices	
(a) Evaporative condensers	15
(b) Cooling towers	15
(c) Wells	25

* Taken from U. S. Bureau of Internal Revenue *Schedule of Probable Useful Life*, revised 1942.

Table 3 .... Owning and Operating Cost

First Cost	
Cost of mechanical system	
Other costs	
First Cost (FC)—Total	
Annual Fixed Charges	
Amortization—Depreciation period Y years	
Interest rate i%	
Amortization and Depreciation $\frac{FC}{Y}$ =	
Interest: $\frac{Y+1}{2Y} \times i \times$ cost =	
Taxes	
Insurance	
Rent	
Annual Fixed Charges: (Total)	
Annual Maintenance Costs	
Replacement or servicing of air filters	
Outside maintenance service	
Water treatment	
Lubricating oil and grease	
Painting for corrosion protection or other purposes	
Replacement of worn parts	
Refrigerant	
Wages of engineer or operator	
Annual Maintenance Cost—Total	
Annual Service Cost	
Electric power costs	
Fans	
Pumps—chilled water	
Pumps—condenser water	
Pumps—well water	
Cooling tower fans	
Cooling tower pumps	
Refrigeration machines	
Miscellaneous or other	
Gas	
Coal	
Oil—for boilers or Diesel engines	
Steam	
For Direct heating	
For Ventilation—preheaters	
For Ventilation—reheaters	
For Turbine driven equipment	
For Engine driven equipment	
Sewers	
Charges for discharging water into public drainage systems	
Condenser water	
Annual Service Costs: Total	
Summary	
Annual Fixed Charges	
Annual Service Costs	
Annual Maintenance Costs	
Annual Owning and Operating Costs—Total	

Table 4 .... Approximate Combined Operating and Maintenance Cost for Large Air-Conditioning Installations, Using High-Quality Equipment*

	Dollars per (Ton) (Year)
Repairs for refrigeration machinery	1.07
Refrigerant	0.40
Oil and grease	0.11
Painting (Water boxes and dehumidifiers)	0.45
Filters, clean and re-oil 4 times per year	1.50
Controls, outside service	0.26
Cleaning air conditioners	1.38
Total	5.17

* Estimated for 1955.

The formulas for computing interest and amortization are given in Table 3. This table also serves as a check list for the various items to be considered in calculating the cost of ownership and operation.

Taxes

The taxes that may be charged to the property as a result of the improvement due to the installation of an air-conditioning system, will vary according to the practice of the official agencies levying property taxes.

Insurance

Insurance against losses by fire is ordinarily secured by increasing the building fire insurance coverage to cover all or part of the first cost of installing the air-conditioning and other mechanical systems. Various types of extended coverage are available for protection against a number of other types of losses.

Boiler insurance may be extended to cover the air-conditioning equipment to protect against losses due to explosion, rupture of piping, and similar hazards. Since insurance rates vary widely depending upon the type of structure in which the equipment is located and the nature of the owner's business, the rates are usually set by rating organizations specializing in this work. Exact figures on insurance cannot be determined without consultation with the owner's underwriter.

Rent

If the equipment under consideration is to be located in rented or leased quarters, or if additional space must be rented, such expense becomes a fixed charge.

MAINTENANCE COSTS

Maintenance charges consist of expense for labor and material necessary to make repairs and replace parts, as well as cleaning, painting, inspection, etc., for the purpose of eliminating or minimizing the need for repairs. Generally, routine maintenance labor requirements will be the function of an operating engineer or staff. If the responsibility of this group may extend to other facilities beyond the equipment being discussed here, it is important that only an equitable share of the group's time be charged to maintenance.

Extraordinary repairs are quite often handled by separate maintenance divisions and the expense charged back to air conditioning, heating, and ventilating. In other cases, all maintenance is handled by outside service firms and the

Table 5 . . . . Equivalent Full Load Operating Hours of Refrigeration Equipment Used for Summer Cooling May 15 to Oct. 15,[a]

Application	Hr Open for Business	Atlanta	Boston	Chicago	Detroit	Los Angeles	New Orleans	New York	Philadelphia	Oklahoma City	St. Louis	Washington, D. C
Barber Shops	1280	1010	650	720	720	680	1080	830	860	1020	890	940
Department Stores	940	840	560	610	610	580	890	700	720	840	750	780
Drug Stores	2100	1630	950	1060	1060	980	1790	1280	1330	1650	1420	1530
Funeral Parlors	600	440	300	330	330	310	470	370	380	440	400	410
Offices	940	870	560	620	620	580	900	710	740	880	770	810
Restaurant (Short Hour)	1290	970	535	620	620	570	1060	760	800	980	830	880
Restaurant (Long Hour)	2100	1510	820	930	930	850	1690	1170	1210	1530	1300	1400
Speciality Shops (5 & 10)	1090	800	530	590	590	560	860	670	690	810	720	750
Theaters—Continuous	1500	1010	700	750	750	720	1080	850	870	1020	910	950
Theaters—Neighborhood	900	640	420	450	450	430	650	500	520	650	550	580

[a] *Modern Air Conditioning, Heating and Ventilating*, by W. H. Carrier, R. E. Cherne, and W. A. Grant (Pitman Publishing Corp., 1940, p. 73).

cost considered as maintenance. These costs vary considerably with factors such as the type of system and the proficiency of the installing and servicing organization. Therefore, any forecast of maintenance costs should include consideration of the equipment as a part of the engineering study. The charges should be based upon the entire period under study rather than the early years of operation when repairs may be expected to be at a minimum.

Various estimates of maintenance costs vary from 5 to 10 percent of the installation cost. Table 4 is based upon an average of 7½ percent for about a 75-ton installation. This probably would apply to most installations using high-quality equipment. The maintenance cost must be modified for size of job and quality of equipment. With lower-cost equipment, higher maintenance costs can be expected.

The accounting practices of the owner and the rules of the Bureau of Internal Revenue affect the charges for major overhauling or complete replacement which may restore the capital value of certain equipment items. In these cases the expenditure may not necessarily be charged as maintenance but will become a fixed charge spread over the remaining years of equipment life. Table 4 gives some approximate costs for maintaining large air-conditioning installations using high-quality equipment.

LABOR FOR OPERATION

In some cases wth the installation of automatic equipment, operating labor may be non-existent, but where such labor is required, the cost is readily calculated. Where operators are required, the expense is often considered maintenance, but since they may have other functions that are not properly charged to operation of the air-conditioning, heating, or ventilating installations, the charges should be properly allocated. The cost of experienced and competent operators is well justified in medium and larger plants by economies that can be achieved in maintenance and energy costs.

ENERGY AND WATER COSTS

Energy costs include the costs for power, water, steam, coal, oil, etc., consumed to operate the system.

From the selected equipment and type of installation, it is possible to segregate the relatively constant power loads and the total brake horsepower. Annual power cost can then be figured from the following formula:

$$\text{annual power cost} = \frac{0.746(\text{bhp})HR}{\eta} \qquad (1)$$

where

bhp = brake horsepower.
H = annual operating hours.
R = power rate, dollars per kwhr.
η = motor efficiency (decimal).

In using Equation 1 it must be pointed out that the electric rate, R, must reflect the proper combination of energy and demand rates. These vary widely between the utility companies, and sometimes the rate structure is such that it is largely the demand charge which determines the proper value of R to use in Equation 1.

Operating Refrigerating Equipment

In an air-conditioning system the refrigerating equipment is usually the largest power consuming item to be considered. Also, the prediction of operating cost is more difficult because the power required for summer cooling is affected by many factors of a variable nature.

Table 5 gives the equivalent full-load operating hours of refrigerating equipment used for summer cooling for the period of May 15th to October 15th. This table was calculated from the following equation:

$$H_e = m(b + cf) \qquad (2)$$

where

H_e = equivalent full load operating hours of refrigeration equipment used for summer cooling during period May 15 to October 15.
m = total hours during period May 15 to October 15 that the establishment is open for business.
b = fraction of maximum load from internal heat under average operating conditions.
c = fraction of maximum load that is due to external sources at maximum design conditions.
f = ratio of the number of hours for a particular city, when the outdoor wet-bulb exceeds 65 F, during the period June 1 to October 1 to the total number of hours during that same period. Total hours are assumed as 8 hr per day period for barber shops, department stores, funeral parlors, offices, short-hour restaurants, and specialty shops, and 12 hr per day period for drug stores, long-hour restaurants, and theaters.

It should be pointed out that certain southern cities may have seasons longer than the 5-month period indicated in Table 5. If it is desired to consider a longer season of operation, the ratio of full-load operating hours to hours open for business is smaller; in other words, the *refrigeration load factor* is lower. This is true because the extra increment of days added will be a relatively light load, since the table already includes the more severe part of the season.

The season electrical power cost for refrigerating equipment is then given by the following equation:

$$\text{season power costs} = \frac{0.746(\text{bhp})_l TH_e R}{\eta} \qquad (3)$$

where

$(\text{bhp})_l$ = brake horsepower per ton (see Fig. 1) for *average* load during period. (Due allowance should be made for poorer compressor efficiency at light load.)
T = maximum refrigeration design load, tons.
H_e = equivalent full load refrigeration operating time, hours (from Table 5).
R = power cost, including demand and energy charges, dollars per kwhr.
η = motor efficiency at average load (decimal).

In considering refrigeration power consumption, it should be noted that the use of weather records for a specific year may lead to large inaccuracies in estimating operating costs, since there may be wide variations from year to year, and therefore, average yearly weather records should be used rather than those for any individual year.

If the refrigeration compressor is steam-turbine driven, the same general method can be followed, taking into account average water rate per brake horsepower-hour and the cost of steam.

Condenser Water

Condenser-water cost estimates can also be based on equivalent full-load operating hours of the refrigeration equipment. The varying temperature of the water at its source, as well as the temperature of the discarded water, must, however, be taken into account. In general, when water is purchased, control is provided to hold the leaving water temperature (or condensing temperature) constant; and in such case the entering water temperature becomes the major variable. The gallons per minute per ton can readily be calculated for any water temperature rise.

The following equation for cost of condenser water is useful:

$$B = 0.060\, a\, T\, H_e\, C \qquad (4)$$

where

B = cost of water for refrigeration during period, dollars.
a = average gallons per (minute) (ton).
T = tons of refrigeration at maximum design load.
H_e = equivalent full load refrigeration operating hours (Table 5).
C = water cost, dollars per 1000 gal.

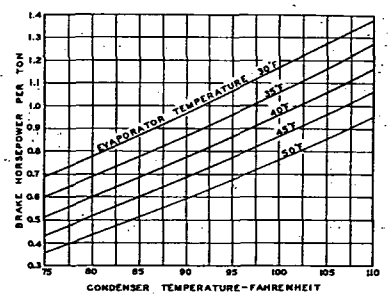
The average gallons per minute per ton must take into account the variable water temperature. When well water is used as a source, and entering and leaving temperatures are considered constant, the average gallons per minute per ton obviously are equal to the design gallons per minute per ton. However, when the source is river or lake water, its maximum seasonal temperature will generally be reached at the same time that the refrigeration load factor is highest. The average gallons per minute per ton should be calculated from known or estimated water temperatures, because they vary through the season. Maximum water main temperatures are given in Chapter 41, but should always be verified locally. In lieu of this tedious work, the average gallons per (minute) (ton) may be taken as 80 percent of design gallons per (minute) (ton) with reasonable accuracy, for the condition of variable temperature of entering water obtained from rivers and lakes.

Cooling towers and evaporative condensers virtually eliminate condensing water charges since the windage and evaporation losses are seldom over two or three percent of the water circulated.

The savings in water consumed often will not in itself justify the installation of water-economizing equipment since the cost for pumping and the fixed or ownership charges may be in excess of the annual cost for once-through condensing operation. If operating costs are to be the basis for selection, then fixed charges should be determined when studying application of this equipment which usually has a shorter life than the other components.

Usually the primary factors influencing the installation of water conservation equipment are the lack of an adequate water supply or local regulations intended to conserve an existing water supply.

Another factor influencing the installation of such equipment is the increasing trend toward the enactment of taxes and service charges for condensing water wasted into city sewers. Certain municipalities will remit the sewer taxes



Values given are representative of dichlorodifluoromethane (Refrigerant 12) reciprocating machines of about 25 tons capacity in air-conditioning applications. Requirements of smaller machines are usually higher, and for larger machines may be lower. Values shown are for liquid refrigerant at condenser temperature (no subcooling). Subcooling of the liquid may decrease these values approximately 0.3 percent to 0.5 percent for each Fahrenheit degree the liquid temperature is lowered.

Fig. 1 . . . . Typical Brake Horsepower Requirements for Refrigeration[a]

or service charge if the condensing water is submetered and discharged into a storm sewer or used for process purposes.

Heating

The method of estimating fuel consumption to balance the building heat loss is given in Chapter 37. It is important to include the fuel required to heat ventilation air as used in ventilating and air-conditioning systems. In estimating fuel consumption for ventilation air, the tendency of the conventional control systems to use less than the estimated quantity of outdoor air in cold weather should be considered in its effect in lowering fuel consumption. In addition, the heat required to accomplish winter humidifying must not be neglected, when this feature is included in the equipment.

BIBLIOGRAPHY

W. H. Carrier, R. E. Cherne, and W. A. Grant: Chapter IV (*Modern Air Conditioning, Heating and Ventilating*, Pitman Publishing Corp., New York, 1950).

W. A. Grant: Predicting operating hours of refrigeration equipment used in air conditioning (*Refrigerating Engineering*, July 1941).

R. E. Cherne: Cost of operation of refrigeration used in air conditioning (*Refrigerating Engineering*, December 1943).

Section VII, Chapter 45, Table 3, 1949 Cost data on variety chain store installations (*ASRE Data Book*, Applications Volume, 1954–55, American Society of Refrigerating Engineers).

CHAPTER 55

CORROSION AND WATER-FORMED DEPOSITS, CAUSES AND PREVENTION

Definitions, Classification and Characteristics of Water, Causes and Prevention of Scales and Sludges, Causes and Prevention of Slimes, Underwater Corrosion, Atmospheric Corrosion, Buried Pipe Lines, Handling Water Treating Chemicals, Legal Regulations

THE purpose of this chapter is to discuss some of the common problems arising from corrosion of heating and air-conditioning equipment and to suggest methods of preventing or reducing corrosion.

DEFINITIONS

Common terms used in the field of water treatment and corrosion are defined as follows:

Alkalinity. The sum of the carbonate, bicarbonate, and hydrate ions in water. Other ions such as phosphate or silicate may also partially contribute to alkalinity. Ordinarily the difference between the total hardness of a water and the alkalinity gives the amount of permanent or non-carbonate hardness present.

Anode. A positive electrode toward which negatively charged non-metallic ions migrate and at which reduction occurs in an electrolytic cell. In corrosion processes, the anode is usually the electrode having the greater tendency to go into solution.

Biological Deposits. Biological deposits[1] are water-formed deposits of biological organisms or the products of their life processes. Biological deposits may be microscopic in nature, such as slimes, or macroscopic, such as barnacles or mussels. Slimes are usually composed of deposits of a gelatinous or filamentous nature.

Cathode. A negative electrode toward which positively charged metallic ions migrate and at which reduction occurs in the electrolytic cell. In corrosion processes, the cathode is usually the electrode tending to resist corrosion.

Corrosion. Corrosion[2] is destruction of a metal by chemical or electrochemical reaction with its environment. In the corrosion process, the reaction products formed may be soluble or insoluble in the contacting environment. Insoluble corrosion products may deposit at or near the attacked area, or be carried along and deposited at a considerable distance therefrom.

Corrosivity. Corrosivity[3] is the capacity of an environment to bring about destruction of a metal by the process of corrosion. Corrosivity is a property of the environment, but it has no significance until the metal in question is specified.

Electrolyte. A solution through which an electric current flows.

Galvanic Corrosion. Corrosion generally resulting from the contact of two dissimilar metals in an electrolyte. It is characterized by an electron flow from the metal of higher potential (anode), to the metal of the lower potential (cathode), resulting in corrosion of the anodic metal. Galvanic cell corrosion may also result from the contact of two similar metals in an electrolyte of non-uniform concentration.

Hardness. The sum of the calcium and magnesium contents in water. It is this hardness that causes the water to resist the formation of soap lather. Temporary or carbonate hardness is that portion of the total hardness which can combine with the carbonate or bicarbonate ions. The balance of the hardness, principally that which will combine with the sulfate and chloride ions, is called permanent or noncarbonate hardness.

Inhibitor. A chemical substance added to an environment to reduce corrosion.

Ion. An electrically charged atom or group of atoms.

Passivity. The tendency of a metal to become abnormally inactive in a certain environment.

pH. The logarithm of the reciprocal of the hydrogen ion concentration of a solution. The *pH* value denotes the degree of acidity or alkalinity of a solution. The value of a neutral solution, such as pure water, is 7. Values below 7 are increasingly acid, while values above 7 are increasingly alkaline.

Polarization. An effect produced on the electrode of an electrolytic cell by products formed by the passage of current through the cell. Hydrogen is the product generally formed, although other corrosion products may also cause polarization. Polarization reduces corrosive activity by offering a resistance to current flow and developing an opposing electromotive force.

Scale. Scale[4] is a deposit formed from solution directly in place upon a confining surface. It is a deposit which will retain its physical shape when mechanical means are used to remove it from the surface on which it is deposited. Scale, which may or may not adhere to the underlying surface, is usually crystalline and dense, frequently laminated, and occasionally columnar in structure.

Sludge. Sludge[4] is a water-formed sedimentary deposit. It usually does not cohere sufficiently to retain its physical shape when mechanical means are used to remove it from the surface upon which it deposits. Sludge is not always found at the place where it is formed. It may be hard and adherent, and baked to the surface on which it has been deposited.

Water-Formed Deposits. A water-formed deposit[4] is any accumulation of insoluble material derived from water or formed by the reaction of water upon surfaces in contact with water.

Deposits formed from or by water in all of its phases may be further classified as scale, sludge, corrosion products, or biological deposits.

CLASSIFICATION AND CHARACTERISTICS OF WATER

Due to the many conditions of water encountered in industry,[4] it is frequently necessary to obtain expert advice and a prescribed treatment to make the water suitable for use in a given process. The heating and air-conditioning engineer is concerned chiefly with two broad classifications of water: (1) those waters containing various amounts of mineral impurities and (2) condensates that are relatively free from mineral impurities but contain various amounts

Table 1 .... Mineral Analyses Typifying Composition of Waters Available and Used Industrially in the USA

Substance	Unit	Location or Area[a,b] (1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
Silica	SiO_2	2	6	12	37	10	9	22	14	.....
Iron	Fe	0	0	0	1	0	0	0	2	.....
Calcium	Ca	6	5	36	62	92	96	3	155	400
Magnesium	Mg	1	2	8	18	34	27	2	46	1,300
Sodium	Na	2	6	7	44	8	183	215	78	11,000
Potassium	K	1	1	1	...	1	18	10	3	400
Bicarbonate	HCO_3	14	13	119	202	339	334	549	210	150
Sulfate	SO_4	10	2	22	135	84	121	11	389	2,700
Chloride	Cl	2	10	13	13	10	280	22	117	19,000
Nitrate	NO_3	1	...	0	2	13	0	1	3	.....
Dissolved Solids	.....	31	66	165	426	434	983	564	948	35,000
Carbonate Hardness	$CaCO_3$	12	11	98	165	287	274	8	172	125
Non-Carbonate Hardness	$CaSO_4$	5	7	18	40	58	54	0	295	5,900

[a] All values are parts per million of the unit cited to nearest whole number (see Reference 5).

[b] Numbers indicate location or area as follows:
(1) Catskill supply—New York City
(2) Swamp Water (Colored) Black Creek, Middleburg, Florida
(3) Niagara River (Filtered) Niagara Falls, New York
(4) Missouri River (Untreated) Average
(5) Well Waters—Public Supply—Dayton, Ohio—30-60 ft
(6) Well Water—Maywood, Illinois—2090 ft
(7) Well Water—Smithfield, Va.—330 ft
(8) Well Water—Roswell, N. Mexico
(9) Ocean Water—Average

of dissolved gases such as carbon dioxide, oxygen, and sulfur dioxide, that tend to make them corrosive.

Dissolved gases are the main factors affecting condensate corrosiveness, but they may also be present in water containing mineral impurities to an extent that will make the water corroeive.

Mineralized Waters

All the waters found in streams, wells, lakes, and the ocean are mineralized. The same is true of all municipal supplies even though they may have been *treated*. For a given area, ground waters are likely to be more highly mineralized than are surface waters. Conversely, surface waters are more likely to be contaminated with municipal sewage and trade wastes. Virtually all mineralized waters also contain biological organisms.

Mineralogical Characteristics. The character and amount of extraneous inorganic materials—including deleterious gases—dissolved and suspended therein, describe the mineralogical characteristics of any water. Revealing such information is the function of a mineralogical chemical analysis.

The analyses in Table 1 disclose the composition of the public water supplies used by about 45 percent of the total population of the cities, in the United States, having more than 20,000 inhabitants.[5]

All values recorded are in terms of parts per million.*

* Parts per million are hereinafter abbreviated ppm. A part per million signifies a unit weight of material per million unit weights of the solution.

Table 2 .... Conversion Factors for Water Analyses

To Convert	Into	Multiply by
Grains per U. S. gallon	ppm	17
Grains per Imperial gallon	ppm	14¼
Grams per liter	ppm	1000
Mg per liter	ppm	1

This is the approved standard terminology for reporting the results of mineral analyses.[5] Values reported in the other terms commonly used may be converted into the standard form by using the factors listed in Table 2.

Data for dissolved gases or *pH* values have been omitted in Table 1 because even waters of the same mineral contents may vary widely in these respects. Unpolluted natural waters usually have *pH* values within the range 5 to 8, depending upon their *free* CO_2 contents. Polluted waters, which include those derived from wells or swamps in marshy ground, may have *pH* values well below 5.

Biological Characteristics. The slime-forming organisms are mostly lower plants, grouped by botanists into the phylum thallophyta. This group is distinguished by the absence of leaves, roots, or stems from the mosses, ferns, and seed plants which comprise the three other phyla of the plant kingdom.

The thallophyta (see Table 3) are divided into algae, which can synthesize chlorophyll for the production of sugar, and the fungi which lack chlorophyll, and must therefore secure already synthesized carbohydrates.

All thallophyta are of universal distribution and many of them are slime-forming. Of the five divisions of algae, only three (the green, the blue-green, and the diatoms) are found in fresh water. Of the five divisions of fungi, all may occur in fresh water, the principal slime formers being indicated in Table 3.

The methods of analysis commonly used in the sanitary examination of a water have, as their principal object, to identify and count pathogens.

Most slime-forming organisms are not pathogens. When a water is subjected to a biochemical analysis for the purpose of evaluating its slime producing characteristics, tests, widely different from sanitary bacteriological tests, must be made. Tests upon the water itself are seldom satisfactory, and true indications of the sliming characteristics of a water can

Table 3 .... Principal Slime Formers

Phyla	Rough Division of Phyla
Algae	Single celled, sometimes forming slimy sheets. Many celled in either sheets or froods.
Fungi	*Bacteria* (Schizomycetes) frequently forming slimy surface coatings. *Slime Molds* (Myxomycetes) forming slimy sheets as one stage of their life history. *Sac fungi* (Ascomycetes) of which one division, the yeasts, occasionally form slimy aggregates. The alga-like fungi (*Phycomycetes*) and the stalked fungi (Basidomycetes) rarely form slimes but their filaments may hold together the slimes of other organisms.

only be determined by the analysis of deposits on surfaces having a temperature that approximates the operating temperature range of the final design equipment. The results of such tests are commonly used for selecting proper preventive measures.

Slime and algae micro-organisms are capable of rapid multiplication and, if not controlled, can cause shut-down or permanent damage to equipment. They can be controlled by chemical treatment of the water or in certain cases by mechanical cleaning of the surfaces affected. The chemicals used are usually toxic, and should therefore be handled only by trained personnel having a full understanding of the quantity required and their effect on operators as well as surfaces.

Chemical treatment is usually preferred to mechanical cleaning particularly where surfaces are inaccessible for cleaning or where a shut-down of the operation is necessary for mechanical cleaning.

Condensates

All condensates are caused by the cooling of vapor. Condensates resulting from the cooling of water vapor in air are *dews, sweats, rain,* and *snow,* while those formed in steam-condensing apparatus are termed condensates or return water.

In the heating and ventilating field, the biological characteristics of condensates are likely to be of concern *only* where the condensate is used as cooling water in recirculating systems. The deleterious gas contents of condensates, however, very often create serious corrosion troubles.

The data in Table 4 typify the chemical composition of

Table 4 .... Data Typifying the Deleterious Gas Content of Different Atmospheres

Name of Gas	Chemical Formula	Air: Rural % by Volume	Air: Rural Partial pressure psia	Air: Metropolitan % by Volume	Air: Metropolitan Partial Pressure psia
Oxygen	O_2	21	3.143	21	3.143
Carbon Dioxide	CO_2	0.03	0.004	0.06	0.009
Sulfur Dioxide	SO_2	None	None	0.003	0.004

Name of Gas	Chemical Formula	Flue Gases: Bitum. Coal % by Volume	Bitum. Coal Partial pressure psia	Fuel Oils % by Volume	Fuel Oils Partial pressure psia	Natural Gas % by Volume	Natural Gas Partial pressure psia
Oxygen	O_2	2	0.299	7	1.048	10	1.497
Carbon Dioxide	CO_2	15	2.245	13	1.946	10	1.497
Sulfur Dioxide	SO_2	0.07	0.010	0.03	0.004	0.0001	0.0015

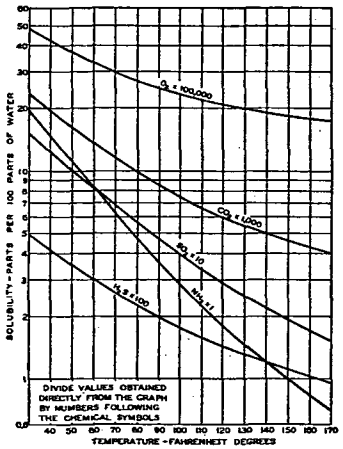


Fig. 1 .... Solubility of Gases at Partial Pressure of 1 Psi

the atmosphere in rural and metropolitan areas, and of stack gases when various types of common fuels are used. The curves in Fig. 1 disclose the solubility of the major deleterious gases present in such atmospheres in otherwise pure water, when the partial pressure of the gas is one psia.

The most common deleterious gases entrained by steam are oxygen and carbon dioxide. In rare instances, hydrogen sulfide, sulfur dioxide, or ammonia are present.

In most steam-condensing equipment,[7,8] the non-condensable gases entrained with steam accumulate so that the amount present in the vapor space is several hundred times higher than in the incoming steam and the amount dissolved in the condensate may therefore approach, or even exceed for short periods, the amount entrained by the steam.

CAUSES AND PREVENTION OF SCALES AND SLUDGES

Scales may be formed on surfaces of equipment in contact with water, and sludges in the body of the water, by the separation from the water of dissolved or suspended solids. According to the nature of a particular piece of equipment and the method of its operation, such separation may be promoted by one or more than one, of several factors:

a. The concentration of solids may be increased by the evaporation of water.

b. The dissolved solids may be rendered less soluble in the water by changes in temperature.

c. Conditions may favor the decomposition of unstable compounds with the formation of less soluble compounds.

Figs. 2 and 3 show that the solubilities of both calcium carbonate and calcium sulfate decrease with the rising temperature within a moderate range of temperatures. Surfaces transferring heat into water, such as condensers and

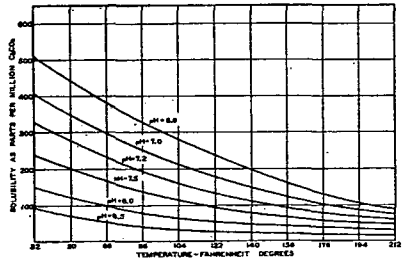


Fig. 2 Adapted from (1) *Ind. & Eng. Chem.*, 20 (1928) 1197—by Balis. (2) J.A.C.S. 60 (1929) 2086—Frear & Johnson.

Fig. 2 .... Solubility of Calcium Carbonate in Distilled Water Containing Carbon Dioxide

(*pH Values at Approximately 73 F*)

coolers, are more susceptible to scale and sludge formation than are the cold parts of the same system using the same water.

The most common of the unstable soluble salts are the bicarbonates of calcium, magnesium, and occasionally iron. Under conditions favoring the removal of carbon dioxide, as when the water is strongly aerated or when it is boiled, the bicarbonates are readily converted to the relatively insoluble carbonates. The type of scale most usually encountered is calcium carbonate (often called alkalinity) from the decomposition of calcium bicarbonate according to the following chemical reaction:

$$Ca(HCO_3)_2 + \text{heat} \rightarrow CaCO_3 + CO_2 + H_2O$$

$$\underset{\text{Bicarbonate}}{\text{Calcium}} + \text{heat} \rightarrow \underset{\text{Carbonate}}{\text{Calcium}} + \underset{\text{Dioxide}}{\text{Carbon}} + \text{water}$$

Conversely, carbonates are readily converted to the more soluble bicarbonate by the addition of carbon dioxide or other acidic materials. This explains the increase in the apparent solubility of calcium carbonate at decreasing *pH* values (increasing concentration of hydrogen ion) shown in Fig. 2, the carbonate really going into solution largely as bicarbonate.

In special cases, primarily with well waters possessing a high natural iron content, deposition of iron oxide from the decomposition of ferrous bicarbonate takes place according to the following reaction:

$$4\,Fe(HCO_3)_2 + O_2 \rightarrow 2\,Fe_2O_3 + 8\,CO_2 + 4\,H_2O$$

$$\underset{\text{Bicarbonate}}{\text{Ferrous}} + \text{Oxygen} \rightarrow \underset{\text{Oxide}}{\text{Iron}} + \underset{\text{Dioxide}}{\text{Carbon}} + \text{water}$$

Low water velocity in heat transfer equipment often causes sludges to settle out in the tubes. This has the same effect on heat transfer as scale formation. When this occurs, increasing the water velocity often cures the condition.

It is sometimes desired to evaluate the tendency in a particular water toward the separation of calcium carbonate, which may be desirable as a means of establishing a corrosion-resistant film on metal surfaces, or in other circumstances may be undesirable because of the impedance of

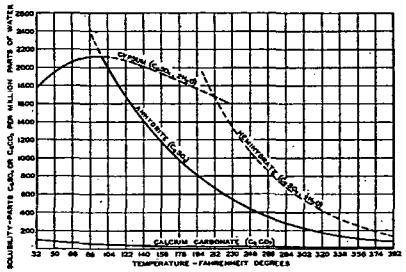


Fig. 3 Adapted from Bull. No. 15, Univ. of Mich. *Formation and Properties of Boiler Scales* by P. E. Partridge.

Fig. 3 .... Solubility of Calcium Sulfate and of Calcium Carbonate for Comparison

($CaCO_3$ *in Equilibrium with Normal* CO_2 *Content of the Atmosphere*)

the calcium carbonate film to heat transfer. This tendency is indicated approximately by the Langelier Index,* which is obtained by subtracting the actual *pH* of a particular water from the *pH* at which it is estimated precipitation of calcium carbonate would just begin. This estimate may be made by the use of Fig. 4.

There are various expedients which may be employed for avoiding or mitigating difficulties due to scales:

a. The water may be treated before use to remove elements such as calcium, magnesium, and iron, which form relatively insoluble compounds. In the various softening processes this removal of these elements is accompanied by the addition of other elements, particularly sodium, the compounds of which are *relatively* soluble.

b. The water may be treated within the equipment to promote the separation of dissolved solids as sludges, rather than as scale which is, in most cases, more objectionable.

c. The increase in total solids, due to the evaporation of water, may be controlled by the displacement, continuously or intermittently, of some of the used water by fresh supply.

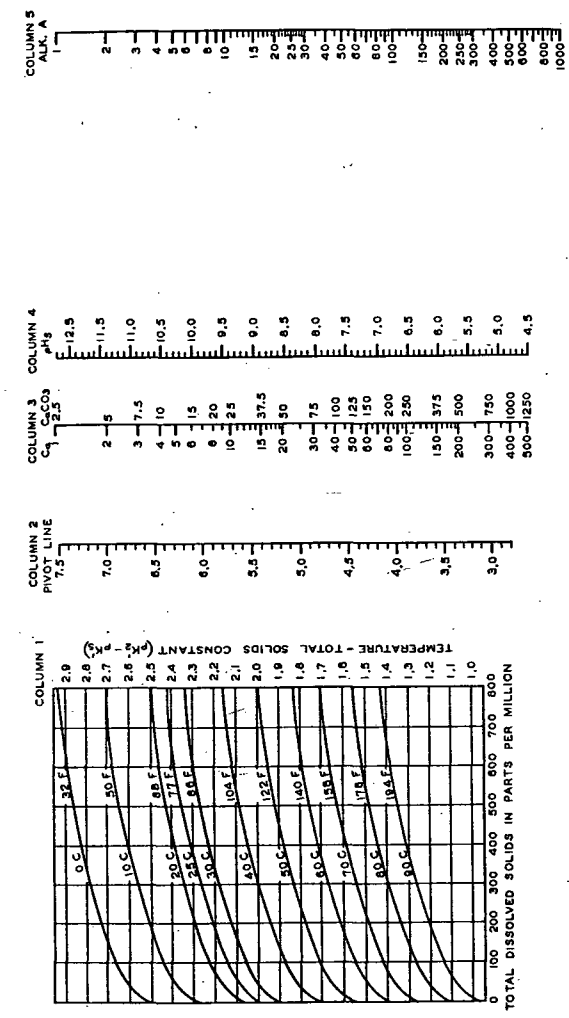
d. Substances, such as the polyphosphates, having the property of inhibiting the precipitation of calcium carbonate from solutions supersaturated with it, may be added.

e. The *pH* of the water may be lowered (hydrogen ion concentration raised) to reduce the tendency for precipitation of carbonate. This is permissible only to such an extent as will not cause a serious increase in rate of corrosion.

The choice of the best expedient or combination of expedients must be made for each type of equipment, and will be affected by operating and water considerations. Item c above must be used, regardless of which other items are used, for all recirculating water systems from which evaporation takes place.

Once-Through Equipment

Where abundant supplies of water are available at low cost, the cooling water may pass through the equipment once, undergoing a rise in temperature. Little difficulty from scale should be encountered if the Langelier Index is lower than +0.5, based on the highest water temperature encountered in the system. This often corresponds to a carbonate hardness of less than 200 ppm when water temperatures are lower than 100 F. However, since many municipal supplies are softened so that incipient calcium carbonate



Based on article in Oct. 1936 issue of *American Water Works Association Journal* and later corrections for Tables 2 and 4 of article. Prepared for Charles P. Hoover of the Columbus, Ohio, Water Softening and Purification Plant by M. L. Rishl.

Instructions for Using Chart:

1. Knowing temperature and total dissolved solids, find temperature and total solids constant on Col. 1.

2. Align this constant with given value of calcium on Col. 3 of Chart, then locate point in Col. 2 of Chart (Pivot Line).

3. Align this point on Pivot Line with given alkalinity on Col. 5; read *pH* saturation on Col. 4.

Saturation Index is *pH* actual minus *pH* saturation.

e.g.—*pH* actual, *pH* saturation, Saturation Index.

7.6	8.1	−0.5 (corrosive)
8.4	7.8	+0.6 (scale forming)

[Data required for determining *pH* saturation]

(1) Total alkalinity, as parts per million of $CaCO_3$.
(2) Calcium in parts per million.
(3) Total dissolved solids, in parts per million.
(4) Temperature, Centigrade, at which *pH* saturation desired.

Fig. 4 .... Graph and Nomogram for Determination of *pH* Saturation by Langelier's Formula

(*Applicable within pH Range 7.0–9.5*)

precipitation is induced, the Langelier Index is a more reliable indication of scaling tendency.

Prevention of scale in once-through systems is based primarily on the adjustment of pH, the application of surface-active agents, or a combination of the two. Normally the addition of a few parts per million of a surface-active agent such as a polyphosphate[20] fed proportionately to a water with a scaling tendency is all that is required to control scale.

Where a given water does not respond satisfactorily to surface-active agents, reduction of pH with sulfuric acid or carbon dioxide is often employed. The pH is reduced only sufficiently to have it fall within the effective range of the surface-active agent. The use of acid or carbon dioxide is sometimes permitted without polyphosphates when corrosion will not be severe. In this case the pH is reduced so that the Langelier Index is less than +0.5. The acid or carbon dioxide must be added with care. Use of automatic pH controllers is required if corrosion is to be prevented.

A special case of scale from ferrous bicarbonate decomposition was mentioned previously. Ferrous bicarbonate is considerably less stable than calcium bicarbonate. The prevention of this decomposition can be accomplished by the addition of polyphosphate in low concentrations.[21] The use of polyphosphate in the ratio of two parts by weight to one part by weight of iron is usually most effective. The polyphosphate must be added to the water before exposure of the water to air or chlorine if effective stabilization of the iron is to occur.

Closed Recirculating Systems

The closed recirculating system is one in which water circulates through one heat exchanger where it absorbs heat, has its temperature elevated, and circulates through another heat exchanger in which its temperature is lowered. In cold water or chilled water systems scaling is seldom a problem. In hot water systems scaling is seldom a problem unless there is a large amount of make-up water.[22] In this case surface-active agents such as polyphosphate are used.

Open Recirculating Systems

Where water from condensers and similar equipment is passed through a spray pond or cooling tower and then returned to the equipment, there is an increase in the concentration of solids because of the evaporation of some of the water into the cooling air, and, moreover, the aeration removes carbon dioxide. Both factors promote the tendency to deposit scale.

The corrosive properties of the water can also be increased through the absorption of acid gases such as sulfur dioxide (SO_2), from the air.[23] The absorption of gases from the air being processed and the concentration of solids due to evaporation are factors in air washer operation when the air is humidified.

The concentration of scale-forming minerals in open recirculated cooling water is limited by natural drift or windage loss. This is the loss of water droplets from the system. Windage losses from typical systems may be classified as follows, based on recirculating rates:

	Percent of Recirculating Rate
Evaporative Condensers and Air Washers	0 to 0.1
Mechanical Draft Cooling Towers	0.1 to 0.3
Atmospheric Cooling Towers	0.3 to 1.0
Spray Ponds	1.0 to 5.0

To show the effect of concentration in a typical mechanical draft system assume:

1. Recirculating rate = 100 gpm.
2. Drift loss = 0.2 percent of recirculating rate = 0.2 gpm.
3. Evaporation rate = 1 percent of recirculating rate = 1.0 gpm.
4. Calcium bicarbonate in make-up water = 100 ppm.
5. Calcium bicarbonate in evaporated water = 0 ppm.

Let K = concentration of calcium bicarbonate in recirculating water = concentration of calcium bicarbonate in drift loss.

The process of concentration in the system may be represented diagrammatically as follows:

Evap. = 1 gpm
$Ca(HCO_3)_2$ = 0 ppm
↑
Recirculating Water System
$K = Ca(HCO_3)_2$ conc.
→ Drift 0.2 gpm, $Ca(HCO_3)_2 = K$ ppm
↑
Make-up = 1.2 gpm
$Ca(HCO_3)_2$ = 100 ppm

The concentration may be obtained from the equation

$$\text{(Make-Up)[Ca(HCO}_3\text{)}_2\text{ conc.]} = \text{(Evap.)[Ca(HCO}_3\text{)}_2\text{ conc.]} + \text{(Drift)[Ca(HCO}_3\text{)}_2\text{ conc.]}$$

$$(1.2)\ (100) = (1)(0) + (0.2)(K)$$

substituting,

$$120 = 0 + 0.2\,K$$

whence,

$$K = 600 \text{ ppm}$$

Therefore the concentration of calcium bicarbonate in the recirculating water equals 600 ppm. This far exceeds the allowable concentration of 170 ppm at which scaling will occur.

To correct this situation, it is imperative to provide a continuous bleed-off or blowdown from the recirculating water circuit. Typical bleed-off requirements are shown in Fig. 5. Reference to the No Treatment curve shows that a

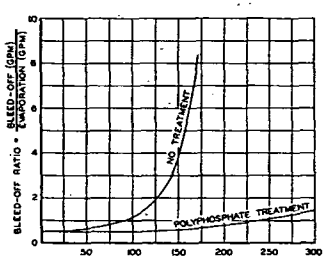


Fig. 5 .... Relation of Bleed-Off Requirement to $CaCO_3$ in Make-Up Water

bleed-off rate of 1.2 times the evaporation rate is required when a make-up water contains 100 ppm calcium bicarbonate (or alkalinity) if scale is to be prevented. Calculation by a material balance will show that the bleed-off plus drift loss will limit the calcium bicarbonate to 171 ppm.

It can be seen from Fig. 5 that very large amounts of bleed-off are required as a make-up water approaches or exceeds 150 ppm of alkalinity. When alkalinity exceeds 175 ppm, bleed-off alone is no longer effective. The use of 5 ppm or less of polyphosphate is extremely effective in reducing the required bleed-off and making possible the use of make-up waters which will not respond to bleed-off alone.

Occasionally, waters are encountered which have exceptionally high alkalinities or which develop high pH values when recirculated. The use of sulfuric acid is common in these cases to maintain pH values between 7 and 8. Polyphosphates are most always used when acid is required.

It is very important to control acid feeding carefully to avoid serious corrosion. Automatic pH controllers are often employed for such a purpose.

Heating Systems

In hot water heating systems or in steam heating boilers where all condensate is returned, troubles from scaling should not be severe. If necessary, sodium phosphate or sodium carbonate may be added to the water to prevent the formation of adherent calcium sulfate scale.

Boilers and High Temperature Equipment

Where temperature exceeds 250 F, complete softening of the water is the only practical method for minimizing sludge formation. This is usually accomplished by artificial or natural zeolites (called also ion-exchange materials) or by hot-process precipitation softeners.

In boilers operating at pressures above 100 psig virtually all the calcium, magnesium, silica, iron, and manganese salts entering with the feed water are potential scale or sludge formers.

In boilers operating at 100 to 250 psig, the formation of adherent calcium sulfate (anhydrite) scales is most to be feared. Such deposits form on the *hottest* evaporative surfaces. The scale has a low heat conductivity. Even a layer of egg shell thickness may so impede the rate of heat transfer as to bring about over-heating of the metal.

The orthophosphates of sodium are most frequently used to prevent sulfate scales. The concentration of phosphate required is such as to cause the precipitation of calcium phosphate as sludge, thus keeping the boiling water undersaturated with respect to calcium sulfate. To a lesser extent, sodium carbonate (called also soda ash and sal soda) is also used. Most of the effective boiler compounds contain either phosphates or soda ash, or both. Certain organic materials and colloids are sometimes found to minimize scale formation. Where chemicals are introduced directly into the boiler in amounts adequate to prevent scale, sludge is formed in amounts proportionate to the calcium and magnesium salts entering with the feed water. To prevent troublesome accumulation of this sludge, as well as soluble salts, as evaporation occurs, some *blowdown* of boiler water is necessary.

CAUSES AND PREVENTION OF SLIMES

A water containing slime-producing organisms will produce prohibitive amounts of slime *only* when the conditions of use are such as to propagate their life processes. Whenever sufficient food material from normal water or from airborne dust combines with optimum temperature conditions, such as exist on cooling surfaces and air washers, serious quantities of slime will be produced.

Some natural well waters do not contain sufficient foods to support luxuriant slime growths. Algae, which require light for carrying on their life processes, are likely to cause difficulty in cooling towers and other areas where sunlight is abundant. The ordinary slime-forming bacteria are capable of using a wide variety of nitrogenous and cellulose material as food sources. These bacteria thrive best under dark conditions such as exist in condensers and other heat transfer surfaces. Other organisms capable of causing similar difficulties use such a wide variety of food material as algae,[24] iron compounds,[25] and inorganic sulfates.[26]

At present, the use of toxic chemicals and irradiation are the two general means employed in slime control. The value of ultra-violet light, used so broadly in the beverage industry, is somewhat in dispute.

Anti-fouling paints have been developed and are fairly satisfactory for the prevention of the growth of macro-organisms such as *barnacles* and *mussels*, but these paints must be renewed at frequent intervals, and are not applicable to inaccessible areas such as the inside of pipe lines and cooling towers. Satisfactory *anti-sliming* paints have *not* been found.

Table 5 .... Common Chemicals Used for Slime Control

Chemical	Trade Name	Physical State*
Chlorine	Chlorine	Gas
Hypochlorites	Calcium Hypochlorites	Crystalline
	Sodium Hypochlorites	
Chlorinated Phenols	Chlorophenylphenate	Briquettes
Sodium—	Tetrachlorophenate	Briquettes
	Pentachlorophenate	Briquettes
Potassium Permanganate	Permanganate of Potash	Crystalline
Copper Sulfate	Blue Vitriol	Crystalline

* As Shipped.

Names and other pertinent data relating to some of the more common chemicals used in slime control are shown in Table 5.

Chlorine is the only chemical to which is attributed the ability to destroy slime-forming organisms. The others are presumed to poison marine organisms, most of which *recover* when the chemical is not used regularly.

While chlorine is the most generally used chemical, the use of others may occasionally prove to be more practicable. Choice of the chemical is conditioned largely by the design and operation of the system.

Open Recirculating Systems

In spray ponds and cooling towers of the open type, light-loving algae growths are likely to cause blocking of the distribution piping and troughs. These organisms are most troublesome in areas accessible to sunlight. Algae slimes are usually stringy in character.

In open recirculating systems, continuous use of small quantities of chlorine is generally most satisfactory. In once-

through systems, where large quantities of water are used, intermittent treatment a few times each day will usually result in satisfactory slime removal and chemical economies.

Neither the phenols nor copper sulfate may be used for the removal of slime already formed. For this purpose, chlorine gas or suitable chlorine liberating compounds may be used. After being cleaned, the other chemicals may be used to prevent the reestablishment of slime in the system. The removal of green algae from a cooling tower should never be used as an indication that the true slime-forming organisms on heat exchanger surfaces have been removed. The more resistant slime formers, which so materially reduce heat transfer efficiency, will often be unaffected by treatment which completely eliminates algae.

Copper sulfate must be used with care because it can cause serious corrosion of steel in a system. It is also ineffective in alkaline water because the copper is precipitated from the water.

Closed Once-Through Systems

In equipment where light is excluded, slime formations are due to fungi. Usually, they predominate on the heat exchange surfaces. Bacteria form thick, soft slime. Yeast and molds form tough rubbery slimes. Chlorine and hypochlorite solutions, fed intermittently, are usual preventatives.

UNDERWATER CORROSION

When deleterious substances are present in water, the corrosivity of the solution is increased in proportion to the amount of deleterious substances present, the temperature, and usually the rate of flow of the solution over the metal surfaces. There are other relevant factors, but their influence in general is subordinate to those mentioned. Dissolved oxygen, acid gases, and chloride salts are the corrosion accelerators most frequently encountered.

Neutral and slightly alkaline waters saturated with air, corrode iron at a rate about triple that for the same water free of air. Hot water containing oxygen will corrode iron at a rate three to four times that for the same water when cold.

Corrosion of iron decreases as the *pH* of water solutions increases, and practically ceases at a *pH* of 11. Film-forming agents such as chromium, nickel, and silicon, can be added in the manufacturing of metals to provide increased corrosion resistance. In some processes, inhibitors, such as chromates, can be added to the water to minimize corrosion.

Soft water, as for example the effluent from zeolite softeners, is likely to be several times more corrosive to iron than hard waters. In small installations, the use of copper or brass pipe usually is a practical expedient. Cement-lined pipe and tanks suitably resist attack.

Where the water contains slime-forming organisms, especially those bacteria that thrive on iron, chlorination of the water is imperative to inhibit tuberculation and subsequent pitting.

Bituminastic paints, applied at *regular* intervals upon *well cleaned* surfaces, will measurably prolong the life of equipment handling cold waters.

It is generally agreed that the rate at which oxygen reaches the surface determines the rate of corrosion of ferrous metals. Underwater, the oxygen diffusion to a surface is often restricted by films and scale that form on the surface, and consequently the rate of corrosion differs from that of surfaces exposed to the atmosphere or to alternately wet and dry conditions.

All ferrous metals will not corrode at the same rate. A thorough knowledge of the conditions to be encountered and experience gained from tests of the materials in service are required when attempting to predict performance or life in a given service.

Copper, aluminum, and other non-ferrous metals and their alloys have been found effective in resisting many types of underwater corrosion. Corrosion of ferrous metals with which these metals are in contact, may be accelerated in water of good conductivity.

The environmental conditions are responsible for length of service of any metal. No single material is suitable for all types of service.

Once-Through Systems

Where corrosion can be expected in once-through systems, it may be minimized through the use of one of the following methods:[32]

1. Forming a protective film of calcium carbonate on the metal surfaces.
2. Providing mechanical or chemical deaeration, or both, of the water.
3. Using organic or inorganic corrosion inhibitors, or both, in low concentrations.

Formation of a thin protective calcium carbonate film is accomplished by adjustment of the water until the Langelier Index has a value greater than +0.5. This is rather difficult in systems which have a wide variation in temperature. Since the Langelier Index is dependent on temperature, when a water is adjusted to form a protective film on those surfaces of higher temperature, no film will form on colder surfaces. Therefore, this method of corrosion control is more applicable to large municipal distribution systems. Where heat transfer is involved, the method is usually impractical except under unusual circumstances. When this method is used, *pH* is increased by using an inexpensive alkali such as lime, caustic soda, or soda ash. Lime is usually used for waters of low calcium content. Caustic soda or soda ash are used for waters of high calcium content.

Mechanical[33] and chemical deaeration are not often used in once-through systems because of relatively high operating costs. Mechanical deaeration also requires the use of costly equipment. Chemical deaeration is usually accomplished by raising *pH* with caustic soda and by continuously feeding catalyzed sodium sulfite in proportion to water flow. The use of catalyzed sulfite is not permitted in water to be used for potable purposes.

There are several corrosion inhibitors such as chromate, polyphosphates, and silicates which are effective in once-through systems.[34] Polyphosphate at a concentration of 2 to 5 ppm has been found useful in controlling tuberculation of iron pipe and in reducing overall corrosion. Usually, more effective treatment is to use chromate polyphosphate at concentrations of less than 60 ppm.

Sodium silicate is often used in relatively soft waters by increasing the silica content about 8 ppm. It is used primarily to reduce corrosion in potable water supply systems.

Open Recirculating Systems

Corrosion in open recirculating systems, such as air washers and cooling towers, is usually controlled by use of corrosion inhibitors such as chromates, polyphosphates, a combination of chromate and polyphosphate, and nitrites.[35, 36] Mechanical or chemical deaeration is not practicable in open systems because of the high rate of aeration. It is not practicable to use high *pH* values of 11 or more in systems containing wood. High *pH* will cause serious delignification of wood. High *pH* values also prevent control of scale. Corrosion control is usually carried out in the *pH* range of 6.5–8.

Chromates are by far the most effective corrosion inhibitors. It is extremely important, however, to maintain an adequate concentration of 300–500 ppm, which is effective for most systems. If for economic reasons substantially lower concentrations are used, serious pitting corrosion may occur.

Polyphosphates are most effective in reducing tuberculation. It is not usually possible to reduce overall corrosion to anywhere near the degree possible with chromate.

Where economy of treatment is of primary importance, a substantial reduction of pitting and overall corrosion can be obtained by using as little as 60 ppm of a mixture of polyphosphate and chromate. Close control over *pH* is a requirement for good corrosion control by this process.

Sodium nitrite has not had widespread use as a corrosion inhibitor in open recirculating systems. It has been reported that difficulty may be encountered in maintaining effective concentrations. Considerable field experience is needed to further qualify this inhibitor.

Closed Recirculating Systems

The term Closed Recirculating System is in reality a misnomer. Except for relatively small systems, most closed systems are open because they most usually require make-up water. Recently, tests conducted on 84 closed systems indicated that more than 50 percent of the systems had one or more water changes per month because of leakage. Weekly water changes were found in more than 10 percent of the systems.[37] *Continuous make-up, of course, replenishes oxygen* in the system, thus promoting corrosion. It is imperative, therefore, that corrosion control be provided for most closed systems. The age old assumption that closed systems are closed is no longer valid.

Corrosion control is usually accomplished by (1) mechanical or chemical deaeration or (2) use of corrosion inhibitors such as chromates and nitrites.

The use of polyphosphates is not generally recommended for closed systems because they will revert to ineffective orthophosphates unless there is a large replacement of water containing polyphosphate.

Higher concentrations of chromate are usually maintained in closed systems as compared with open systems. This is primarily due to the fact that, since water losses are usually small, the cost of maintaining excess chromate as a safety factor is small.

Treating Chemicals

Whenever chemicals are used to control scale, corrosion, algae, and slime, competent advice from a water chemist is desirable. Often factors considered irrelevant to the layman, will be of much importance in securing effective treatment. Very often troubles are created through improper use of chemicals and are more serious than if they were not used.[38]

Refrigerating Systems

Corrosion in refrigerating systems is confined to surfaces in contact with brines or those in contact with refrigerant.

Brines. Refrigerating brines usually are comprised of sodium chloride, calcium chloride, or calcium and magnesium chlorides. The corrosivity of dilute brines is higher than their more concentrated solutions. The corrosivity of sodium brines, other conditions being fixed, is about 1.5 times greater than brines of the alkaline earth metals.

Brines are excellent electrolytes. Contact of dissimilar metals of wide potential differences, when in contact with brines, results in rapid corrosion by galvanic action.

The leakage of air, acid refrigerants, or both, accelerates the corrosivity of brines. Ammonia precipitates calcium and magnesium salts, thus clogging the system at restricted points.

The addition of caustic soda and sodium dichromate to brine solutions to inhibit corrosion of iron, is a more or less general practice. Sodium silicate and sodium phosphate are also used at times, but tests indicate they are not as effective as is sodium dichromate. It has been suggested[39] that 125 lb of sodium bichromate per 1000 cubic feet of calcium chloride brine, and 200 lb per 1000 cubic feet of sodium chloride brine, be added to inhibit brines; that when salt or calcium chloride is added to "strengthen" brine, sodium dichromate also be added in the amounts shown in Table 6.

Table 6 . . . . Quantities of Sodium Dichromate to be Added to Maintain Initial Concentration

Specific Gravity of Brine to be Strengthened	Lb Sodium Dichromate per 100 Lb $CaCl_2$ Added
1.16	0.695
1.18	0.621
1.20	0.556
1.22	0.502
1.24	0.455
	Lb of Sodium Dichromate per 100 Lb NaCl Added
1.12	1.79
1.14	1.47
1.16	1.32
1.175	1.18

Refrigerants.[40] The common refrigerants, except those of the hydrocarbon type, will attack the common metals and alloys if *moisture is present*. Even a very small amount of water may cause severe corrosion with certain refrigerants. The amount required need only be sufficient to produce a water film on the metal surface.

With the halogenated hydrocarbons, complete elimination of water is much to be desired. Where ammonia is used, copper and its alloys, aluminum and zinc, are attacked especially at elevated temperatures. When sulfur dioxide is used, more than 50 ppm (0.005 percent) of water will cause appreciable corrosion of virtually all the common materials.

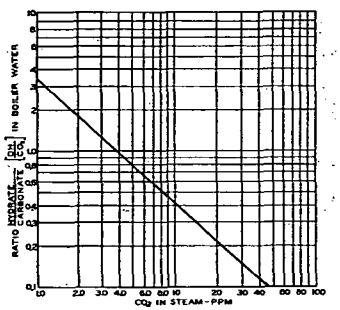
Minimizing Steam Condensate Corrosion

There are four expedients that may be utilized to minimize corrosion in steam condensate systems: (1) treatment of the boiler feed water so as to eliminate deleterious gases entrained with the steam, (2) design of the condensing equip-

ment to minimize dissolution in the condensate of the deleterious gases entrained with the steam, (3) chemical treatment of the condensate, (4) use of resistant metals.

Boiler Feedwater Treatment. Elimination of oxygen from boiler feedwater and, therefore, from the steam developed, can be accomplished either mechanically or chemically. In some steam generating stations, both expedients are employed.

Tests[28] have indicated that in small low-pressure heating boilers, where the boiler input contains less than about 50 ppm of carbonate hardness, the CO_2 in the steam can be controlled by adding calcium hydroxide to the boiler. In Fig. 6 are shown the equilibria conditions proposed for boilers operating at pressures up to about 5 psi gage. This



(All analytical values are ppm by weight)

Fig. 6. . . . . Relation of Hydrate/Carbonate Content in Hard Boiler Water and CO_2 in Steam at About 5 Psi Operating Pressure

expedient may *not* be used in higher pressure boilers, because of the possibilities of scale and sludge formations. In the latter, the only method used to date for treating the feedwater consists of removing the alkaline earth salts, i.e., softening, and subsequent acidulation followed by deaeration at temperatures near the atmospheric boiling point of water.[29]

Design of Condensing Equipment. In the design of water heaters and comparable types of condensing equipment,[27] it is possible to shift the accumulation of non-condensable gases to a location away from the condensate level and, subsequently, vent these gases to the atmosphere. The venting of an amount of steam equal to about one-half percent of the total steam entering the condenser is the optimum vent rate.

Venting is of little practical value when the CO_2 content of the incoming steam is below about 5 ppm. When the steam contains more than 5 ppm, venting provides a means of producing a condensate containing a minimum of about 3 ppm. However, even as little as 3 ppm of dissolved CO_2 can produce active corrosion if large amounts of condensate are flowing.

Chemical Treatment of Condensate. Condensates containing comparatively large amounts of oil, are practically non-corrosive, due to the protective film provided by the oil. When oil is intentionally added to condensate,[28] inadequate quantities may accelerate rather than decelerate corrosion on those surfaces not covered by the oil. Sodium silicate added to CO_2-bearing condensate has been shown to decrease, but not entirely prevent, corrosive action. It is not known whether the protection afforded by silicate solutions is due to the establishment of a protective film on the metal surface or to neutralization of the CO_2 by the alkali in the silicate solution.

It has been postulated that ammonia,[30] cyclohexylamine,[30] ethylene diamine, and morpholine[31] will retard corrosion of condensate lines. Tests with benzylamine have also been reported.[32] Where copper and its alloys are involved, the use of alkaline inhibitors is believed inadvisable. The use of small amounts of sodium hexametaphosphate has been suggested too, but tests[28] indicate that this salt accelerates rather than decelerates, the rate of attack on steel by condensate containing CO_2 and oxygen. Whether chemical treatment of steam or condensate is feasible, must be determined not only upon the basis of the acuteness of corrosion troubles, but also upon the uses made of the steam or condensate.

Use of Resistant Metals

Steam condensates vary greatly in their ability to corrode metals. The amount of gases that have been dissolved are the determining factors, with oxygen and carbon dioxide, which turns to carbonic acid, being the most common.

Steel, wrought iron, and copper are the metals ordinarily used for pipelines carrying steam condensate. All of these materials have been used with various degrees of success, but under severe corrosive conditions only a few years of service can be anticipated. With the proper metal or alloy carefully selected to suit the existing corrosive conditions, a service life of 15 to 20 years or more is not at all unusual. To obtain a long service life, an analysis must be made of the existing conditions, and if the condensate contains undue amounts of oxygen, carbon dioxide, and other harmful impurities, mechanical elimination of these gases should be a requirement. Treatment of the boiler water and the condensate itself has also proved to be somewhat effective.

Copper and ferrous metals are attacked by combinations of carbon dioxide and oxygen. When the condensate is being returned to the boiler, any copper corrosion products in the water occasionally present a serious problem. The copper that goes into solution is carried along in the condensate, and upon reaching ferrous materials in the boiler and elsewhere will plate out on the ferrous surfaces and set up galvanic corrosion due to the contact of dissimilar metals.

One of the most successful means of preventing corrosion near the junction point of copper and ferrous piping, or where copper piping is connected to steel tanks or boilers and other ferrous vessels, is to install a *throw-away* section at or near the junction. Since the corroding effect will always be found on the ferrous side of the junction rather than on the copper or copper-alloy side, and the effect is usually localized in the immediate vicinity of the joint, the installation of a ferrous pipe section 8 to 10 in. long will absorb practically all of the corrosion. When this *throw-away* section becomes corroded, it can readily be replaced. It should be noted, however, that in ordinary hot water heating systems this problem of potential corrosion at junctions of dissimilar metals seldom presents itself. Where there is any suspicion in this regard, precaution can be readily and commonly taken by separating the two metals electrically by means of a simple insulating union.

No paint or similar protective coating has thus far proven satisfactory where dissimilar metals are used in the same condensate pipe runs. Tests of cement-lined and vitreous-lined pipe have shown the linings to be readily dissolved by the hot condensates. At condensate temperatures, galvanizing on iron or steel pipe has not proved to have any advantages, and in some cases has been found to be detrimental.

ATMOSPHERIC CORROSION

Most of the problems originated by atmospheric corrosion occur in connection with the fire-side of boilers and furnaces (including their flues and stacks), sewer vents, air ducts, and coal and ash handling equipment. Usually such equipment is fabricated from common types of ferrous metals.

Generally little or no atmospheric corrosion occurs at temperatures higher than the boiling point of water, because at such temperatures little or no condensate is formed. If it does form at the higher temperatures; only negligible amounts of carbon dioxide and oxygen present in the atmosphere, will dissolve in the hot liquid, but sulfur gases may dissolve and cause rapid attack. Oxygen, sulfur dioxide, sulfur trioxide, and carbon dioxide are the deleterious gases most frequently accountable for corrosion in moist atmospheres.

Coal Storage and Handling Equipment

Virtually all coals contain sulfur in the form of pyrite, and some moisture. In storage, the pyrite is likely to be decomposed by oxidation. Moisture dissolves the products of decomposition forming sulfurous and sulfuric acid. The acid solutions vigorously attack the supporting metal.

Rubber linings have been developed for coal chutes and bins to effectively resist corrosion and the abrasive action of the coal, but they are expensive.[34] Concrete linings for steel bunkers have also been effectively employed.[35]

The use of high chromium steels is not always a sure cure, especially with coals treated with dust-allaying agents high in chlorides.

Flues, Stacks, and Fire-side of Boilers

The surfaces of flues and boilers contacting the products of combustion, seldom experience corrosive attack when the equipment is in operation. Breechings, smoke hoods, and canopies in contact with flue gas may, however, be subject to attack during the warming-up period of an appliance, or when the rate of operation is so low that the temperature of the flue gas is below the dew point. It is common practice to use cast-iron or acid-resistant vitreous enameled steel in flue gas connections to appliances, to prolong the life of these parts. The shut-down period, when condensation of moisture occurs on the metal surfaces, is usually the time when most damage is done.[36] In those sections of the stacks where flue gas temperature drops below the dew point, corrosion is inevitable during operation.

It is clear that where long shut-down periods are anticipated, a practical method for mitigating corrosion is to clean the surface thoroughly and to provide adequate clean, dry air circulation to prevent condensation. (See also Care of Idle Heating Boilers, Chapter 35.)

Protective coatings with organic binders are destroyed rather rapidly above 400 F because of the decomposition of the organic materials. The surfaces of metals, whose temperature does not exceed 400 F, may be protected by periodically applying paints such as those specified in the following paragraphs entitled Air Ducts.

Air Ducts

The most practical method for protecting air duct surfaces made of steel from atmospheric corrosion, is to apply protective paints. One of the most effective protective coatings is red lead paint.

Three coats of paint should be applied, of which the first two coats should be rust-inhibitive paint such as red lead paint, with the second coat tinted to a light brown color with carbon black, and the finishing coat may be red lead paint tinted to a black or brown color, black paint made according to *Federal Specification* TT-P-61, red iron oxide paint conforming to *Federal Specification* TT-P-31, or white or light tinted paint made according to *Federal Specification* TT-P-40.

Another paint which has had some use for priming iron and steel is zinc chromate paint.

Under some conditions, a chlorinated rubber base paint made according to *Federal Specification* TT-P-91 may be used for the finishing coat, particularly where the presence of highly corrosive gases or contact with strong alkaline water would injure the standard paints. Rubber base paints should be used only for the finishing coat over regular priming and second coats.

BURIED PIPE LINES

Lines that are cold and in intimate contact with the earth are corroded from the same causes as in mineral waters, but pitting is usually more intense due to variations in concentration of salts and oxygen in solution, acidity, drainage, and presence of solid materials (such as cinder) in contact with metal pipe. Galvanic currents, induced by contact of certain dissolved constituents in the soil, often act over a large area, and accelerate corrosion where they leave the pipe line.

Certain bacteria that thrive in the absence of oxygen have the power to obtain hydrogen and dissociate sulfates in the soil, with a resultant production of hydrogen sulfide which attacks iron to form iron sulfide.

Stray electric currents from electric power generating stations sometimes find their way into buried steel structures, and do damage in proportion to the current density where the current leaves the metal to enter the ground.

Pipe Materials

Some underground corrosive environments found in the air-conditioning and heating industries require special materials. The selecton of such materials must be based upon an economic evaluation, as the use of expensive first cost materials is not wise if the life of lower cost materials is adequate. On the other hand, a material low in cost and corrosion resistance should be avoided if it leads to costly shut-downs, repairs, and early replacements. Underground piping materials should be selected for their ability to resist exterior as well as interior corrosion, and careful evaluation of the soil and water should be made.

Underground corrosion of metals is particularly difficult to predict. There are many different types of soils varying in composition and in ability to corrode both ferrous and non-ferrous metals. Where excessively corrosive soils are encountered, special materials may be necessary, but gener-

ally the commonly used ferrous materials have proved to be the most economical. Copper and many of its various alloys are also used advantageously, but their use is more restricted to selected localities. It has been found that soils with high content of organic matter or alkaline soils in which the ratio of chlorides and carbonates to sulphates is high, may be corrosive to copper.[21] Copper or the commonly used ferrous metals should not be embedded directly in cinders or in tidal marshes, as they may be unduly attacked by sulphur compounds. It is also wise to avoid the embedding of any pipe materials *in soils where unusually high salt contents* are known to exist.

Galvanized wrought-iron pipe and steel will resist underground corrosion quite satisfactorily, particularly if the galvanized coating is 3 oz per sq ft or more. The National Bureau of Standards reports that where an underground piping material contains 3 oz of galvanizing per square foot, or more, the life of the pipe materially increased. The zinc used for galvanized coating is on the electrochemical protective side of the iron; and the zinc as it corrodes is changed to zinc compounds before the underlying base metal is attacked. This fact and the mechanical protection provided by the zinc coating account for the protection afforded by the galvanized coating. Once the galvanized coating has been destroyed, the base metal must then retard the corrosive attack, so the rate of attack depends upon the composition of such base metals. Lead-coated pipe[22] has had a limited application for underground service. It has been found that it corrodes chiefly in soils deficient in oxygen or containing cinders.

Protective Coating

Protective coatings for buried pipe lines are in a class by themselves because of the unusual service conditions, and because it is not possible to maintain them by recoating when necessary. Buried steel pipe lines have been protected against corrosion with considerable success by the use of very thick bituminous coatings applied in molten condition. The best results are obtained by applying the bituminous coatings over a standard priming coat such as red lead or a bituminous paint, and for long service it has been found that after the bituminous coatings are applied, a wrapping of asbestos fabric saturated with bitumens will prevent movement and displacement of the bituminous coatings, and add greatly to the length of time satisfactory protection will be maintained.

Cathodic Protection

Protection is obtained by rendering the structure cathodic to the surrounding water or soil by means of a controlled difference of potential. This method, which has proved satisfactory and economical on a number of gas and oil pipe lines underground, has also been applied with some success to the protection of the inside of water-storage tanks and other structures that are in contact continuously with water. Protective coatings that insulate a large portion of the metal surface will reduce very materially the total amount of protective current that must be impressed on bare anodic areas to arrest corrosion.

Because of differences in environmental conditions, it is necessary to determine or estimate the minimum current density required for each structure, and design the anode or anodes so that the necessary protection can be obtained most economically. In water having relatively high electrical conductivity such as in sea water, this is comparatively easy compared with fresh water. In the latter, the composition of the water is a major factor. It is therefore desirable to obtain an accurate estimate of the minimum current density required. The current is then controlled by the potential between the anode and the structure to be protected.

Rectifiers have generally proved to be the most practical means for supplying the necessary current for protection of surfaces in contact with neutral waters.[23]

HANDLING WATER TREATING CHEMICALS

Virtually all the chemicals used in water conditioning are injurious if taken internally in large doses. Many also cause severe skin irritation. Thus, they should be handled with caution.

Caustic soda, lime, and concentrated sulfuric acid will burn the flesh. In addition, if mixed with small amounts of water, sufficient heat may be generated so that spattering occurs or the container becomes too hot to handle.

The chlorophenol compounds, even in the low concentrations used in water conditioning, have been reported[24] to produce dermatitis. Chrom-itch is not uncommon among workers handling chromates. The amines are said to be absorbed through the skin.[25] Morpholine is said to cause kidney and lung trouble when so absorbed.

Chlorine gas irritates the skin, eyes, and mucous membranes. Concentrations as low as 0.004 percent by volume in air cause dangerous illness in one-half to one hour.

When relatively large amounts of the non-gaseous chemicals are to be handled, protective clothing, including goggles, should always be provided, and a shower head or its equivalent provided at or very near the point where the chemicals are mixed. *Chemicals should always be washed from the skin with large volumes of water.*

For the handling of chlorine and chlorinators, the U. S. Public Health Service[26] stipulates the following safety requirements:

1. Suitable gas masks and a small bottle of ammonia for testing for leaks should be kept at convenient points immediately outside the room or enclosure in which chlorine is being stored or is in use. Gas masks should be inspected at regular intervals and kept in serviceable condition. Note:—All-purpose masks offer adequate protection only when the concentration of acid gases does not exceed two percent—see *Safe Practices Pamphlet #64* National Safety Council.
2. Chlorinating equipment and cylinders of chlorine should be housed preferably in separate buildings above the ground level.
3. The room or building housing chlorinators in service should be maintained at a temperature above 60 F, but never in excess of the normal summer temperature. The cylinders of chlorine should be shielded, where necessary, from excessive heat or cold. Direct heat should not be applied to cylinders of chlorine, nor should hot water be poured over them or come in contact with the cylinder valve.
4. Adequate ventilation should be provided for all enclosures in which chlorine is being fed or stored.
5. All joints of tubing connecting chlorine cylinder and chlorinators should be kept absolutely tight and inspected frequently to insure tightness. Tubing should slope upward from the cylinder.

LEGAL REGULATIONS

In a number of states, the water used for humidification, even in industrial plants, is required to meet drinking water standards insofar as bacteriological quality is concerned. A ruling of the U. S. Department of Agriculture, Meat Inspection Division, prohibits the use of chromate in water used for air washing when the air later contacts foodstuffs.[27]

There is an ever growing consciousness on the part of public health officials, of the necessity for regulations to *protect* potable water supplies. Attesting this is an ordinance[28] now in effect in Detroit, Michigan, which stipulates in part:

"No physical connection shall be maintained between lines carrying city water and pipes, pumps, or tanks supplied from any other source. Where dual supplies are necessary or desired, lines carrying city water must be protected against back flow of polluted water by an *atmospheric gap*. Secondary supplies and emergency sources shall include: surface waters from rivers, lakes, ponds, lagoons, and reservoirs; well waters both deep and shallow; any supply of water which has been stored, held, or reserved after being used for industrial purposes; cooling water, or water which has in any way been treated, processed, or has been subjected or exposed to any contamination of a bacteriological or chemical nature; and water from any other source than the city supply."

The U. S. Public Health Service stipulates:

"Salts of barium, hexavalent chromium, heavy-metal glucosides, or other substances with deleterious physiological effects, shall not be allowed in the water supply system."

Table 7 .... Recommended Maximum Allowable Content in Water Supply*

Substance	Max. Concentration, ppm
Copper	3.0
Iron & Manganese (Total)	0.3
Magnesium	125.0
Zinc	15.0
Lead	0.1
Fluorine	1.0
Arsenic	0.05
Selenium	0.05
Phenols (Total)	0.001
Polyphosphate of Sodium	10.0
pH Value @ 25C	10.6

* U. S. Public Health Service.

The same agency recommends that the concentration of the substances listed be held below the values cited in Table 7. The Board of Directors of the American Water Works Association has accepted these values as standard for *all* public water supplies in the United States.[29] While their action is not binding, prudence dictates that no form of treatment should be used that will result in raising the concentration of the substances listed above the value cited.

Since virtually all of the permissible chemicals used for scale, slime, and corrosion control have deleterious physiological effects if taken internally in *relatively large doses*, they should always be carefully *proportioned*. To insure this, the Detroit ordinance stipulates that the chemical feeding device must have the following major characteristics:

"1. There shall be a visible means of checking the quantity of material being applied by the feeding device.
2. A water metering device, sealed to prevent tampering, shall be installed to measure the flow of water being treated.
3. The device shall be constructed so that in the event of back-flow or vacuums, the maximum amount of material that may be possibly back-siphoned from the device or any of its attachments or parts shall not exceed one fluid ounce.
4. Should there be a failure of the water metering device or the water supply, the feeding device shall automatically cease operating."

REFERENCES

[1] Committee D-19: *Annual Report* (American Society for Testing Materials, 1947).
[2] H. H. Uhlig (ed.): *Corrosion Handbook* (John Wiley & Sons, New York, p. 27).
[3] Discussion (*Corrosion and Material Protection*, May 1945, p. 2).
[4] W. C. Schroeder: Round-table discussion on organizing the classification of industrial waters (*American Society for Testing Materials Proceedings*, Vol. 44, 1944, p. 1057).
[5] W. D. Collins: Typical water analyses for classification with reference to industrial use (*American Society for Testing Materials Proceedings*, Vol. 44, 1944, p. 1057).
[6] *ASTM Standard D-596-41* (American Society for Testing Materials).
[7] E. W. Guernsey: Discussion on preventing solution of CO_2 in condensates (ASHVE Transactions, Vol. 51, 1945, p. 69).
[8] L. F. Collins: Studies of the mechanism of solution of CO_2 in condensates formed in steam heating systems of buildings (ASHVE Transactions, Vol. 51, 1945, p. 39).
[9] W. F. Langelier: The analytical control of anti-corrosion water treatment (*American Water Works Association Journal*, Vol. 28, 1936, p. 1500).
[10] G. B. Hatch and Owen Rice: Surface-active properties of hexametaphosphate (*Industrial and Engineering Chemistry* Vol. 31, 1939, p. 51).
[11] G. B. Hatch and Owen Rice: Threshold treatment of water systems (*Industrial and Engineering Chemistry*, Vol. 37, 1954, p. 710).
[12] *Betz Handbook of Industrial Water Conditioning* (W. H. & L. D. Betz Co., 4th ed.).
[13] S. Sussman: Cooling water problems in the New York metropolitan area (*Industrial and Engineering Chemistry*, August 1952, p. 1740).
[14] J. A. Holmes: Slime control in cooling equipment with phenol derivatives (*Annual Water Conference Proceedings*, Engineers' Society of Western Pennsylvania, 1944, p. 61).
[15] H. G. Reddick and S. E. Linderman: Tuberculation of mains as affected by bacteria (*New England Water Works Association Journal*, Vol. 46, No. 42).
[16] R. F. Hadley: Microbiological anaerobic corrosion of steel pipe lines (*The Oil and Gas Journal*, September 1939).
[17] S. T. Powell: Cold water vacuum deaeration (*Water Conference Proceedings*, Engineers' Society of Western Pennsylvania, 1945, p. 51).
[18] G. B. Hatch and Owen Rice: Corrosion control with threshold treatment (*Industrial and Engineering Chemistry*, Vol. 32, 1940, p. 1572).
[19] J. H. Wilson and E. C. Groesback: Tests of corrosion inhibitors for water treatment in air conditioning equipment (*National Bureau of Standards Journal of Research*, Vol. 24, 1940, p. 665).
[20] C. M. Sterne: The control of corrosion in air-conditioning equipment by chemical methods (*American Society for Testing Materials Proceedings*, Vol. 38, 1935, Part 2, p. 261).
[21] Sidney Sussman and J. B. Fullman: Corrosion in closed recirculating water systems (*Heating and Ventilating*, October 1953, p. 77).
[22] F. N. Speller: *Corrosion—Causes and Prevention* (McGraw-Hill Book Co., New York, 1951, pp. 805, 905, 910, 912).
[23] L. F. Collins: Engineering problems of water treatment (*Power Plant Engineering*, July 1946, pp. 78, 120).
[24] *Corrosion and its Prevention* (Air-Conditioning and Refrigeration Institute, Washington, D. C., 1958).
[25] See p. 470 of Reference 19.
[26] Studies in The Detroit Edison Co. (Unpublished).
[27] D. S. McKinney, J. J. McGovern, C. W. Young, and L. F. Collins: Preventing the solution of CO_2 in condensates by venting of the vapor space of steam heating equipment (ASHVE Transactions, Vol. 51, 1945, p. 53).
[28] L. F. Collins and E. L. Henderson: Corrosion in steam heating systems (*Heating, Piping and Air Conditioning*, October 1939, p. 620).

[39] U. S. Patent 1,395,730.
[40] U. S. Patent 2,053,024.
[41] U. S. Patent 1,903,287.
[42] A. A. Berk: Treating steam chemically to reduce return line corrosion (*Industry and Power*, November 1947, p. 79).
[43] Tests in The Detroit Edison Co. (Unpublished).
[44] J. J. McNeil: Rubber linings and coatings (*Corrosion and Material Protection*, March-April 1947).
[45] J. V. Schaefer: Protection of steel bins from corrosion (*Power Plant Engineering*, Vol. 26, 1922, p. 632).
[46] E. R. Walters: *Some Notes on Corrosion of Cast-Iron Sectional Boilers* (The Institution of Heating and Ventilating Engineers, preprint, 1944).
[47] See p. 67 of Reference 2.
[48] K. H. Logan and M. Romanoff: Soil corrosion studies, 1941 (*National Bureau of Standards Journal of Research*, Vol. 33, 1944, p. 145).
[49] L. P. Sudrabin: Cathodic protection of steel equipment submerged in water (*Water Conference Proceedings*, Engineers' Society of Western Pennsylvania, 1944).
[50] D. W. Haering: A discussion. (See p. 66 of Reference 14.)
[51] T. S. Carswell and H. L. Morrill: Cyclohexylamine and dicyclohexylamine (*Industrial and Engineering Chemistry*, Vol. 29, 1937, p. 1247).
[52] *Drinking Water Standards*, etc. (*Public Health Reports*, Vol. 58, No. 3, Reprint No. 2440, January 15, 1943).
[53] R. M. Palmer: Discussion (*Heating and Ventilating*, October 1953).
[54] *Official Plumbing Code of the City of Detroit*, Article V.
[55] Private Communication from H. S. Jordan, AWWA.

CHAPTER 56

WATER SERVICES

Sizing Cold Water Supply Piping, Procedure for Sizing Cold Water Systems, Cooling Water Piping, Estimating Heating Load and Storage Capacity, Methods of Heating Water, Direct-Fired, Electric, and Indirect Water Heaters, Computing Heat Transfer Surface, Hot Water Supply Piping, Control of Service Water Temperature, Safety Devices, Solar Water Heaters, Domestic Hot Water by Heat Pump

PROPER design of the water distributing system in a building is necessary in order that the various fixtures may function properly. The amount of either hot or cold water used in any building is variable, depending on the type of structure, usage, occupancy, and time of day. It is necessary to provide piping, water heating, and storage facilities of sufficient capacity to meet the peak demand without wasteful excess in either piping or equivalent cost.

SIZING COLD WATER SUPPLY PIPING

One of the important items that must be determined before any part of the water-piping system can be sized, is the probable rate of flow in any particular reach of piping. The rate of flow in the service line, risers, and main branches, however, will rarely be equal to the sum of the rates of flow of all connected fixtures. In fact, the probability that every fixture in a large group will be in use at the same time is so remote that it would be very poor engineering practice to design the piping to take care of such simultaneous flow.

The demand load in building water supply systems cannot be determined exactly and is not readily standardized. The two main problems to be considered are: (1) the satisfactory supply of water for a given fixture, and (2) the number of fixtures which may be assumed to be in use at the same time.

The minimum flow that will be satisfactory to the consumer depends greatly on the consumer, his standard of living, his professional needs, size of family, garden requirements, and similar factors. Depending on these factors, the per capita water consumption for domestic use usually varies between 20 and 80 gal per day. Experience indicates that the type of dwelling also has considerable influence on the water consumption.

In *apartment houses* the per capita daily water consumption is generally higher than in single-family houses. This is due to the use of a central metering system which is not conducive to the saving of water, and to the long hot water lines which cause high heat losses and an increase in the wasting of the cooled water. In designing water supply systems for apartment houses, a daily per capita water consumption of 50 gal may be considered a safe design figure.

Although a considerable number of *housing projects* have been developed throughout the United States, conclusive water consumption data have not yet been gathered. Nevertheless, it seems that the daily per capita water consumption in *housing projects* falls in between the consumption in apartment houses and that in single dwellings at the same geographical location. In general, a daily per capita water consumption of 40 gal can be used as a safe design figure for *housing projects*.

Table 1 gives the rate of flow desirable for many common types of fixtures, and the average pressure necessary to give this rate of flow. The pressure necessarily varies with fixture design, a much greater pressure being necessary with some fixtures to give the same rate of flow as in others. In general, the lower the quality of the faucet the greater will be the pressure required.

In estimating the load, the rate of flow is frequently computed in *fixture units*. One *fixture unit* is equivalent to 7.5 gal per min. Table 2 gives the demand weights in terms of fixture units for different plumbing fixtures under several conditions of service, and Fig. 1 gives the estimated demand in gallons per minute corresponding to any total number of *fixture units*. Fig. 2 shows an enlargement of Fig. 1 for a range up to 250 *fixture units*.

The estimated demand load for fixtures used intermittently on any supply pipe will be obtained by multiplying the number of each kind of fixture supplied through that pipe by its weight from Table 2, adding the products, and then referring to the appropriate curve of Figs. 1 or 2 to find the demand corresponding to the total fixture units. In using this method it should be noted that the demand for fixture or supply outlets other than those listed in the table of fixture units is not yet included in the estimate. The demands for outlets (such as hose connections, air-conditioning apparatus, etc.) which are likely to impose continuous demand during times of heavy use of the weighted fixtures, should be estimated separately and added to the demand for fixtures used intermittently, in order to estimate the total demand.

So far, the information presented makes possible the de-

Table 1.... Proper Flow and Pressure Required During Flow for Different Fixtures

Fixture	Flow Pressure[a]	Flow gpm
Ordinary basin faucet	8	3.0
Self-closing basin faucet	12	2.5
Sink faucet—$\frac{3}{8}$ in.	10	4.5
Sink faucet—$\frac{1}{2}$ in.	5	4.5
Bathtub faucet	5	6.0
Laundry tub cock—$\frac{1}{2}$ in.	5	5.0
Shower	12	5.0
Ball-cock for closet	15	3.0
Flush valve for closet	10–20	15–40[b]
Flush valve for urinal	15	15.0
Garden hose, 50 ft, and sill cock	30	5.0

[a] Flow pressure is the pressure in the pipe at the entrance to the particular fixture considered.
[b] Wide range due to variation in design and type of flush-valve closets.

termination of the *design rate of flow* in any particular section of piping. The next general step is to determine the *size* of piping.

As water flows through a pipe, the pressure continually decreases along the pipe, due to loss of energy from friction. The problem is then one of ascertaining the minimum pressure in the street main, and the minimum pressure required for the operation of the topmost fixture. (A pressure of 15 psi is ample for flush valves, but reference should be made to the manufacturers' requirements. A minimum of 8 psi should be allowed for other fixtures.) The pressure differential thus obtained will be available for overcoming pressure losses in the distributing system, and in overcoming the difference in elevation between the water main and the highest fixture.

The pressure loss, in pounds per square inch, caused by the difference in elevation between the street main and the highest fixture, may be obtained by multiplying the difference in elevation in feet by the conversion factor 0.43.

When water flows through a pipe, friction occurs as the result of the sliding of water particles past one another. If the pipe wall is rough, the roughness projections cause additional friction, owing to the development of increased turbulence in the flowing water. As the water flows along a uniform pipe, the pressure decreases as a result of a dissipation of energy arising from the internal friction set up by viscosity of the water. This loss in energy is shown by the loss of pressure. The pressure loss is proportional to the length of straight uniform pipes, and varies greatly with flow velocity, pipe diameter, roughness of pipe, and water temperature.

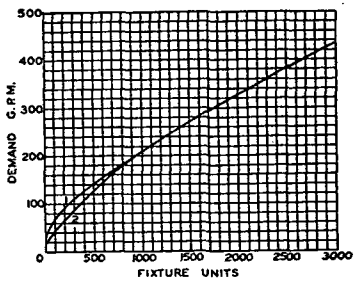
Table 2.... Demand Weights of Fixtures in Fixture Units[a]

Fixture or Group[b]	Occupancy	Type of Supply Control	Weight in Fixture Units[c]
Water closet	Public	Flush valve	10
Water closet	Public	Flush tank	5
Pedestal urinal	Public	Flush valve	10
Stall or wall urinal	Public	Flush valve	5
Stall or wall urinal	Public	Flush tank	3
Lavatory	Public	Faucet	2
Bathtub	Public	Faucet	4
Shower head	Public	Mixing valve	4
Service sink	Office, etc.	Faucet	3
Kitchen sink	*Hotel or restaurant*	Faucet	4
Water closet	Private	Flush valve	6
Water closet	Private	Flush tank	3
Lavatory	Private	Faucet	1
Bathtub	Private	Faucet	2
Shower head	Private	Mixing valve	2
Bathroom group	Private	Flush valve for closet	8
Bathroom group	Private	Flush tank for closet	6
Separate shower	Private	Mixing valve	2
Kitchen sink	Private	Faucet	2
Laundry trays (1–3)	Private	Faucet	3
Combination fixture	Private	Faucet	3

[a] For supply outlets likely to impose continuous demands, estimate continuous supply separately and add to total demand for fixtures.

[b] For fixtures not listed, weights may be assumed by comparing the fixture to a listed one using water in similar quantities and at similar rates.

[c] The given weights are for total demand. For fixtures with both hot and cold water supplies, the weights for maximum separate demands may be taken as ¾ the listed demand for the supply.



No. 1 for system predominantly for flush valves.
No. 2 for system predominantly for flush tanks.

Fig. 1.... Estimate Curves for Demand Load

On the basis of inside surface conditions, pipes may be classified as smooth, fairly rough, and rough, as follows:

Smooth. The pipe surface shows no perceptible roughness. *Pipes made of copper, brass, or lead may usually be classified as smooth.*

Fairly Rough. All ordinary pipes, such as wrought iron, galvanized iron, steel, and cast iron, after a few years of usage, may be called fairly rough.

Rough. Pipes that have deteriorated fairly rapidly for some *10 or 15 years after being laid, are classified as rough.*

Figs. 3, 4, and 5 give the pipe friction losses corresponding to these three types of pipes for various nominal diameters for a water temperature of 50 F.[1] *Example 1* will illustrate the use of the charts.

Example 1: A 2½-in. fairly rough pipe supplies 100 gpm of water. Find the friction loss in head if the pipe length is 200 ft.

Solution: Enter Fig. 4 at 100 gpm, and move along this line until it intersects the 2½-in. diameter line. From this intersection point, move vertically down and read 4.5 psi friction loss per 100 ft of pipe length. Then the *total friction* loss will be $2 \times 4.5 = 9$ psi.

The pressure losses in the distributing system will consist of the pressure losses in the piping itself, plus the pressure losses in the pipe fittings, valves, and the water meter, if any. Approximate design pressure losses for disc-type meters for various rates of flow are given in Fig. 6. Flow limits for disc-type meters, which may be regarded as the limits of recommended ranges in capacities, are given in Table 3. Manufacturers should be consulted for data on exact pressure losses and capacities since these vary for meters of different internal design.

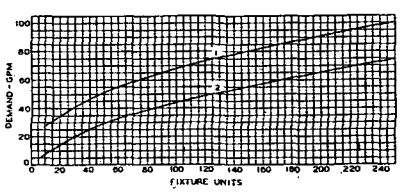


Fig. 2.... Section of Fig. 1 on Enlarged Scale

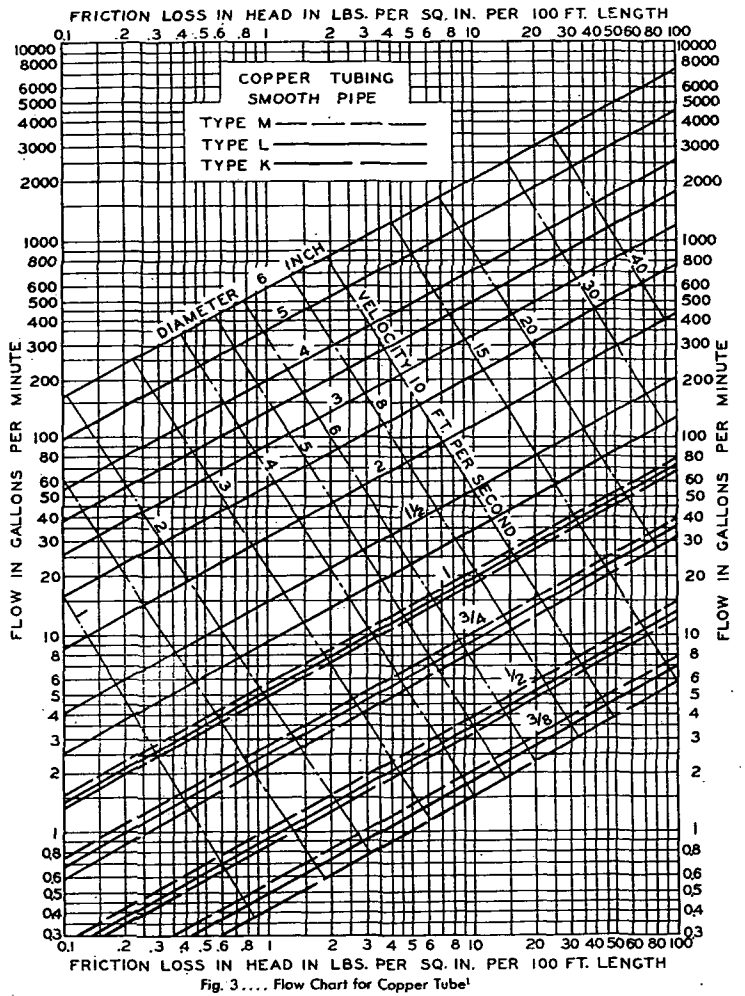


Fig. 3.... Flow Chart for Copper Tube[1]

Fig. 7 shows the variation of pressure loss with rate of flow for various types of faucets and cocks, based on experimental data obtained at the State University of Iowa.

The loss of pressure through any fitting or valve can be expressed in pounds per square inch for any given rate of flow. Experience has shown, however, that the simplest method of expressing losses in fittings and valves is to use the concept of an equivalent length of straight pipe. Thus it has

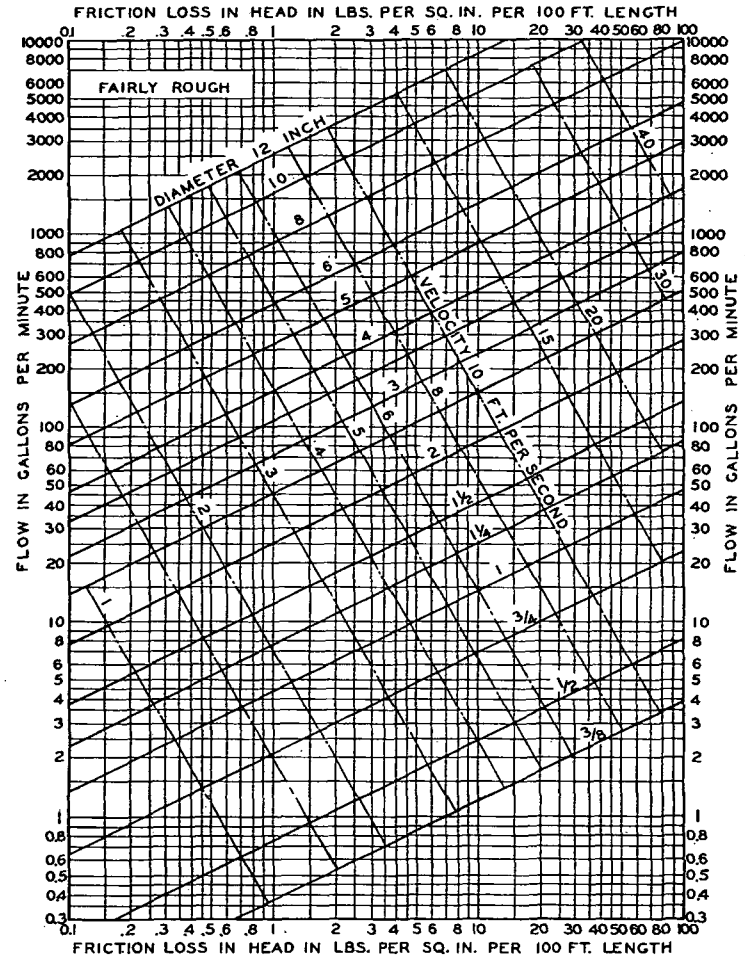


Fig. 4 .... Flow Chart for Fairly Rough Pipe[1]

been found, for example, that a 1-in., 90-deg elbow introduces a loss equivalent to 2.2 ft of straight 1-in. pipe. Therefore, for each 1-in., 90-deg elbow, 2.2 ft of 1-in. pipe are added to the total length of 1-in. pipe.

Estimated pressure losses for pipe fittings and valves in terms of equivalent pipe length are shown in Table 4.

Table 5 lists the equivalent lengths for various special types of apparatus and fittings. The loss in water meters varies considerably with the design even in meters of the same nominal size. The values given in Table 5 are ample for the well-known meters now on the market.

The water demand for hose bibbs or other large demand

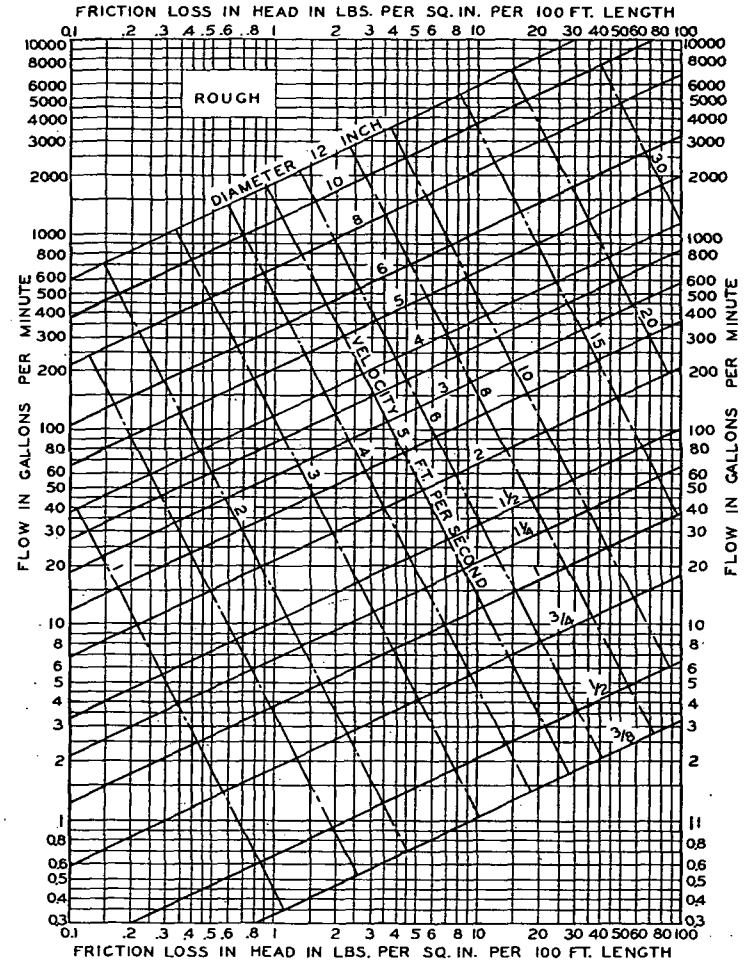
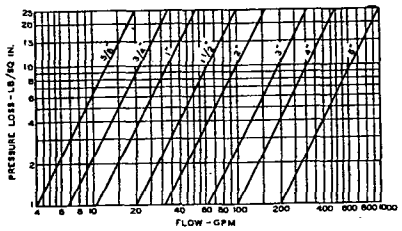


Fig. 5 .... Flow Chart for Rough Pipe[1]

fixtures taken off the building main is frequently the cause of inadequate water supply to the upper floor of a building. This condition may be prevented by sizing the distribution system so that the pressure drops from the street main to all fixtures are the same. It is good practice to maintain the building main of ample size (not less than 1 in. where possible) until all branches to hose bibbs have been connected. Where the street main pressure is excessive and a pressure reducing valve is used to prevent water hammer or excessive pressure at the fixtures, it is frequently desirable to connect hose bibbs ahead of the reducing valve.

The principles involved in sizing either up-feed or down-



(See manufacturer's data for exact value for meter used)

Fig. 6 .... Pressure Losses in Water Meters

feed systems are the same. The principal difference in procedure is that in the down-feed system, the difference in elevation between the house tank and the fixtures provides the pressure required to overcome pipe friction.

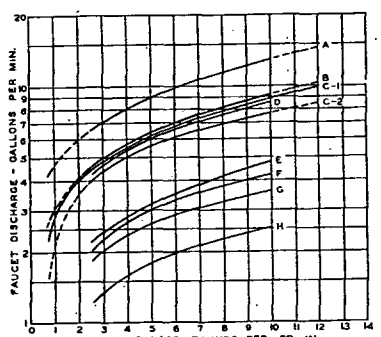
Procedure for Sizing Cold Water Systems

The recommended procedure for sizing piping systems is outlined in following paragraphs 1 to 6, inclusive.

1. Draw a sketch of the main lines, risers, and branches, and indicate the fixtures to be served. *Indicate the rate of flow of each fixture.*

2. Using Table 2, compute the demand weights of the fixtures in fixture units.

3. Determine the total demand in fixture units and, using Fig. 1 or Fig. 2, find the expected demand in gallons per minute.



A. ½-in. laundry bibb (old style).
B. Laundry compression faucet.
C-1. ½-in. compression sink faucet (Mfr. 1).
C-2. ½-in. compression sink faucet (Mfr. 2).
D. Combination compression bathtub faucets (both open).
E. Combination compression sink faucet.
F. Basin faucet.
G. Spring self-closing faucet.
H. Slow self-closing faucet.
(Dashed lines indicate recommended extrapolation)

Fig. 7 .... Variation of Pressure Loss with Rate of Flow for Various Faucets and Cocks

Table 3 .... Performance Requirements of Water Meters[a]

Size, In.	Normal Test-Flow Limits, Gpm	Minimum Test-Flow, Gpm
⅝	1 to 20	¼
¾	2 to 34	½
1	3 to 53	¾
1½	5 to 100	1½
2	8 to 160	2
3	16 to 315	4
4	28 to 500	7
6	48 to 1,000	12

[a] *American Water Works Association Standards:*
Registration. The registration on the meter dial shall indicate the quantity recorded to be not less than 98 percent nor more than 102 percent of the water actually passed through the meter while it is being tested at *rates of flow* within the specified limits herein under normal test flow limits: There shall be not less than 90 percent of the actual flow recorded when a test is made at the rate of flow set forth under minimum test flow.

4. Determine the equivalent length of pipe in the main lines, risers, and branches. Since the sizes of the pipes are not known, the exact equivalent length for various fittings, *etc.*, cannot be made. Add the equivalent lengths, starting at the street main and proceeding along the service line, the main line in the *building, and up the riser to the top* fixture of the group served.

5. Determine the average minimum pressure in the street main and the minimum pressure required for the operation of the topmost fixture. This latter pressure should be 8 to 15 psi.

6. Calculate, by means of Equation 1, the approximate design value of the average pressure drop per 100 ft of pipe in the equivalent length determined in paragraph 4.

$$p = [P - 0.43H - 10]\frac{100}{L} \qquad (1)$$

where

p = average pressure loss per 100 ft of equivalent length of pipe, psi.
P = pressure in street main, psig.
H = height of highest fixture above street main, feet.
L = equivalent length determined in paragraph 4, feet.

If the system is of the down-feed supply from a gravity tank, the height of water in the tank, converted to pounds per square

Table 4 .... Allowance in Equivalent Length of Pipe for Friction Loss in Valves and Threaded Fittings

Diameter of Fitting, In.	Equivalent Length of Pipe for Various Fittings: 90-Deg Standard Ell, Ft	45-Deg Standard Ell, Ft	90-Deg Side Tee, Ft	Coupling or Straight Run of Tee, Ft	Gate Valve, Ft	Globe Valve, Ft	Angle Valve, Ft
⅜	1	0.6	1.5	0.3	0.2	8	4
½	2	1.2	3	0.6	0.4	15	8
¾	2.5	1.5	4	0.8	0.5	20	12
1	3	1.8	5	0.9	0.6	25	15
1¼	4	2.4	6	1.2	0.8	35	18
1½	5	3	7	1.5	1.0	45	22
2	7	4	10	2	1.3	55	28
2½	8	5	12	2.5	1.6	65	34
3	10	6	15	3	2	80	40
3½	12	7	18	3.6	2.4	100	50
4	14	8	21	4.0	2.7	125	55
5	17	10	25	5	3.3	140	70
6	20	12	30	6	4	165	80

Table 5 .... Equivalent Lengths of Iron Pipe to Give Same Loss as Special Fittings and Apparatus

Fitting Apparatus	Nominal Diameter of Pipe—Inches: ½	¾	1	1¼
30-gal Vertical hot water tank, ¾-in. pipe	4	17	56	—
30-gal Horizontal hot water tank, ¾-in. pipe	1.2	5	16	—
Water meters (No valves included)				
⅝ in. with ½-in. connections	6.7	28	90	—
⅝ in. with ¾-in. connections	4.8	20	64	—
¾ in. with ¾-in. connections	3.4	14	45	—
1 in. with 1-in. connections	—	9	30	115
1¼ in. with 1-in. connections	—	4.4	14	54
Water softener	—	50–200	—	—

inch by multiplying by 0.43, replaces the street main pressure, and the term 0.43 H in Equation 1 is added instead of subtracted in calculating the term p. In this case, H will be the vertical distance of the fixture below the bottom of the tank.

7. From the expected rate of flow, determined as in paragraph 3, and the value of p, calculated as in paragraph 6, choose the sizes of pipe from Figs. 3, 4, or 5.

Example 2: Assume a minimum street main pressure of 55 psig; a height of topmost fixture above street main of 50 ft; a developed pipe length from water main to highest fixture of 100 ft; a total load on the system of 50 fixture units; and that the water closets are flush-valve operated. Find the required size of supply main.

Solution: From Fig. 2 the estimated peak demand is found to be 51 gpm. From Table 3 it is evident that several sizes of meters would adequately measure this flow. For a trial computation choose the 1½-in. meter. From Fig. 6 the pressure drop through a 1½-in. disc-type meter for a flow of 51 gpm is found to be 6.5 psi.

Then the pressure drop available for overcoming friction in pipes and fittings is $55 - (15 + 50 \times 0.43 + 6.5) = 12$ psi.

At this point it is necessary to make some estimate of the equivalent pipe length of the fittings on the direct line from the street main to the highest fixture. The exact equivalent length of the various fittings cannot now be determined since the pipe sizes of the building main, riser, and branch leading to the highest fixture are not known as yet, but a first approximation is necessary in order to make a tentative selection of pipe sizes. If the computed pipe sizes differ from those used in determining the equivalent length of pipe fittings, a recalculation will be necessary, using the computed pipe sizes for the fittings. For the purposes of this example assume that the total equivalent length of the pipe fittings is 50 ft.

Then the permissible pressure loss per 100 ft of equivalent pipe is $12 \times 100/(100 + 50) = 8$ psi.

Assuming that the corrosive and caking properties of the water are such that Fig. 4 for fairly rough pipe is applicable, a 2-in. building main will be adequate.

Table 6 .... Computation of Branch Size in Example 2

Fixtures No. and Kind	Fixture Units (From Table 2 and Note c)	Demand (From Fig. 2) Gpm	Pipe Size (From Fig. 4) In.
3 flush valves	3 x 6 = 18		
2 bathtubs	¾ (2 x 2) = 3		
3 lavatories	¾ (3 x 1) = 2.25		
Total	23.25	38	1½

Table 7 .... Pipe Sizes for Cooling Towers[a]

Rated Tons of Refrig.	Cooling Water Gpm	Pipe Sizes (Nominal Inches): Inlet to Tower	Outlet from Tower
3 to 5	10 to 18	1½	1½
7 to 15	20 to 45	2	2
20	65	2½	3
25	86	2½	4
35	115	3	4
50	170	3	4
75	225	5	6
100	300	5	6
150	450	5	8
200	600	6	8
250	750	8	8

The sizing of the *branches* of the building main, the *risers*, and *fixture branches* follows the principles outlined. For example, assume that one of the branches of the building main carries the cold water supply for 3 water closets, 2 bathtubs, and 3 lavatories. Using the permissible pressure loss of 8 psi per 100 ft, the size of branch determined from Table 2 and Figs. 1 and 4 is found to be 1½ in. Items entering the computation of pipe size are given in Table 6.

COOLING WATER PIPING

Water is very frequently used in refrigeration systems, cooling towers, and other similar installations. In designing the piping system of such installations, the principles of hydraulics, as already outlined, are employed. Nevertheless, there are several practical items having particular application to cooling installations which are outlined in the following paragraphs. It is important that the designer be familiar with them.

In choosing pipe material, the problem of corrosion should be kept in mind to prevent failure of the system. If the water is not severely corrosive, wrought-iron or steel piping may be used; otherwise, galvanized-steel piping may be preferred. If sea water is used as the circulating medium, it is advisable to use alloys such as admiralty metal in pipe and tubing. In refrigeration condensers where water is the cooling medium, iron pipe is commonly employed.

In regard to assembly, cast-iron flanges or welded joints are to be preferred to screwed joints wherever possible.

Valves used in circulating systems may be of the globe, gate, or angle types. If water is the circulating medium, brass valves are usually used. However, if the circulating medium is an electrolyte, such as brine, then it is preferable to use valves made of the same material as the piping itself.

The friction loss in the piping may be determined from Fig. 4 for fairly rough pipe. If the coolant is brine, a correction for the proper density must be made. Experience indicates that in sizing piping for cooling systems, a pressure drop of the order of 2 to 3 psi per 100 ft of pipe length, and a fluid velocity of 3 to 8 fps, yield most economical results.

In the case of cooling towers, the amount of circulating water is about 3 gpm per ton of refrigeration, when based on a design wet-bulb temperature of about 76 F. Table 7 gives pipe sizes frequently used for various sizes of cooling towers, assuming a hot water temperature of 95 F and a cold water temperature of 85 F.[2]

Table 8.... Maximum Daily (24-Hr) Requirements for Hot Water in Gallons

	No. of Rooms	Number of Bathrooms 1	2	3	4	5
Apartments and Private Homes	1	60	—	—	—	—
	2	70	—	—	—	—
	3	80	—	—	—	—
	4	90	120	—	—	—
	5	100	140	—	—	—
	6	120	160	200	—	—
	7	140	180	220	—	—
	8	160	200	240	250	—
	9	180	220	260	275	—
	10	200	240	280	300	—
	11	—	260	300	340	—
	12	—	280	325	380	450
	13	—	300	350	420	500
	14	—	—	375	460	550
	15	—	—	400	500	600
	16	—	—	—	540	650
	17	—	—	—	580	700
	18	—	—	—	620	750
	19	—	—	—	—	800
	20	—	—	—	—	850

Hotels	Room with basin	10
	Room with bath—transient	50
	Room with bath—resident	60
	2 Rooms with bath	80
	3 Rooms with bath	100
	Public shower	200
	Public basins	150
	Slop sink	30
Office Buildings	White-collar worker (per person)	2.0
	Other workers (per person)	4.0
	Cleaning per 10,000 sq ft	30.0
Hospitals	Per bed	80–100

ESTIMATING HEATING LOAD AND STORAGE CAPACITY

The maximum daily and the maximum hourly hot water demand form the basis for the selection of the heater and the storage tank.

In general, two-thirds of the total daily water consumption is hot water. For residential dwellings, a design value of about 20 to 30 gal per (person) (day) may be assumed, but it should be remembered that the hot water used will depend on the number of rooms and the number of bathrooms in any house or apartment. Table 8 gives estimates of the maximum hot water requirements in 24 hr in various types of buildings.

In estimating the size of hot water storage tank required, and the heating capacity to be provided, either from the boiler or from an independent domestic hot water heater, it is necessary to know the total quantity of water to be heated per day, and the maximum amount which will be used in any one hour, as well as the duration of the peak load.

In cases where the requirements for hot water are reasonably uniform, as in residences, apartment buildings, hotels, and the like, smaller storage capacity is required than in the case of factories, schools, and office buildings, where practically the entire day's usage of hot water occurs during a very short period. Correspondingly, the heating capacity must be proportionately greater with uniform usage of hot water than with intermittent usage, where there may be several hours between peak demands during which the water in the storage tank can be brought up to temperature. As a general rule, it is desirable to have a large storage capacity in order that the heating capacity, and consequently the size of the heater, or the load on the heating boiler, may be as small as possible.

Table 9.... Estimated Hot Water Demand Characteristics for Various Types of Buildings

Type of Building	Hot Water Required Per Person	Max. Hourly Demand in Relation to Day's Use	Duration of Peak Load Hours	Storage Capacity in Relation to Day's Use	Heating Capacity in Relation to Day's Use
Residences, apartments, hotels, etc.	40 gal per day[a]	1/7	4	1/5	1/7
Office buildings	2 gal per day[a]	1/5	2	1/5	1/6
Factory buildings	5 gal per day[a]	1/3	1	2/5	1/8
Restaurants				1/10	1/10
Restaurants 3 meals/day		1/10	8	1/5	1/10
Restaurants 1 meal/day		1/5	2	2/5	1/6

[a] At 140 F.

In estimating the hot water which can be drawn from a storage tank, it should be borne in mind that only about 75 percent of the volume of the tank is available, since, by the time this quantity has been drawn off, the incoming cold water has cooled the remainder down to a point where it can no longer be considered hot water.

Where steam from the heating boiler is used to heat domestic hot water, the computed load on the boiler should be increased by 4 sq ft EDR (equivalent direct radiation) for every gallon of water per hour heated through a 100 deg rise. The actual requirement is $(100 \times 8.33)/240 = 3.48$ sq ft per gal heated 100 deg. The value of 4 allows for transmission losses.

There are two methods in common use for estimating the hot water requirements of a building: (1) by the number of people, and (2) by the number of plumbing fixtures installed. Where the number of people to be served can be reasonably estimated, the data in Table 9 may be used.

Example 3: Determine the heater size and storage tank capacity for a residence housing five people.

Solution: From Table 9, a residence housing five people would have a daily requirement of $5 \times 40 = 200$ gal per day, and a maximum hourly demand of $200 \times 1/7 = 28.5$ gal. The heater should have a storage capacity of $200 \times 1/5 = 40$ gal, and a heating capacity of $200 \times 1/7 = 28.5$ gal per hr.

The conditions given in *Example 3* may be cited as average. It is possible to vary the storage and heating capacity by increasing and decreasing one over the other. Such a condition is illustrated in *Example 4*.

Example 4: Determine the required heater capacity for an apartment housing 200 people, if the storage tank has a capacity of 1000 gal. What heater capacity will be required if the storage tank is changed to 2500 gal capacity?

Solution: Assume an apartment house housing 200 people. From the data in Table 9: Daily requirements $= 200 \times 40 = 8000$ gal. Maximum hours demand $= 8000 \times 1/7 = 1140$ gal.

Duration of peak load = 4 hr. Water required for 4-hr peak $= 4 \times 1140 = 4560$.

If a 1000-gal storage tank is used, hot water available from the tank $= 1000 \times 0.75 = 750$. Water to be heated in 4 hr $= 4560 - 750 = 3710$ gal. Heating capacity per hour $= 3710/4 = 930$ gal.

If instead of a 1000-gal tank, a 2500-gal tank had been installed, the required heating capacity per hour would be $[4560 - (2500 \times 0.75)]/4 = 671$ gal.

Table 10 may be used to determine the size of water heating equipment from the number of fixtures. To obtain the *probable maximum demand*, multiply the total quantity for the fixtures by the *demand factor* in line 11. The heater or coil should have a water heating capacity equal to this *probable maximum demand*. The storage tank should have a capacity equal to the *probable maximum demand* multiplied by the storage capacity factor in line 12. *Example 5* will illustrate the procedure.

Example 5: Determination of heater and storage tank size for an apartment building from number of fixtures.

60 lavatories	× 2 =	120 gal per hr
30 bathtubs	× 20 =	600 gal per hr
30 showers	× 75 =	2250 gal per hr
60 kitchen sinks	× 10 =	600 gal per hr
15 laundry tubs	× 20 =	300 gal per hr
Possible maximum demand	=	3870 gal per hr
Probable maximum demand	=	3870 × 0.30
	=	1161 gal per hr
Heater or coil capacity	=	1161 gal per hr
Storage tank capacity	=	1161 × 1.25
	=	1450 gal

Although, in private dwellings a water temperature of 140 F is reasonable for dishwashing, in public places sanitation regulations call for 180 F water. Most of the dishwashing machines now available on the market require 180 F water. The amount of 180 F water needed in restaurants per day may be determined according to the method outlined by the American Gas Association,[a] in the following paragraphs:

1. Multiply the number of meals per day by the number of dishes per meal (6 for low-price restaurants, 8 for medium-price restaurants, and 10 for high-price restaurants) to determine the total number of dishes per day.

2. Divide the total number of dishes per day, as determined by method in paragraph 1, by the average number of dishes per rack to find the number of racks per day.

3. Multiply the number of racks per day by the gallons of 180 F water (using 1.5 for single tank machines and 0.75 for two tank machines). This product will give the gallons of 180 F water per day for rinse sprays.

4. Multiply the number of meal periods per day (one, two, or three) by the dishwashing tank capacity in gallons, giving the gallons of 180 F water per day necessary to fill the tanks.

5. Add values from paragraphs 3 and 4 to obtain the total number of gallons of 180 F water required per day.

For purposes other than dishwashing, a considerable amount of 140 F water is also used. To find the daily 140 F water requirement in a restaurant, multiply the total number of meals served per day by the gallons of 140 F water per meal. Low priced restaurants on the average utilize 0.9 gal of 140 F water per meal, while medium- and high-price restaurants use 1.2 and 1.5 gal per meal, respectively.[a]

METHODS OF HEATING WATER

Hot water may be heated either by the direct combustion of fuel, by an intermediate carrier such as steam or hot water, or by electrically heated surfaces. The simplest method is to have the fire on one side of a metal barrier and water on the other. In such a method, if the surfaces for transferring heat are small, and if the water carries a heavy proportion of precipitable salts, the water passages may soon become clogged with resultant cracking or burning of the surface. A familiar example of such trouble is the water back in the kitchen stove, or the pipe coil inserted into the firebox of a warm air furnace or small boiler. The critical water tem-

Table 10.... Hot Water Demand per Fixtures for Various Types of Buildings

Gallons of water per hour per fixture, calculated at a final temperature of 140 F

	Apartment House	Club	Gymnasium	Hospital	Hotel	Industrial Plant	Office Building	Private Residence	School	Y.M.C.A.
1. Basins, private lavatory	2	2	2	2	2	2	2	2	2	2
2. Basins, public lavatory	4	6	8	6	8	12	6	....	15	8
3. Bathtubs	20	20	30	20	20	30	....	20	....	30
4. Dishwashers	15	50–150	....	50–150	50–200	20–100	....	15	20–100	20–100
5. Foot basins	3	3	12	3	3	12	....	3	3	12
6. Kitchen sink	10	20	....	20	20	20	....	10	10	20
7. Laundry, stationary tubs	20	28	....	28	28	....	....	20	....	28
8. Pantry sink	5	10	....	10	10	....	....	5	10	10
9. Showers	75	150	225	75	75	225	....	75	225	225
10. Slop sink	20	20	....	20	30	20	15	15	20	20
11. Demand factor	0.30	0.30	0.40	0.25	0.25	0.40	0.30	0.30	0.40	0.40
12. Storage capacity factor[a]	1.25	0.90	1.00	0.60	0.80	1.00	2.00	0.70	1.00	1.00

[a] Ratio of storage tank capacity to probable maximum demand per hour.

perature at which the lime, magnesia, etc., collect on hot surfaces, varies with the character and proportions of the solids, but generally such deposits are not a serious trouble below 140 F.

Direct-Fired Water Heaters

Coal-burning, direct-fired water heaters may be constructed of cored cast-iron sections, or of steel. In some cases the external appearance of the cast-iron sections is the same as in heating boilers, but internally the cores are changed to enable the sections to withstand the city water pressure. In small capacity water heaters, efficiency is not considered so important as low first cost and ability to maintain a fire at a low rate of combustion, and consequently, such heaters are generally built with a *dry* section or firebrick lining at the base of the firepot to prevent too much chilling of the fuel. While mud and scale will eventually clog the waterways of any direct-fired heater, increased life may be obtained by providing a three-way cock in the return line between the heater and the bottom of the storage tank, so that water can be blown through the heater or the tank separately, at full line pressure, to clean out loose sediment. Cleanout openings in the bottom of the heater are advantageous, if used by operators of water heaters for periodic cleaning out of sediment.

Oil-burning, direct-fired water heaters usually are of steel, and operate with higher flame temperature and better efficiency than commensurate sized coal-burning heaters. As they have the same tendency as coal boilers to accumulate lime deposits, the water passages should be large in cross-section and accessible for periodic cleaning.

Gas-burning, direct-fired water heaters may be of the instantaneous or storage type. Instantaneous heaters are generally constructed of spiral water tubes of copper, around which the products of combustion circulate upward from high capacity burners. Storage-type heaters may include in one unit an insulated storage tank, a combustion chamber, flues, burner equipment, and controls, or may consist of a separate storage tank and external direct-fired water heater, which may be a so-called *side-arm* heater for small capacity, or a gas-fired boiler for larger capacity. Gas boilers used for direct hot water supply must be able to withstand the city water operating pressure. While direct-fired gas heaters are used generally for residences and small installations of 100 gal storage capacity or less, indirect heaters are recommended for larger installations.

Chimney connections for all direct-fired, fuel-burning water heaters are an important consideration. Refer to Chapter 36.

Electric Water Heaters

Electric water heaters for hot water service in residences and commercial buildings are predominantly of the automatic storage type. These generally have a cylindrical tank mounted vertically, with a primary electric heating element or resistor near the bottom and a secondary unit located within the upper one-quarter of the tank. Water heaters of single-heating-unit type have an element only at the bottom. The first cost of automatic electric heaters is somewhat higher than that of fuel-fired types, owing to the need for larger tank storage to compensate for the limitation of recovery rate, since for economical electric service the connected load should be kept low. Large tank capacity is necessary also with off-peak operation, where there are low electric rates for this class of service.

Table 11 .... Standard Rated Capacities of Electric Water Heaters

Tank Size Gallons		Rated Input, Watts		
		Two-Unit Heater		Single-Unit Heater
Nominal	Range	Primary	Secondary	
30	30 to 35	600	1000	1500
40	35 to 45	750	1250	2000
52	45 to 55	1000	1500	2500
66	55 to 70	1250	2000	3000
80	70 to 90	1500	2500	4000
110	90 to 115	2000	3000	
120	115 to 135	2500	4000	
140	135 to 175	3000	4000	

Instantaneous heaters are used in special cases only, because of the high electric power input required, for example, 15 kw to deliver 1 gpm with 100 deg temperature rise.

Capacities of household automatic storage-type water heaters, as partially standardized within the electrical industry, range from 30 to 140 gal nominal tank capacity with input 1600 to 7000 watts. Wattage input ratings of two-unit heaters are based on approximately 30 watts per gallon of tank capacity for the secondary unit and 20 watts for the primary unit. Single-unit heaters have tank sizes up to 80 gal with an input of 4000 watts, the ratings being based on 50 watts per gallon. The schedule of heater capacities shown in *ASA Standard* C72.1-1949 is reproduced in Table 11.

The wattages listed in Table 11 are minimum recommended values for continuous service. Some departure from the standardized watts-per-gallon ratios is to be expected, as experience indicates that increased hot water use with automatic laundry machines and dishwashers may call for larger heating units than have heretofore been found desirable. With off-peak operation the watts-per-gallon ratio likewise must be raised.

Rated voltage for water heaters is 240 volts, with a range 220 volts minimum to 248 volts maximum, but specified test performance is required to be met only at the 236 volts taken as normal design voltage. Thermostats controlling the heating units are normally set for 150 F, with upper limit of adjustment range of 170 to 180 F.

Heating units are of three general types, all of which give approximately the same service efficiency if selected to suit the operating conditions, as follows:

1. Immersion element, inserted horizontally through a threaded or flanged opening in side of tank.
2. Strap-on unit, externally mounted directly against the tank shell and held in close contact by a band or strap encircling the tank.
3. Side-arm or *outside circulation* unit, consisting of a vertical cylindrical housing or heat exchanger which contains an immersion heating element, connected by pipes into top and bottom of the storage tank.

For the third type, which is less commonly used, thermostats separate from the heating units are mounted within the storage tank.

A fourth type of heater, applied only in exceptional cases and for large installations with alternating current, employs the water itself as a resistor in which electrodes are immersed. Close temperature regulation and compensation for the effect of variable amount and composition of solids and gases in the water are difficult to obtain with small unattended heaters of this kind. The electrode type of construction has in the past been used also for generating steam in so-called electric boilers, mainly for industrial plants; however, in only few instances today is electric energy obtainable at cost low enough to make it competitive with fuel-fired boilers for steam production.

Tanks of electric heaters are regularly constructed to withstand a 300 psig hydrostatic test pressure and are rated for a safe working pressure of 127.5 psig. Materials in contact with the water are selected to resist corrosion caused by solid and gaseous impurities occurring in some locations. It may be desirable also to provide *sacrificial metal* anodes of magnesium, zinc, or alloys to protect the tank and its piping connections. A safety relief valve of pressure-actuated type must be included; it may optionally be actuated also by internal temperature. A heat trap at the hot water outlet, if not provided by manufacturer within or outside the tank shell and inside the insulated jacket, may be placed in the external piping. The trap consists of vertical pipe at least 6 in. long having downward flow, to retard circulation of hot water within the distribution piping system under standby conditions.

Occasionally, an uninsulated tempering tank is installed ahead of an electric water heater, enabling the cold inlet water to absorb heat from the basement or room air—if conditions are such that objectionable sweating does not occur. The practice of connecting a tempering tank with a firebox coil, so that the electric heater furnishes heat only when there is deficiency from the boiler or furnace, is not recommended.

Delivery performance efficiency of heaters meeting the electrical-industry standard 24-hour operation test is 90 percent minimum, which allows 10 percent for thermal loss through the jacket insulation. In service, the quantity of water withdrawn from an automatic storage heater has considerable influence on overall efficiency. With increasing quantities, service efficiency rises and approaches the upper limit fixed solely by the jacket thermal loss; on the other hand, if little or no water is drawn, efficiency approaches zero since no useful work is done. Hence oversizing a water heater by having too large a tank impairs the service efficiency. Excessive heating-unit wattage causes undesirably high electric demand. Undersizing of heaters obviously results in inadequate and unsatisfactory hot water service.

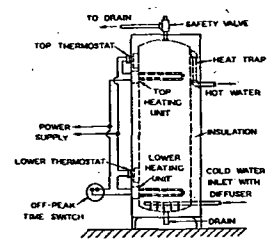


Fig. 8 .... Domestic Hot Water Heater for Off-Peak Service

Selection of electric heater capacity for residences is made by reference to rating tables which take into account the number of occupants, number of bathrooms or plumbing fixtures, and presence of appliances such as laundry equipment, dishwashers, and the like. Information of this kind is given in Tables 2, 8, 9 and 10 and in design manuals issued by the electric utility industry and by heater manufacturers. For commercial applications, the hourly hot water requirements must be analyzed and the best combination of tank capacity and heating element rating determined, with due consideration of electric power demand created in relation to the user's other electric demands.

A common type of electric service for water heating is the *off-peak* service under which the charge for energy consumed during designated hours is lower than for normal or unrestricted service. With this system, the operation of heaters is confined to such hourly intervals during the 24 hours as will (*a*) avoid creating peak electrical demands by the customer in excess of the demand resulting from the customer's other uses of electricity, or (*b*) keep the water heating load off the electric utility system during peak hours. In general, Case (*a*) occurs where rate schedules contain a demand charge component, and Case (*b*) where a special low rate applies to off-peak consumption. Water heater circuits are controlled with relays actuated by an electric-clock mechanism or by a *carrier-current* impulse impressed at suitable intervals on the electric distribution system, or by an automatic load-limiting type of control.

For general information on automatic control see Chapter 43.

One arrangement of domestic hot water heater for off-peak service, shown in Fig. 8, has the lower heating unit under control of an off-peak switch. Upper unit is connected directly to the line, so that if the reserve of hot water becomes depleted, the top thermostat brings its heating unit into operation only until the upper portion (usually about 25 percent) of tank contents reaches the thermostat temperature setting.

Indirect Water Heaters

In the indirect method, either steam or hot water is used for heating the water. With steam, the water to be heated is preferably circulated around the outside of the steam tubes which are submerged within a tank. A typical indirect heater using steam is shown in Fig. 9. The coils usually are of copper, and are U-shaped to permit expansion and contraction. The shell may be of steel, with a protective coating or with a special inside protective lining, or may be of copper or copper alloy. Where straight heating tubes are used, one end of the tube is usually expanded into a *floating* head to take

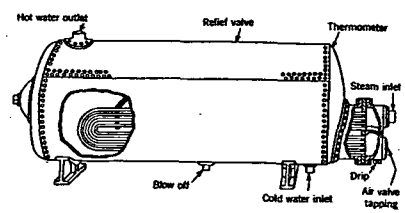


Fig. 9 .... Indirect Water Heater

care of expansion. The coils should be capable of easy withdrawal for inspection and for removal of scale. Instead of steam, the heating medium may also be hot water inside the tubes.

Another method of transferring heat from a heating boiler to the domestic water is illustrated in Fig. 10. The water heater is generally a cast-iron shell within which there is located a spiral copper coil. Hot water from the boiler circulates inside the shell and around the coil, and returns to the boiler, while domestic water from the storage tank circulates inside the coil. The storage tank should be installed with the

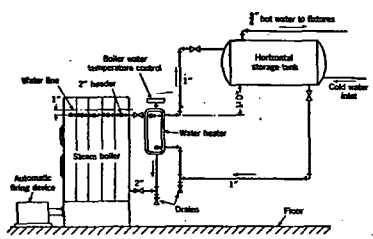


Fig. 10 .... Indirect Water Heater Mounted on Side of Boiler

bottom of the tank as far above the boiler as possible. Horizontal storage tanks of less than 18- or 20-in. diameter are not recommended because of the difficulty of preventing the hot and cold water from mixing, especially when large quantities of water are withdrawn. In Fig. 11 the heat-transfer surface is placed inside the boiler instead of in a separate vessel, but otherwise the operation is similar to that of Fig. 10. This arrangement with vertical tank is commonly used for small domestic installations.

Sometimes the heating element is located inside of the larger-type fire tube boilers and small residential boilers. In this case the heat-transfer surface is in the form of a number of straight copper tubes, with rear U-bends or a floating head, inserted through a special opening in the boiler. While the coil may be placed in the stream space above the waterline of a steam boiler, it is usually placed below the waterline. Long coils of small diameter tubing, immersed in the water, are widely used without storage tanks. The rate of flow through the coil is limited by the friction loss in the coil, and by fittings and restrictions, so that the water attains the desired temperature in one passage through the coil. This arrangement is frequently found in oil burner installations where the heating boiler, either steam or hot water type, is used to supply hot water during the summer. A thermostatic three-way mixing valve is frequently used to maintain a uniform temperature of the hot water supply to the plumbing fixtures.

In order to reduce clogging by precipitated solids, water heating plants sometimes develop steam in a closed circuit, transferring the heat through a tubular heater to the domestic water. The water in the primary heater, exposed to the high temperature of the fire, is repeatedly used and hence, has no appreciable tendency to deposit scale, while the domestic water, heated by steam at a much lower temperature than that of the fire, also exhibits a much reduced tendency to precipitate dissolved salts. Water characteristics, the effect of impurities, and means of improving the quality of the water are important items, as brought out in Chapter 55.

COMPUTING HEAT-TRANSFER SURFACE

The area of the inside surface of a heating coil may be determined from Equation 2.

$$A = \frac{Q \times 8.33(t_2 - t_1)}{U \times t_m} \qquad (2)$$

where

A = surface area of coil, square feet.
Q = quantity of water heated, gallons per hour.
t_2 = hot water outlet temperature, Fahrenheit.
t_1 = cold water inlet temperature, Fahrenheit.
U = coefficient of heat transmission, Btu per (hour) (square foot) (Fahrenheit degree logarithmic mean temperature difference).
For copper or brass coils U = 240 (steam) and 100 (hot water).
For iron coils U = 160 (steam) and 67 (hot water).

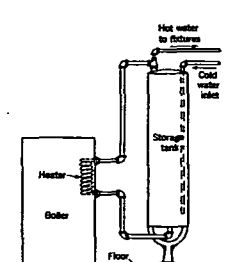


Fig. 11 .... Indirect Water Heater Placed in Boiler

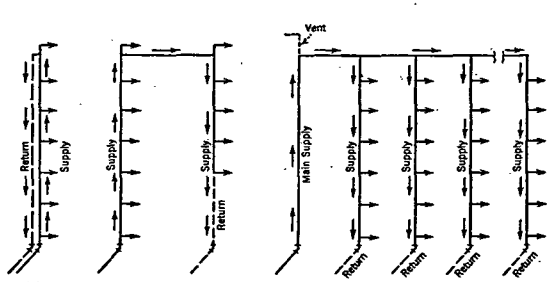


Fig. 12 .... Methods of Arranging Hot Water Circulation Lines

Table 12 .... Coefficient of Heat Transfer of Instantaneous Water Heaters

U = Btu per (hr) (sq ft) (Fahrenheit degree logarithmic mean temperature difference)

Boiler water temperature	210	200	180
U	225	175	150

t_m = logarithmic mean of the difference between the temperature of the heating medium and the average water temperature, and is *approximately*:

$$t_m = \left[\frac{(t_2 + t_1)}{2}\right]$$

t_t = temperature of the heating medium, Fahrenheit.

Equation 2 may be used to check the heating coil ratings under temperatures other than those stated in the manufacturer's published ratings.

Example 6: What area of copper transfer surface will be required to heat 70 gal of water per hour from 40 to 180 F with boiler water at 220 F?

Solution:

$$t_m = \left[220 - \frac{(180 + 40)}{2}\right] = 110$$

$$A = \frac{70 \times 8.33(180 - 40)}{100 \times 110} = 7.42 \text{ sq ft}$$

For instantaneous submerged heaters, the surface required will depend upon (1) the velocity of water in the tubes, (2) the boiler water temperature, (3) the inlet water temperature, (4) the outlet water temperature, (5) the cleanliness of the coil surface, and (6) the condition of the boiler water surrounding the coil. If the heater is located in the water of an actively steaming part of a boiler, the heat transfer may be twice as great as would be obtained if the water surrounding the coil were circulating slowly. Ratings of instantaneous water heating coils will therefore vary greatly, depending upon the assumptions made regarding the conditions of operation. The values of the coefficient of heat transmission for instantaneous heaters, shown in Table 12, are conservative.

For a coil in which heat is transferred from steam to water, the value of $U = 300\sqrt{v}$ may safely be used (v = velocity of water in feet per second).

The rate of heat transfer, between steam or water as the carrier, and the domestic water, is influenced by the rate of movement of both the carrier and the water which receives the heat. For this reason, where the transfer occurs from heating system water to domestic water, it is good practice to install a circulating pump to insure rapid movement of the boiler water.

In view of the high condensation rates obtained when steam is used with gravity circulation from the boiler, particularly when there is a sudden demand followed by an inflow of cold water, the bottom of a steam heat transfer element always should be at least 30 in. above the boiler water line, and the steam and condensate return pipes should be of liberal size. Otherwise, water hammer and reduced capacity may result, due to imperfect drainage of condensate.

When connecting a transfer-type hot water heater below the waterline of a cast-iron steam boiler having vertical sections, there should be a separate tapping for water circulation into every section of the boiler, as shown in Fig. 10, unless the boiler has large top nipple ports providing intersectional circulation. If the top nipples are entirely within the boiler steam space, no internal circulation occurs between sections. Steaming may then occur in the boiler sections not connected to the heater and, further, the unconnected sections will not deliver any heat to the water heater.

HOT WATER SUPPLY PIPING

It is common practice to provide circulating piping in all hot water supply systems in which it is desirable to have hot water available continuously at the fixtures. In average-sized and small residences and systems, in which the piping from the heater to the fixtures is short, return circulating piping is generally omitted in order to reduce installation cost, and to reduce heat loss from the piping, particularly during periods of no water demand.

The hot water supply may be distributed by either an up-feed or down-feed piping system. Three common methods of arranging the circulating lines are shown in Fig. 12. Although the diagrams apply to multi-story buildings, the arrangements (*a*) and (*b*) are sometimes used in residential designs.

A check valve should be provided in the runout from each return riser to prevent temporary reversal of flow in the piping when a faucet is open. Proper air venting of a circulating system is extremely important, particularly if gravity circulation is employed. In Fig. 12 (*a*) and (*b*), this is accomplished by connecting the circulating line below the top fixture supply. With this arrangement, air is eliminated from the system each time the top fixture is opened.

Where an overhead supply main is located above the highest fixture as in Fig. 12 (*c*), an automatic float-type air vent is installed at the highest point of the system, or a fixture branch is connected to the top of the main where air venting is desired, and then dropped to the fixture outlet.

It is sometimes necessary to make an allowance for pressure drop through the heater when sizing hot water lines, particularly where instantaneous hot water heaters are used and the available pressure is low.

The principles involved in the sizing of the hot water supply pipes are the same as those for the sizing of cold water supply lines. For small and medium sized installations a ¾-in. hot water return will be ample. For larger installations, the size of the hot water return may be computed from considerations of the heat losses in the hot water piping.[4] A throttling valve should be placed in the hot water return pipe so that the rate of circulation may be adjusted.

Where the hot water piping system is exceedingly long, a water circulator is frequently installed. It is controlled by an immersion thermostat (in the return line) set to start and stop the pump over approximately a 20 F deg temperature range. In calculating the pump capacity, the product of (1) the pounds of water circulated and (2) the temperature difference (usually 20 deg) between the flow and return, must be equivalent to the Btu heat loss from the piping at the average temperature of the water. Insulation reduces the heat loss and pump capacity required. The friction head against which the pump must operate may be calculated from the tables and charts in Chapter 28, Hot Water Heating Systems. The friction head seldom exceeds 6 to 10 feet.

Since service water is corrosive due to the high oxygen content and high temperature, circulating pumps should be made of bronze.

CONTROL OF SERVICE WATER TEMPERATURE

Coal-fired boilers are usually controlled by an immersion thermostat (located in the heated water) which opens or closes draft dampers at the boiler to adjust the rate of fuel combustion. With oil- or gas-fired boilers, the immersion thermostat controls the oil burner or the automatic gas valve. The gas pilot flame usually burns continuously. With electric heaters, the immersion thermostat operates a switch on the source of energy.

When steam or hot water is the medium for heating the water in the tank, an immersion thermostat is used to control a valve in the steam or hot water supply line. In small residence installations, using water as the carrier, a combined immersion thermostat and butterfly valve in one simple fitting may be installed in the transmitting circuit to prevent overheating of the service water.

In residences heated by pump-circulated hot water, the house temperature is controlled by operating the circulating pump intermittently, while domestic hot water is warmed by transfer from the house boiler, independent of the pump operation. The domestic water is heated from the heating boiler the year-round. Under such an arrangement, to prevent overheating the house by thermal circulation when the pump is not running, it is usual to insert a weighted check valve in the house heating main so that no circulation to the house heating system can occur unless the pump operates. In summer the fire may be controlled to maintain a boiler water temperature lower than when heating, and generally about 20 F warmer than that desired in the domestic hot water system.

The immersion thermostat in a hot water storage tank should be located no higher than the center of the tank, and possibly should be even closer to the bottom, since water in a tank stratifies proportionally to the temperature. When hot water is removed, the cold water entering to replace it quickly reduces the temperature in the lower parts of the tank.

For general information on automatic control see Chapter 43.

SAFETY DEVICES FOR HOT WATER SUPPLY SYSTEMS

There are still numerous plumbing codes which do not have regulations for the prevention of hot water storage tank explosions.

An ordinary storage tank is under certain water pressure, depending on the static pressure in the system. When the water in the tank is heated by circulation through an external heater, or by heating units in the tank, it gradually expands. For instance, if the entire contents of a 30-gal storage tank are heated from 70 F to 160 F, there will be an increase in volume of about one-half gallon. If the tank is connected to a supply line without any intervening check valve, the increase in volume causes part of the water to be pushed back into the supply line. If the hot water reaches the water meter, it may ruin the composition discs.

If back-flow cannot occur, as for example, due to the use of a check valve or a pressure-reduction valve in the line, or because of temporary shut-off of the cold water line, the pressure in the tank rises as heating continues. Such a pressure rise, if the heating continues for any length of time, may result in rupturing of the tank.

If the tank water is not confined, continued heating would, of course, cause no increase in pressure beyond ordinary line pressure. However, the temperature would continue to increase. If this should happen at elevated temperatures, the flashing of the water into steam might cause a serious tank explosion.

In order to guard against the development of excessive pressure inside a hot water storage tank (owing to the thermal expansion of the water), it is customary to install a spring-loaded pressure-relief valve which is set to open at a pressure about 20 psi higher than the normal line pressure.

The amount of water discharged by any pressure-relief device is usually quite small, since it takes only a small quantity of water to relieve any pressure rise due to the thermal expansion of the water. The rate of discharge should be such as to limit the pressure rise for any given heat input to 10 percent of the pressure at which the valve is set to open. For any given installation, the discharging capacity *should be in excess of the water to be discharged by the heater. The heater discharge* Q_m *in gallons per minute may* be computed by Equation 3:

$$Q_m = 0.00005b \quad (3)$$

where b is the heat input of heater in Btu per minute.

To prevent danger of overheating, temperature-relief devices should be used. One type of such a device has a fusible plug of metal which melts at about 212 F. The hot water then runs from the opening until the device is serviced. A better type of relief device is one which incorporates a bellows or bimetallic disc, which opens at a temperature of 210 F and closes when the temperature drops to approximately 160 F.

Another type of safety device is a fuel cut-off switch or valve in which a fusible plug is melted by excessive water temperature. The fuel supply is cut off until the fusible plug is replaced.

The capacity of temperature-relief devices may be calculated by Equation 4:

$$Q_m = 0.0008b \quad (4)$$

where Q_m and b denote the same quantities as in Equation 3.

Pressure-relief valves should be installed in the cold water line near the tank out of contact with the hot water in order to prevent excessive corrosion and lime deposit on the valve seat. *Temperature-relief* valves, in general, must be installed at the point of maximum water temperature.

SOLAR WATER HEATERS

Solar heaters utilize the energy of the sun for heating water. The successful operation of such heaters requires the availability of sunshine practically every day in the year, which has limited their use to Florida and the southern portions of California. When supplemented with some other means of gas, coal, or oil water heating, solar heaters may be used in climates where sunshine may be more or less intermittent. They have been used in summer homes as far north as Chicago. When properly installed and proportioned, solar water heaters render satisfactory service, especially in climates where the outdoor temperatures are high and extremely hot water is not necessarily desirable. Such installations consist essentially of a storage tank, heating coil, and hot box. The coil is installed in the hot box, and is arranged to circulate water to and from the storage tank. The advantage in the use of this type of heater is the fact that it requires no fuel. The same materials should be used for the coil, circulation lines, and tank. A copper coil is more efficient in absorbing heat in the box, but galvanized iron or steel may be substituted, depending on the local water conditions, cost, and other considerations.

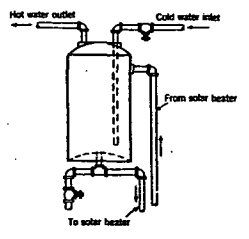


Fig. 13.... Solar Heater Tank Connections

The storage tank must be able to store sufficient heated water for the night period of about 16 hr when the coil is not functioning, or is operating under such poor sun conditions as to make its heating effect negligible. Due to the fact that the *no sun* period includes the night period when little or no hot water is used, an available storage of 50 percent of the average daily usage is considered adequate. Since about 25 percent of stored hot water cannot be drawn out of a storage tank before the incoming cold water reduces the temperature of all of the water in the tank to an unsatisfactory point for usage, the equation for calculating the storage capacity of the tank becomes:

$$S = \frac{Q_d \times 0.50}{0.75} = 0.666Q_d \quad (5)$$

where

S = storage capacity of tank, gallons.

Q_d = average daily usage, gallons.

Thus, for a family of four persons using an average of 40 gal of hot water per (person) (day), the size of the tank would be 4 persons x 40 gal x 0.666 or 106 gal, and the nearest standard size of tank would be used. The tank should be well insulated to prevent undue loss of heat during the 16-hour period when the coil is inoperative, and it should be located as high as possible in the building (under the peak of the roof if such exists) so as to secure a maximum circulation head from the coil. The hot water supply line to the house, as shown in Fig. 13, is connected to the top of the tank and serves to vent the air from the tank through the hot water faucets as fast as it accumulates.

Table 13.... Suggested Solar Heater Design Data[a]

Design Item	Based on Rate of 30 Gal per Day per Person								Based on Rate of 40 Gal per Day per Person							
No. of occupants in residence	1	2	3	4	5	6	7	8	1	2	3	4	5	6	7	8
Hot water used at night, gal per person	15	15	15	15	15	15	15	15	20	20	20	20	20	20	20	20
Hot water used at night, gal total	15	30	45	60	75	90	105	120	20	40	60	80	100	120	140	160
Retained in tank, 25 percent, gal	4	8	11	15	19	23	27	30	5	10	15	20	25	30	35	40
Tank capacity required, gal	20	40	59	75	94	113	130	150	25	50	75	100	125	150	175	200
Hot water used during day, gal	15	30	45	60	75	90	105	120	20	40	60	80	100	120	140	160
Total water to be heated:																
Gal per 8-hr period	35	70	104	135	169	203	235	270	45	90	135	180	225	270	315	360
Gal per hour	4.5	9	13	17	21	26	29	34	6	12	17	23	28	34	39	45
Copper coil required:																
Surface area, sq ft	25	50	75	100	121	145	168	192	32	64	96	128	160	192	224	256
Equivalent length 1-in. coil, ft	100	200	300	400	484	580	664	768	128	256	384	512	640	768	896	1024
Box size:																
Area, sq ft	25	50	75	100	121	145	168	192	32	64	96	128	160	192	224	256
Width, ft	4	6	7	8	9	10	10	11	4	6	8	9	10	11	12	12
Length, ft	6	8	11	12.5	13.5	14.5	16.5	17.5	8	10	12	14	16	18	19	21

[a] Sun Effect and the Design of Solar Heaters, by H. L. Alt (ASHVE Transactions, Vol. 41, 1935, p. 131).

Thus, for a family of four persons using an average of 40 gal of hot water per (person) (day), the size of the tank would be 4 persons x 40 gal x 0.666 or 106 gal, and the nearest standard size of tank would be used. The tank should be well insulated to prevent undue loss of heat during the 16-hour period when the coil is inoperative, and it should be located as high as possible in the building (under the peak of the roof if such exists) so as to secure a maximum circulation head from the coil. The hot water supply line to the house, as shown in Fig. 13, is connected to the top of the tank and serves to vent the air from the tank through the hot water faucets as fast as it accumulates.

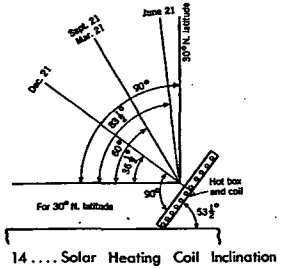


Fig. 14.... Solar Heating Coil Inclination

The coil should be of the return-bend type (square or slightly rectangular in form), and should have the pipes running east and west, with the coil on the south side of the building where it can receive the full sun effect all day long without shadows from the building itself, or from the adjacent obstructions such as trees or other structures. The coil should be placed as low as possible in relation to the storage tank level, such as on a porch roof, the roof of a one-story extension or, if necessary, even on the ground. Both the coil and the circulation lines should be designed to facilitate the circulation flow as much as possible, using long-radius copper fittings or recessed galvanized-iron fittings to match the materials of the coil, circulation lines, and tank. The coil should be inclined, as shown in Fig. 14, so that the north end is raised above the south end to secure an angle with the horizontal of about 53 deg. This will result in the inlet end of the coil being on the south side (or bottom), and the outlet end being on the north side (or top). This will satisfy conditions along the 30-deg N latitude, which includes the portions of Florida and Southern California where these heaters are most frequently used.

The hot box is usually constructed of wood on the four sides and bottom, and is insulated. Glass sash are placed over the top of the box which should be airtight. The interior surfaces should be painted white to reflect the heat, while the coil should be painted black to absorb the heat. The box need not be deeper than necessary to house the coil and to protect it from the weather.

The addition, on the bottom of the box, of a light gage copper plate to which the pipe of the coil is soldered for good metallic contact, will add to the amount of heat received by the coil, due to the fact that this plate will receive all of the sun's rays which fail to directly strike the coil. The heat from this source is transmitted to the coil through the plate rather than from the heated air surrounding the coil. Otherwise, only part of the heat enters the coil, the balance being transmitted through the glass.

Design data given in Table 13 may be used with judgment in selecting the size of solar heater coil and box for a particular application. These data are based on consumptions of 30 and 40 gal of hot water per (day) (person).

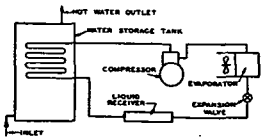


Fig. 15 .... Heat Pump Arrangement for Hot Water Supply

DOMESTIC HOT WATER BY HEAT PUMP

Hot water may suitably be obtained by using a heat-pump installation. The hot water heater may be either a heat exchanger installed just ahead of the compressor of a heat-pump installation, or may be a self-contained domestic-water heat pump.

Various designs of self-contained domestic water heat pumps are available, and one particular arrangement is shown in Fig. 15.

Hot water heating by means of a heat pump is not yet advisable where high water temperatures are desired. This is due to the fact that higher water outlet temperatures result in lower coefficients of performance.

For coefficients of performance of 4 or higher, the heat-pump water heater may be more economical to operate than a conventional water heater.

Although the first cost of domestic hot water heat pumps is somewhat high, they have the advantages of operating without products of combustion, odors, soot, or chimney. A further advantage is that they may be used for cooling purposes. With a coefficient of performance of 2½ to 3, water temperatures of 140 to 150 F may be obtained.[6]

REFERENCES

[1] R. B. Hunter: *Water Distributing Systems for Buildings* (*National Bureau of Standards Report* BMS79, p. 6). (Charts extended to flow of 0.30 gpm.)

[2] Private communication from H. E. Degler.

[3] J. S. Setchell: *Enough Hot Water—Hot Enough* (American Gas Association, 1950).

[4] Svend Plum: *Plumbing Practice and Design* (John Wiley and Sons, New York, 1943).

[5] P. Sporn and E. R. Ambrose: Progress report on a heat pump water heater (*Heating and Ventilating*, Vol. 46, February 1949, p. 78).

BIBLIOGRAPHY

H. C. Russell: Laundry, kitchen and hospital equipment (ASHVE TRANSACTIONS, Vol. 35, 1929, p. 45).

G. C. St. Laurent: Water consumption, cost and savings (*Hotel Engineering*, Vol. 1, American Hotel Association, 1940).

F. M. Dawson and A. A. Kalinske: *Water-Supply Piping for the Plumbing System* (*National Association of Master Plumbers Technical Bulletin* No. 3).

F. A. Brooks: *Use of Solar Energy for Heating Water* (Smithsonian Institution, Washington, D.C.).

R. B. Hunter: *Methods of Estimating Loads in Plumbing Systems* (*National Bureau of Standards Report* BMS65, 1940).

Plumbing Manual, Report of the Subcommittee on Plumbing, Central Housing Committee on Research, Design and Construction (*National Bureau of Standards Report* BMS66, 1940).

R. B. Hunter: *Water-Distributing Systems for Buildings* (*National Bureau of Standards Report* BMS79, 1941).

M. B. Mackay: Hot water requirements (*Modern Sanitation*, Vol. 1, August 1949, p. 30).

M. A. Pond: Urban domestic water consumption (*Journal of the American Water Works Association*, Vol. 31, No. 12, 1939, p. 2003).

R. Murray: How to size and install gas water heaters correctly (*Air Conditioning, Heating and Ventilating*, February 1955, p. 86).

J. C. Church: Water supply for tall buildings (*Air Conditioning, Heating and Ventilating*, February 1955, p. 99).

F. M. Reiter: Service hot water design for multi-story buildings (*Air Conditioning, Heating and Ventilating*, December 1955, p. 79).

F. M. Reiter: Service hot water for commercial and industrial use (*Air Conditioning, Heating and Ventilating*, February 1956, p. 89).

F. M. Reiter: Design data for service hot water (*Air Conditioning, Heating and Ventilating*, April 1956, p. 81).

G. R. Jerus: Design of swimming pools (*Air Conditioning, Heating and Ventilating*, February 1957, p. 113).

J. Nachbar: Water supply for industrial plants (*Air Conditioning, Heating and Ventilating*, December 1957, p. 53).

CHAPTER 57

CODES AND STANDARDS

THE Codes and Standards listed in Table 1 represent accepted practice, methods, or standards prepared and accepted by the organizations indicated. They are valuable guides for the practicing engineer in determining test methods, ratings, performance requirements, and limits applying to equipment used in heating, ventilating, and air conditioning. Copies can usually be obtained *from the organization listed in the reference column.*

Table 1 .... Codes and Standards Prepared and Accepted by Various Societies and Associations

Subject	Title	Sponsor	Reference
Acoustics (Terminology)	American Standard Acoustical Terminology (1951).	AS of A	ASA Z24.1
Air Conditioners	ASRE Standard Methods of Rating and Testing Air Conditioners (1956).	ASRE	ASRE Standard 16-56
Air Conditioners (Room)	Standards for Room Air-Conditioners.	ARI	ARI Standard 110-58
Air Conditioning	Code of Minimum Requirements for Comfort Air Conditioning (1938).	ASHVE ASRE	ASHAE
Air Conditioning (120,000 Btu/hr or less)	Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems (1953).	NWAH & ACA	NWAH & ACA Manual No. 7 4th Edition
Air Conditioning (Above 120,000 Btu/hr)	Code and Manual for the Design and Installation of Mechanical Warm Air Heating Systems (1950).	NWAH & ACA	NWAH & ACA Manual No. 9 5th Edition
Air Conditioning	Standards for the Installation of Air Conditioning and Ventilating Systems of Other Than Residence Type (1955).	NFPA	NBFU or NFPA No. 90A
Air Conditioning	Standards for the Installation of Residence Type Warm Air Heating and Air Conditioning Systems (1956).	NFPA	NBFU or NFPA No. 90B
Air Conditioning (Residential)	Design and Installation of Summer Air Conditioning for New and Existing Residences, Tentative (1955).	NWAH & ACA	NWAH & ACA Manual No. 11
Air Conditioning (Unitary Equipment)	Standard for Unitary Air Conditioning Equipment.	ARI	ARI Standard 210-57
Air Conditioning (Year-Round Residential)	Standard for Year-Round Residential Air-Conditioning.	ARI	ARI Standard 230-57
Airplane	Air Conditioning Equipment, Airplane—General Requirements for (1948).	SAE	SAE ARP 85B
Airplane	Heaters, Airplane, Internal Combustion Heat Exchanger Type (1949).	SAE	SAE ARP 143B
Airplane	Heaters, Airplane, Steam Type.	SAE	SAE ARP 87
Attic Ventilation	Residence Ventilation Guide (1950).	PFMA	PFMA
Boilers	I=B=R Testing and Rating Code for Low Pressure Cast Iron Heating Boilers, 6th Edition, 1958.	IBR	IBR
Boilers	Net Load Recommendations for Heating Boilers. Publ. semi-annually.	HP & ACCNA	MCAA
Boilers	Net Square Feet Radiation Loads in 70 Deg Fahr, Recommended for Low Pressure Heating Boilers (1948).	HP & ACCNA	MCAA
Boilers	ASME Boiler and Pressure Vessel Code (1956, 8 Sections).	ASME	ASME
Boilers	ASME Boiler and Pressure Vessel Code, Section IV, Low Pressure Heating Boilers (1956 with 1957 and 1958 addenda).	ASME	ASME
Boilers (Gas)	American Standard Approval Requirements for Central Heating Gas Appliances Vol. 1, Steam and Hot Water Boilers (1956).	A.G.A.	ASA Z21.13.1-1956
Boilers (Miniature)	ASME Boiler and Pressure Vessel Code, Section V, Miniature Boilers (1952).	ASME	ASME
Boilers (Power)	ASME Boiler and Pressure Vessel Code, Section I, Power Boilers (1956).	ASME	ASME
Boilers (Power)	ASME Boiler and Pressure Vessel Code, Section VII, Suggested Rules for Care of Power Boilers (1954).	ASME	ASME
Boilers (Steel)	Steel Boiler Institute Rating Code for Steel Boilers (1958).	SBI	SBI 7th Edition
Boilers (Steel)	Simplified Practice Recommendation for Steel Firebox Boilers and Steel Heating Boilers (Commercial and Residential Types) (1950).	BS SBI	CSD R157-50

Table 1 .... Codes and Standards—(Continued)

Subject	Title	Sponsor	Reference
Buildings	National Building Code (1955 Edition).	NBFU	NBFU
Building Requirements	American Standard Building Requirements (1946).	NHA USPHS	ASA A53.1-1946
Buildings	Basic Building Code (Also published in abridged form as Abridged Building Code) 1955.	BOCA	BOCA
Burners (Anthracite)	Commercial Standard for Domestic Burners for Pennsylvania Anthracite (Underfeed Type) (1940).	BS AIL	CSD CS48-40
Burners (Gas)	American Standard Requirements for Installation of Domestic Gas Conversion Burners (1948). Reaffirmed 1952.	A.G.A.	ASA Z21.8-1948
Burners (Gas)	American Standard Requirements for Installation of Gas Burning Equipment in Large Boilers (1950). Reaffirmed 1956.	A.G.A.	ASA Z21.33-1950
Burners (Gas)	American Standard Listing Requirements for Domestic Gas Conversion Burners (1948). Reaffirmed 1952.	A.G.A.	ASA Z21.17-1948
Burners (Oil)	Commercial Standard for Mechanical-Draft Oil Burners Designed for Domestic Installations (1956).	BS OBI	CSD CS75-56
Burners (Oil)	Standards for the Installation of Oil Burning Equipment (1956).	NFPA	NBFU or NFPA No. 31
Burners (Oil)	Standard for the Installation of Residential Oil-Fired Central Heating Equipment (1958).	OHIA	OHIA B-58
Burners (Pulverized Coal)	Standards for the Installation of Pulverized Coal Systems (1956).	NFPA	NBFU or NFPA No. 60A
Chillers	Liquid-Chilling Packages with Reciprocating Compressors (2HP and Larger) (1956).	ARI	ARI Standard 590-56
Chimneys (Flue Linings)	American Standard Sizes of Clay Flue Linings (1947).	AIA PC	ASA A62.4-1947
Cleaners (Air)	Code for Testing Air Cleaning Devices Used in General Ventilation: Section I, Unit or Panel Type Air Filtering Devices (1953).	AFI	AFI
Coal (For Stokers)	Tentative Standard Procedure for Testing and Evaluating Bituminous Stoker Coals (1952).	SMA BCR	SMA BCR
Coils	Proposed Commercial Standard for Rating and Testing Air Cooling Coils Using Nonvolatile Refrigerants (1945).	BC BS	CSD TS 4044
Coils	Joint ASRE-ASHAE Standard Methods of Testing and Rating Forced-Circulation Air Heating and Air Cooling Coils.	ASHAE ASRE	ASRE Circular No. 33-58
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1956).	HP & ACCNA	MCAA Engrg. Stds. Sec. 2, Part V
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1956).	ASME	ASA A13.1-1956
Compressors	ASRE Standard Methods of Rating and Testing Refrigerant Compressors.	ASRE ASHVE ACRMA	ASRE Standard 23-R
Compressors (Reciprocating)	Standard Nomenclature for Reciprocating Compressors.	ARI	ARI Standard 510-57
Condensers	ASRE Standard Methods of Rating and Testing Evaporative Condensers.	ASRE ASHVE ACRMA	ASRE Standard 20
Condensers	ASRE Standard Methods of Rating and Testing Water-Cooled Refrigerant Condensers.	ASRE ASHVE ACRMA	ASRE Standard 22
Condensing Units	ASRE Standard Methods of Rating and Testing Mechanical *Condensing Units* (1940).	ASRE ASHVE ACRMA	ASRE Standard 14-41
Conductance	Method of Test for Thermal Conductance and Transmittance of Built-up Sections by Means of Guarded Hot Box (1949).	ASTM	ASTM C 236-49 T
Conductivity	Standard Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate (1945).	ASHVE ASRE ASTM NRC	ASTM C-177-45
Control Equipment (Industrial)	Underwriters' Laboratories, Inc., Standard for Industrial Control Equipment (July, 1938).	UL	UL Subject 508
Controls	Underwriters' Laboratories, Inc., Standard for Temperature Indicating and Regulating Equipment (June 1953).	UL	UL Subject 873
Controls Temperature	NEMA Standards for Automatic Temperature Controls (1953).	NEMA	NEMA DC1-1953
Controls Temperature	Temperature Control Equipment, Automatic, Airplane Cabin	SAE	SAE ARP 89A
Convectors	*Commercial Standard for Testing and Rating Convectors* (1947).	BS CMA IBR	CSD CS 140-47

Table 1 .... Codes and Standards—(Continued)

Subject	Title	Sponsor	Reference
Coolers (Air)	ASRE Standard Methods of Rating and Testing Forced Circulation and Natural Convection Air Coolers for Refrigeration (1945).	ASRE ASHVE ACRMA REMA	ASRE Standard 25-44
Coolers (Air)	Forced-Circulation Free-Delivery Air Coolers for Refrigeration.	ARI	ARI Standard 420-57
Coolers	ASRE Standard Methods of Rating and Testing Liquid Coolers.	ASRE	ASRE Standard 24-57
Cooling Towers	Accepted Test Procedure for Water Cooling Towers, Mechanical Draft, Industrial Type (1955).	CTI	CTI Bulletin ATP-105
Cooling Towers (Mechanical Draft)	Method of Testing for Rating Mechanical Draft Cooling Towers Under Controlled Conditions.	ASRE	ASRE Standard 38-57
Cooling Towers	Standard for Mechanical Draft Cooling Towers	ARI	ARI Standard 910-58
Ducts and Fittings	Simplified Practice Recommendation for Pipes, Ducts and Fittings for Warm Air Heating and Air Conditioning (1945).	Mfrs. BS	CSD R207-49
Exchangers (Heat)	Standards of *Tubular Exchanger Manufacturers Association* (1941).	TEMA	TEMA
Exhaust Systems	American Standard for Grinding, Polishing, and Buffing Equipment Sanitation (1941).	AFA	ASA Z43-1941
Exhaust Systems	Tentative Code of Recommended Practices for Testing and Measuring Air Flow in Exhaust Systems (1937).	AFA	AFA Preprint 36-27
Exhaust Systems	Tentative Recommended Good Practice Code and Handbook on the Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems.	AFA	AFA
Exhaust Systems	Standards for Blower and Exhaust Systems (1949).	NFPA	NFPA or NBFU No. 91
Exhaust Systems (Open Tanks)	American Standard Safety Code for Ventilation and Operation of Open-Surface Tanks (1951).	AIHA ASHVE NAFM	ASA Z9.1–1951
Fans	Definitions and Terms in Use by the Blower Industry (1950) (Was NAFM Bulletin No. 105).	NAFM	NAFM-AMCA Bulletin No. 110
Fans	Standard Test Code for Testing Centrifugal and Axial Fans (1950) (Was NAFM Bulletin No. 103).	NAFM* ASHVE	NAFM-AMCA Bulletin No. 110
Fans	Standard Code for Testing Centrifugal, Axial, and Propeller Fans. (Having Wheels Less than 12-in. Dia). 1953.	NAFM	NAFM-AMCA Bulletin No. 116
Fans	Standards, Definitions, Terms and Test Codes for Centrifugal, Axial and Propeller Fans.	NAFM ASHVE**	NAFM-AMCA Bulletin 110 2nd Edition, 1952
Fans	Standards Publication Electric Fans (1955).	NEMA	NEMA FM1-1955
Fans	Test Code for Fans (1946).	ASME	ASME PTC 11-1946
Fans	Testing and Rating Ventilating Fans (Axial and Propeller Type).	CSD	CSD
Fire Prevention	National Fire Codes (1954). Vol. I—Flammable Liquids and Gases, Vol. II—Combustible Solids, Dusts, Chemicals, and Explosives, Vol. III—Building Construction and Equipment, Vol. IV—Extinguishing Equipment, Vol. V—Electrical, Vol. VI—Transportation.	NFPA	NFPA
Furnaces (Duct)	American Standard Approval Requirements for Gas-Fired Duct Furnaces (with 1956 and 1957 Addenda).	A.G.A.	ASA Z21.34-1955
Furnaces (Gas)	American Standard Approval Requirements for Central Heating Gas Appliances Vol. II, Gravity and Forced Air Central Furnaces (1956).	A.G.A.	ASA Z21.13.2 1956
Furnaces (Forced Air, Solid-Fuel)	Commercial Standard for Solid-Fuel-Burning Forced Air Furnaces (1944).	FHA NWAH & ACA AIL	CSD CS109-44
Furnaces (Heavy Duty)	Code for Testing and Rating Heavy Duty Furnaces and Direct-Fired Unit Heaters (1955).	ASHAE	ASHAE
Furnaces (Oil-Fired)	Commercial Standard for Warm Air Furnaces Equipped with Vaporising Type Oil Burners (1949).	Mfrs. BS	CSD CS104-49
Furnaces (Oil-Fired)	Recommended Commercial Standard for Warm Air Furnaces Equipped with Pressure Atomizing or Rotary Type Oil Burners.	BS NWAH & ACA	CSD CS195-54
Furnaces (Oil)	Commercial Standard for Oil Burning Floor Furnaces Equipped with Vaporizing Type Burners (1951).	BS OPA	CSD CS113-51

* Also endorsed by PFMA.

** Refers to Test Code for Centrifugal and Axial Fans.

Table 1 .... Codes and Standards—(Continued)

Subject	Title	Sponsor	Reference
Furnaces (Oil)	A Tentative Code for Testing Oil-Fired Furnaces.	NWAH & ACA	NWAH & ACA
Garage Ventilation	Recommended Good Practice Requirements for the Construction and Protection of Garages (1932, Reprint 1952)	NFPA	NFPA Pamphlet No. 88
Garages	Code of Minimum Requirements for Heating and Ventilating Garages (1935).	ASHVE	ASHAE
Gas Equipment (Large Boilers)	American Standard Requirements for Installation of Gas Equipment in Large Boilers (1950). Reaffirmed 1956.	A.G.A.	ASA Z21.33-1950
Gases (Toxic) and Dust	American Standard Allowable Concentration of Harmful Gases:	ASA	ASA
	Carbon Monoxide		Z37.1-1941
	Hydrogen Sulfide		Z37.2-1941
	Carbon Disulfide		Z37.3-1941
	Benzene		Z37.4-1941
	Manganese		Z37.6-1948
	Chromic Acid and Chromates		Z37.7-1943
	Mercury		Z37.8-1943
	Xylene		Z37.10-1948
	Lead and Certain Inorganic Lead Compounds		Z37.11-1943
	Toluene		Z37.12-1943
	Oxides of Nitrogen		Z37.13-1944
	Methanol		Z37.14-1944
	Styrene Monomer		Z37.15-1944
	Formaldehyde		Z37.16-1944
	Carbon Tetrachloride		Z37.17-1957
	Methyl Chloride		Z37.18-1949
	Trichloroethylene		Z37.19-1946
Hangars Aircraft	Standard for the Construction and Protection of Aircraft Hangars (1954).	NFPA	NBFU or NFPA No. 409
Heat Pumps (Unitary Equipment)	Standard for Unitary Heat Pump Equipment.	ARI	ARI Standard 510-57
Heat Transfer (Walls)	Thermal Conductance and Transmittance of Built-up Sections by Means of Guarded Hot Box (1949).	ASTM	ASTM C-236-49 T
Heaters (Recessed Gas Fired)	American Standard Approval Requirements for Central Heating Gas Appliances, Vol. IV Gravity and Fan Type Vented Recessed Heaters (1955 with 1956 and 1957 Addenda).	A.G.A.	ASA Z21.13.4-1955
Heaters (Room Gas Fired)	American Standard Approval Requirements for Gas-Fired Room Heaters (formerly called Space Heaters) (1956).	A.G.A.	ASA Z21.11-1956
Homes (Prefabricated)	Commercial Standard for Prefabricated Homes	PHMI BS	CSD CS125-47
Mineral Wool	Commercial Standard for Mineral Wool Insulation for Heated Industrial Equipment (1949).	BS IMWI	CSD CS117-49
Mineral Wool	Commercial Standard for Mineral Wool Insulation for Low Temperature Installations (1948).	BS IMWI	CSD CS105-48
Mineral Wool	Recommended Commercial Standard for Industrial Mineral Wool Products—All Types—Testing and Reporting (1946).	IMWI BS	CSD CS131-46
Motors	Nema Motor and Generator Standards (June 1945).	NEMA	NEMA 45-102
Motors	Test Code for Single-Phase Motors (1941).	AIEE	AIEE Report 502
Panel System (Warm Air)	Code and Manual for the Design and Installation of Warm Air Ceiling Panel Systems (1950).	NWAH & ACA	NWAH & ACA Manual No. 7-A 3rd Edition
Perimeter (Warm Air Small Pipe)	4-Inch Pipe Warm Air Perimeter Heating (1956).	NWAH & ACA	NWAH & ACA Manual No. 10 4th Edition
Perimeter (Warm Air)	Warm Air Perimeter Heating (1956, 5th Edition).	NWAH & ACA	NWAH & ACA Manual No. 4
Pipe & Tubing (Copper & Brass)	Simplified Practice Recommendation for Copper Water Tubes and Brass Pipe.	Mfrs. BS	CSD R217-49
Piping	American Standard Code for Pressure Piping (1955).	ASME	ASA B31.1-1955
Piping (Gas)	American Standard for Installation of Gas Piping and Gas Appliances in Buildings (1954).	A.G.A.	ASA Z21.30-1954
Pumps	Hydraulic Institute Test Code for Centrifugal Pumps. Hydraulic Institute Test Code for Rotary Pumps (1943).	HI	HI Section F
Radiation (Baseboard)	I=B=R Testing and Rating Code for Baseboard Type of Radiation (1950).	IBR	IBR

Table 1 .... Codes and Standards—(Continued)

Subject	Title	Sponsor	Reference
Radiation (Finned-Type)	I=B=R Testing and Rating Code for Finned-Type Radiation (First Edition 1951, with Addenda 1954).	IBR	IBR
Radiators	Simplified Practice Recommendation for Cast Iron Radiators (1943).	IBR BS	CSD R174-47
Refrigerants	Standard Designation of Refrigerants.	ASRE	ASRE Standard 34-57
Refrigeration (Equipment)	Underwriters' Laboratories, Inc., Standard for Air Conditioning and Commercial Refrigerating Equipment (Feb. 1946).	UL	UL Subject 207A
Refrigeration (Mechanical)	American Standard Safety Code for Mechanical Refrigeration (1958).	ASRE	ASA B9.1-1958***
Refrigeration (Unit Systems)	Underwriters' Laboratories, Inc., Standard for Unit Refrigerating Systems (Feb. 1946).	UL	UL Subject 207C
Refrigerators (Gas-Fired)	American Standard Approval Requirements for Refrigerators Using Gas Fuel (1942, Reaffirmed 1950).	A.G.A.	ASA Z21.19-1941
Refrigerators (Household)	American Standard Test Procedures for Household Electric Refrigerators (Mechanically Operated) (1944). Reaffirmed 1953.	ASRE USDA	ASA B38.2-1944
Sound (Measurement)	American Standard for Sound Level Meters for Measurement of Noise and Other Sounds.	AS of A	ASA Z24.3-1944
Sound (Measurement)	American Standard Method for the Pressure Calibration of Laboratory Standard Pressure Microphones.	AS of A	ASA Z24.4-1949
Sound (Measurement)	Sound Measurement Test Code for Centrifugal and Axial Fans (1950) (Was NAFM Bulletin No. 104).	NAFM	NAFM-AMCA Bulletin No. 110, 1950
Sound (Measurement)	Sound Measurement Test Code for Unit Heaters (1955).	IUHA	IUHA-AMCA Bulletin No. 13
Space Heaters	Commercial Standard for Flue Connected Oil-Burning Space Heaters Equipped with Vaporizing Pot-Type Burners (1943).	ICHAM	CSD CS101-43
Stokers	Code for Determination of Rated Capacities of Anthracite *Underfeed Stokers (1944).*	SMA	SMA
Stokers	Code for Determination of Rated Capacities of Bituminous Underfeed Stokers (1944).	SMA	SMA
Stokers	Recommended Minimum Firebox Dimensions and Base Heights (1944).	SMA	SMA
Stokers	*Recommended Standards Governing Minimum Setting* Heights (1944).	SMA	SMA
Tubing (Seamless Copper and Copper Alloy)	Simplified Practice Recommendation for Copper and Copper-Alloy Round Seamless Tube (1948).	Mfrs. BS	CSD R235-48
Tubing (Seamless Copper Water Tube)	Standard Specifications for Seamless Copper Water Tube (1956).	ASTM	ASA H23.1-1956
Unfired Pressure Vessels	ASME Boiler and Pressure Vessel Code Section VIII, Unfired Pressure Vessels (1956).	ASME	ASME
Unit Heaters	American Standard Approval Requirements for Gas Unit Heaters (1957).	A.G.A.	ASA Z21.16-1957
Unit Heaters	Standard Code for Testing and Rating Steam Unit Heaters (1950).	ASHVE IUHA	IUHA-AMCA Bulletin 10
Unit Heaters	Standard Code for Testing Hot Water Unit Heaters (1942).	IUHA	IUHA-AMCA Bulletin 11
Unit Ventilators	ASHVE Standard Code for Testing and Rating Steam Unit Ventilators (1934).	ASHVE	ASHAE
Vacuum Pumps	ASHVE Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps (1934).	ASHVE	ASHAE
Warm Air (Gravity)	Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems (1954).	NWAH & ACA	NWAH & ACA Manual No. 5 5th Edition
Water Heaters	American Standard Household Automatic Electric Storage-type Water Heaters.	NEMA	ASA C72.1-1949
Water Heaters	American Standard Approval Requirements for Gas Water Heaters Volume I (1956).	A.G.A.	ASA Z21.10.1-1956
Water Heaters	NEMA Standards for Electric Water Heaters (1945).	NEMA	NEMA 45-104
Water Heaters	Testing and Rating Hand-Fired Hot Water Supply Boilers (1948).	FHA	CSD CS145-47
Wiring	Interior Wiring Design for Commercial Buildings.	AIEE	AIEE
Wiring	National Electrical Code (1956).	NFPA	NFPA or NFBU No. 70

*** Also designated ASRE Standard No. 15-58.

ABBREVIATIONS AND ADDRESSES

The Codes and Standards listed in preceding pages of this table can be obtained from the organizations listed in the *Reference Column*.

ACMA	Air Conditioning Manufacturers Association, superseded 1940 by ACRMA.
ACRMA	Air Conditioning and Refrigerating Machinery Association, Southern Bldg., Washington, D. C. superseded 1953 by ARI.
AFA	American Foundrymen's Association, 616 S. Michigan Ave., Chicago, Ill.
AFI	Air Filter Institute, 3 Center Park, Rochester 3, N. Y.
A.G.A.	American Gas Association, 420 Lexington Ave., New York 17, N. Y.
AGAEM	Association of Gas Appliance and Equipment Manufacturers, superseded 1945 by GAMA.
AIA	American Institute of Architects, 1741 New York Ave., N. W., Washington 6, D. C.
AIEE	American Institute of Electrical Engineers, 33 West 39th St., New York 18, N. Y.
AIHA	American Industrial Hygiene Association, 14125 Prevost, Detroit 27, Mich.
AIL	Anthracite Industries Laboratory, 237 Old River Rd., Wilkes-Barre, Pa.
AMCA	Air Moving and Conditioning Association, 2159 Guardian Building, Detroit 26, Mich.
ARI	Air-Conditioning and Refrigeration Institute, 1346 Connecticut Ave., N. W. Washington, D. C.
ASA	American Standards Association, 70 East 45th St., New York 17, N. Y.
AS of A	Acoustical Society of America, 57 East 55th St., New York 22, N. Y.
ASHAE	American Society of Heating and Air-Conditioning Engineers, 62 Worth St., New York 13, N. Y.
ASHVE	American Society of Heating and Ventilating Engineers, name changed to American Society of Heating and Air-Conditioning Engineers, Dec. 8, 1954.
ASME	American Society of Mechanical Engineers, 29 West 39th St., New York 18, N. Y.
ASRE	American Society of Refrigerating Engineers, 234 Fifth Ave., New York 1, N. Y.
ASTM	American Society for Testing Materials, 1916 Race St., Philadelphia 3, Pa.
BCMI	Blast Coil Manufacturers Institute, superseded by Heating and Cooling Coil Manufacturers Association HCCMA.
BCR	Bituminous Coal Research, Inc., 121 Meyran Ave., Pittsburgh 13, Pa.
BOCA	Building Officials Conference of America, 110 East 42nd St., New York 17, N. Y.
BS	National Bureau of Standards, Washington, D. C.
CSD	Commodity Standards Division, U. S. Dept. of Commerce, Washington, D. C.
CMA	Convector Manufacturers Association, 2159 Guardian Bldg., Detroit 26, Mich.
CTI	Cooling Tower Institute, 444 Emerson St., Palo Alto, Calif.
FHA	Federal Housing Administration, Washington, D. C.
GAMA	Gas Appliance Manufacturers' Association, 60 East 42nd St., New York 17, N. Y.
HCCMA	Heating and Cooling Coil Manufacturers Association, superseded by Air Moving and Conditioning Association, AMCA.
HI	Hydraulic Institute, 122 East 42nd St., New York 17, N. Y.
HP & ACCNA	Heating, Piping and Air Conditioning Contractors National Association, superseded 1956 by Mechanical Contractors Association of America, MCAA.
IBR	Institute of Boiler and Radiator Manufacturers, 608 Fifth Ave., New York 20, N. Y.
ICHAM	Institute of Cooking and Heating Appliance Manufacturers, Shoreham Hotel, Washington, D. C.
IMFI	Industrial Mineral Fiber Institute, 441 Lexington Ave., New York 17, N. Y.
MWI	Industrial Mineral Wool Institute, superseded 1953 by IMFI.
IUHA	Industrial Unit Heater Association, superseded 1956 by Air Moving and Conditioning Association, AMCA.
MCAA	Mechanical Contractors Association of America, Suite 570, 45 Rockefeller Plaza, New York 20, N. Y.
NAFM	National Association of Fan Manufacturers, superseded 1956 by Air Moving and Conditioning Association, AMCA.
NBFU	National Board of Fire Underwriters, 85 John St., New York 33, N. Y.
NEMA	National Electrical Manufacturers Association, 155 East 44th St., New York 17, N. Y.
NFPA	National Fire Protection Association, 60 Batterymarch St., Boston 10, Mass.
NHA	National Housing Agency, Washington, D. C.
NRC	National Research Council, 2101 Constitution Ave., Washington, D. C.
NWAH & ACA	National Warm Air Heating and Air Conditioning Association, 640 Engineers Bldg., Cleveland 14, Ohio.
OBI	Oil Burner Institute, superseded 1942 by OHIA
OHIA	Oil Heat Institute of America, Room 1618 (Lower Level) 500 Fifth Ave., New York 36, N. Y.
OPA	Office of Price Administration, Washington, D. C.
PC	Producers Council, 2029 K St., N.W., Washington 6, D. C.
PFMA	Propeller Fan Manufacturers Association. Became Power Fan Manufacturers Association before merger with AMCA.
PHMI	Prefabricated Home Manufacturers Institute, 908 20th St., N.W., Washington 6, D. C.
REMA	Refrigeration Equipment Manufacturers Association, superseded 1953 by ARI.
RMA	Refrigerating Machinery Association. See ACRMA.
SAE	Society of Automotive Engineers, 29 West 39th St., New York 18, N. Y.
SBI	Steel Boiler Institute, 1308 Land Title Bldg., Philadelphia 10, Pa.
SMA	Stoker Manufacturers Association, 307 N. Michigan Ave., Chicago 1, Ill.
TEMA	Tubular Exchanger Manufacturers Association, 53 Park Place, New York, N. Y.
UL	Underwriters' Laboratories, 207 East Ohio St., Chicago 11, Ill.
USDA	United States Department of Agriculture, Washington, D. C.
USPHS	United States Public Health Service, Washington, D. C.

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Tjernlund Mfg. Co., •127
Trane Co., The, •128–139

FURNACES, Suspended
Airtemp Div., Chrysler Corp., •88–89
Airtherm Mfg. Co., •320
Chicago Steel Furnace Co., •117
Crane Co., •322–322
Dravo Corp., •118
General Automatic Products Corp., •310
General Electric Co., •83–85
Hayes Furnace Mfg. & Supply Co., •106–107
Jackson & Church Div., York-Shipley, Inc., •119
Johnson Heater Corp., •120–121
Lennox Industries, Inc., •109, 122
Majestic Co., Inc., The, •110
Mammoth Furnace Co., The, •123
National Heater Co., •124–125
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325–327
Peerless Heater Co., •330–331
Thermobloc Div., Frat-Daniel Corp., •126
Tjernlund Mfg. Co., •127
York-Shipley, Inc., •280

FURNACES, Wall
Dura-Vent Corp., Affiliate of Peerless Mfg. Div. of Dover Corp., •368
Weaz Electric Heater Co., •148

FURNACES, Warm Air, Heavy Duty
Air Devices, Inc., •155, 240–241
Airtherm Mfg. Co., •320
Aladdin Heating Corp., •199
American Foundry & Furnace Co., •104–105
E. K. Campbell Co., •116
Campbell Heating Co., •114–115
Carrier Corporation, •74–75, 130
Chicago Steel Furnace Co., •117
Dravo Corp., •118
Hayes Furnace Mfg. & Supply Co., •106–107
Jackson & Church Div., York-Shipley, Inc., •119
S. T. Johnson Co., •374–375
Johnson Heater Corp., •120–121
Lennox Industries, Inc., •109, 122
Majestic Co., Inc., The, •110
Mammoth Furnace Co., The, •123
National Heater Co., •124–125
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325–327
Herman Nelson Div. American Air Filter Co., Inc., •128–129
Petro, •378–379
Ray Burner Co., •377
Thermobloc Div., Frat-Daniel Corp., •126
Tjernlund Mfg. Co., •127

FURNACES, Warm Air, Residence
Airtemp Div., Chrysler Corp., •88–89
Aladdin Heating Corp., •199
Crane Co., •322–323
Dura-Vent Corp., Affiliate of Peerless Mfg. Div. of Dover Corp., •368
General Automatic Products Corp., •310
General Electric Co., •83–85
Hayes Furnace Mfg. & Supply Co., •106–107
Jackson & Church Div., York-Shipley Inc., •119
S. T. Johnson Co., •374–375
Johnson Heater Corp., •120–121
Lennox Industries, Inc., •109, 122
Majestic Co., Inc., The, •110
Mammoth Furnace Co., The, •123
National Heater Co., •124–125
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325–327
Peerless Heater Co., •330–331
Pennsylvania Furnace & Iron Co., •111, 333
Ray Burner Co., •377
Trane Co., The, •128–139
York Corp., Sub. Borg-Warner Corp., •96
York-Shipley, Inc., •280

GAGES, Altitude
Jas. P. Marsh Corp., •423

GAGES, Compound
Jas. P. Marsh Corp., •423

GAGES, Liquid Level
Henry Valve Co., •274
Mueller Brass Co., •424

GAGES, Pressure
General Controls, •273–273
Jas. P. Marsh Corp., •423

GAGES, Steam
Jas. P. Marsh Corp., •423

GAGES, Vacuum
Jas. P. Marsh Corp., •423
Moeller Instrument Co., •282

GAS BURNERS
American-Standard, Plumbing & Heating Div., •315–319
Babcock & Wilcox Co., The, •339
Dura-Vent Corp., Affiliate of Peerless Mfg. Div. of Dover Corp., •368
Gordon-Piatt, Inc., •369
Industrial Combustion, Inc., •373
Lennox Industries, Inc., •109, 122
Majestic Co., Inc., The, •110
Mammoth Furnace, The, •123
National Heater Co., •124–125
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325–327
Peerless Heater Co., •330–331
Thermobloc Div., Frat-Daniel Corp., •126
Tjernlund Mfg. Co., •127
York-Shipley, Inc., •280

GAS FILTERS (*See Filters, Gas*)

GAS SAFETY PILOTS (*See Pilots*)

GAS VENT PIPE (*See Pipe, Gas Vent*)

GAS VENTS
Amerivent, A Div. of American Metal Products Co., Inc., •365
Dura-Vent Corp., Affiliate of Peerless Mfg. Div. of Dover Corp., •368
Hart & Cooley Mfg. Co., •250–251
Metalbestos Div., William Wallace Co., •356–357
National Clay Pipe Mfgrs., Inc., •314–317

GASKETS, Cork
Armstrong Cork Co., Industrial Div., •444

GLASS BLOCK ROOFLIGHTS (*See Skylights Insulated*)

GLASS BLOCKS
Owens-Illinois, •441
Pittsburgh Corning Corp., Foamglas Div., •442

GLASS, Cellular
Pittsburgh Corning Corp., •442

GOVERNORS, Pump
Leslie Co., •278
Robertshaw-Fulton Controls Co., Fulton Sylphon Div., •285–287
Spence Engineering Co., Inc., •289
Warren Webster & Co., •416–417

GREASE FILTERS (*See Filters Grease*)

GRILLES, REGISTERS AND ORNAMENTAL METAL WORK (*See Louvers, Registers*)
A-J Mfg. Co., •344–345
Air Control Products, Inc., •238
Air Devices, Inc., •155, 240–241
Air Distribution Products, •239
Air-Factors, Inc., •243
Auer Register Co., The, •246
Barber-Colman Co., •247, 271
Carnes Corp., •248–249
Dole Valve Co., The, •420
General Electric Co., •82–83
Hart & Cooley Mfg. Co., •250–251
Hendrick Mfg. Co., •252–253
Independent Register Co., The, •255
Kritzer Products, A Div. of Peerless of America, Inc., •103
Krueger Air Conditioning Corp., •256
Lau Blower Co., •212–212
Pyle-National Co., The, Multivent Div., •257
Titus Mfg. Corp., •Insert Section
Tuttle & Bailey, •258–259
United States Register Co., •260–261
Universal Diffuser Corp., •264
Vulcan Radiator Co., The, •Insert Section
Waterloo Register Co., Inc., •265
Young Regulator Co., •262–263

HANGERS, Pipe
American Tube Products, Inc., •284

HANGERS, Radiator
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325–327

HEADS, Exhaust
V. D. Anderson Co., The Div. of International Basic Economy Corp., •404–405
Muckle Mfg. Co., •235

HEAT EXCHANGERS
Acme Industries, Inc., •64–65
Adsco Div., Yuba Consolidated Industries, Inc., •296
Aero Corp., •331
Aerofin Corp., •186–187
American Blower Div. of American-Standard, •66–69
Bell & Gossett Co., •383–385
Binks Mfg. Co., •178
Davis Engineering Corp., •385
Dole Refrigerating Co., •194
Dreyer-Hanson, Div. of National-U. S. Radiator Corp., •78
Dunham-Bush, Inc., •406–409
Ilg Air Conditioning Corp., •89
Fluor Products Co., A Div. of The Fluor Corp., Ltd., •179
General Fittings Co., •286
International Boiler Works Co., The, •349
Killebrew Engineering Corp., •387
Marlo Coil Co., •86
McQuay, Inc., •132–133
Mueller Brass Co., •424
National Heater Co., Inc., •124–125
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325–327
Patterson-Kelley Co., Inc., The, •191
Recold Corp., •90–91
Richmond Engineering Co., Inc., •388
Taco Heaters, Inc., •392
H. A. Thrush & Co., •390–391
Trane Co., The, •128–139
Western Blower Co., •224
Westinghouse Electric Corp., Sturtevant Div., •175, 223
Edwin L. Wiegand Co., •146–147
Worthington Corp., Air Cond'g & Refrigeration Dept., •95
York Corp., Sub. Borg-Warner Corp., •96
York-Shipley, Inc., •280
Yuba Corp., formerly Yuba Water Heaters, Inc., •393

HEAT PUMPS
Acme Industries, Inc., •64–65
Addison Products Co., •97
Lennox Industries, Inc., •109, 122
Majestic Co., Inc., The, •110
Trane Co., The, •128–139

HEAT RECLAIMERS
Richmond Engineering Co., Inc., •388

HEAT SURFACE
Aerofin Corp., •186–187
American Blower, Div. of American-Standard, •66–69
G & O Mfg. Co., The, •190
D. J. Murray Mfg. Co., •125
John J. Nesbitt, Inc., •142
New York Blower Co., •216
Niagara Blower Co., •87
Richmond Engineering Co., Inc., •388
Rittling Corp., The, •192
Warren Webster & Co., •416–417
Edwin L. Wiegand Co., •146–147
Young Radiator Co., •143

HEATERS, Air
Aerovent Fan Co., Inc., •197
Air Devices, Inc., •155, 240–241
American Foundry & Furnace Co., •104–105
Buffalo Forge Co., •202
E. K. Campbell Co., •116
Campbell Heating Co., •114–115
Chicago Steel Furnace Co., •117
Dravo Corp., •118
Electromode, Div. of Commercial Controls Corp., •114
Hayes Furnace Mfg. & Supply Co., •106–107
Industrial Engineering & Equipment Co., •145
Jackson & Church, Div. of York-Shipley, Inc., •119
Johnson Heater Corp., •120–121
Lennox Industries, Inc., •109, 122
Mammoth Furnace Co., The, •123
National Heater Co., •124–125
Tjernlund Mfg. Co., •127
Weaz Electric Heater Co., •148

HEATERS, Blast
Aerofin Corp., •186–187
Aerovent Fan Co., Inc., •197
American Blower, Div. of American-Standard, •66–69
Bayley Blower Co., •200
M. Blazer & Son, •93
Carrier Corporation, •74–75, 130
Dunham-Bush, Inc., •406–409
Electromode, Div. of Commercial Controls Corp., •144
G & O Mfg. Co., The, •190
Industrial Engineering & Equipment Co., •145
Marlo Coil Co., •86
McQuay, Inc., •132–133
D. J. Murray Mfg. Co., •135
Herman Nelson, Div., American Air Filter Co., Inc., •128–129
Niagara Blower Co., •87
Trane Co., The, •128–139
Weaz Electric Heater Co., •148
Westinghouse Electric Corp., Sturtevant Div., •225
Edwin L. Wiegand Co., •146–147
Young Radiator Co., •143

HEATERS, Cabinet
Airtherm Mfg. Co., •320
American Blower, Div. of American-Standard, •66–69
Carrier Corporation, •74–75, 130
Dunham-Bush, Inc., •406–409
Electromode, Div. of Commercial Controls Corp., •144
Ilg Electric Ventilating Co., •194, 214
Kritzer Resistant Coils, Inc., •311
McQuay, Inc., •132–133
Herman Nelson, Div., American Air Filter Co., Inc., •128–129
John J. Nesbitt, Inc., •142
Rittling Corp., The, •192
Trane Co., The, •128–139
Weaz Electric Heater Co., •148
Edwin L. Wiegand Co., •146–147
Young Radiator Co., •143

HEATERS, Convection (*See Baseboard Heating, Convection Heaters*)

HEATERS, Duct
Campbell Heating Co., •114–115
Carrier Corporation, •74–75, 130
Chicago Steel Furnace Co., •117
Dravo Corp., •118
Electromode, Div. of Commercial Controls Corp., •114
Jackson & Church, Div. of York-Shipley, Inc., •119
Johnson Heater Corp., •120–121
Lennox Industries, Inc., •109, 122
Mammoth Furnace Co., The, •123
National Heater Co., •124–125
Tjernlund Mfg. Co., •127
Weaz Electric Heater Co., •148

HEATERS, Electric
Electromode, Div. of Commercial Controls Corp., •144
Industrial Engineering & Equipment Co., •145
Herman Nelson Div., American Air Filter Co., Inc., •128–129
Trade-Wind Motorfans, Inc., •222
Weaz Electric Heater Co., •148
Edwin L. Wiegand Co., •146–147

HEATERS, Feed Water
Adsco Div., Yuba Consolidated Industries, Inc., •296
Bell & Gossett Co., •383–385
Davis Engineering Corp., •385

Patterson-Kelley Co., Inc., The, •191
Richmond Engineering Co., Inc., •388
Warren Webster & Co., •416-417
Worthington Corp., Air Condg. & Refrigeration Div., •85
Yuba Corp., formerly Yuba Water Heaters, Inc., •393

HEATERS, Fuel Oil
Adsco Div., Yuba Consolidated Industries, Inc., •296
Bell & Gossett Co., •382-383
Campbell Heating Co., •114-115
Cyclotherm, Div. of National-U. S. Radiator Corp., •345
Davis Engineering Corp., •385
Dravo Corp., •118
General Fittings Co., •386
Industrial Engineering & Equipment Co., •145
Jackson & Church, Div. of York-Shipley, Inc., •119
Johnson Heater Corp., •120-121
Killebrew Engineering Corp., •387
Mammoth Furnace Co., The, •123
National Heater Co., Inc., •124-125
Patterson-Kelley Co., Inc., The, •191
Richmond Engineering Co., Inc., •388
Taco Heaters, Inc., •392
H. A. Thrush & Co., •390-391
Titusville Iron Works Co., The, Div. of Struthers-Wells Corp., •353
Tjernlund Mfg. Co., •137
Edwin L. Wiegand Co.; •146-147
Wilson Engineering Corp., •350

HEATERS, Gas
E. K. Campbell Co., •116
Campbell Heating Co., •114-115
Chicago Steel Furnace Co., •117
Cyclotherm Div. of National-U. S. Radiator Corp., •345
Dravo Corp., •118
Hayes Furnace Mfg. & Supply Co., •106-107
Jackson & Church, Div. of York-Shipley, Inc., •119
Johnson Heater Corp., •120-121
Lennox Industries, Inc., •122
Mammoth Furnace Co., The, •123
National Heater Co., Inc., •124-125
Herman Nelson Div., American Air Filter Co., Inc., •128-129
Thermobloc Div., Prat-Daniel Corp., •126
Tjernlund Mfg. Co., •137
Westinghouse Electric Corp., Sturtevant Div., •225

HEATERS, Hot Water Service
Aero Corp., •381
Air Devices, Inc., •155, 240-241
Aldrich Co., •377
Bell & Gossett Co., •382-383
Bryan Steam Corp., •344
Burnham Corp., •324
Carlin Co., The, •371
Crane Co., •322-323
Cyclotherm Div. of National-U. S. Radiator Corp., •345
Davis Engineering Corp., •385
General Fittings Co., •386
S. T. Johnson Co., •374-375
Johnston Bros., Inc., •350
Killebrew Engineering Corp., •387
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325-327
Patterson-Kelley Co., Inc., The, •191
Peerless Heater Co., •330-331
Raypak Co., Inc., •352
Richmond Engineering Co., Inc., •388
H. B. Smith Co., Inc., The, •333
Taco Heaters, Inc., •392
Titusville Iron Works Co., The, Div. of Struthers-Wells Corp., •353
Weil-McLain Co., •326
Yuba Corp., formerly Yuba Water Heaters, Inc., •393

HEATERS, Indirect
Adsco Div., Yuba Consolidated Industries, Inc., •296
Aero Corp., •381
Aerofin Corp., •136-137
American Tube Products, Inc., •384
Bell & Gossett Co., •382-383
Campbell Heating Co., •114-115
General Fittings Co., •386
Killebrew Engineering Corp., •387
Patterson-Kelley Co., Inc., The, •191
Taco Heaters, Inc., •392
H. A. Thrush & Co., •390-391
York-Shipley, Inc., •380
Yuba Corp., formerly Yuba Water Heaters Inc., •393

HEATERS, Space, Direct-Fired
American Foundry & Furnace Co., •104-105
Campbell Heating Co., •114-115
Chicago Steel Furnace Co., •117
Dravo Corp., •118
Jackson & Church, Div. of York-Shipley, Inc., •119
Johnson Heater Corp., •120-121
Lennox Industries, Inc., •122
Mammoth Furnace Co., The, •123
National Heater Co., Inc., •124-125
Thermobloc Div., Prat-Daniel Corp., •126
Tjernlund Mfg. Co., •137
Vapor Heating Corp., •394
York-Shipley, Inc., •380

HEATERS, Storage
Adsco Div., Yuba Consolidated Industries, Inc., •296
Davis Engineering Corp., •385
General Fittings Co., •386
Killebrew Engineering Corp., •387
Patterson-Kelley Co., Inc., The, •191
Richmond Engineering Co., Inc., •388
Taco Heaters, Inc., •392
Yuba Corp., formerly Yuba Water Heaters, Inc., •393

HEATERS, Tank
Adsco Div., Yuba Consolidated Industries, Inc., •296
Aero Corp., •381
Bell & Gossett Co., •382-383
Drayer-Hanson, Inc., Div. of National-U. S. Radiator Corp., •78
General Fittings Co., •386
Industrial Engineering & Equipment Co. •145
Killebrew Engineering Corp., •387
Patterson-Kelley Co., Inc., The, •191
Peerless Heater Co., •330-331
Richmond Engineering Co., Inc., •388
Weil-Mclain Heater Co., •148
Edwin L. Wiegand Co., •146-147
Yuba Corp., formerly Yuba Water Heaters, Inc., •393

HEATERS, Unit
Aerovent Fan Co., Inc., •197
American Blower, Div. of American-Standard, •66-69
American Foundry & Furnace Co., •104-105
Buffalo Forge Co., •202
Burnham Corp., •324
Campbell Heating Co., •114-115
Carrier Corporation, •74-75, 130
Chicago Steel Furnace Co., •117
Clarage Fan Co., •76, 206
Crane Co., •322-323
Dravo Corp., •118
Dunham-Bush, Inc., •406-409
Fedders Corporation, •131
Hayes Furnace Mfg. & Supply Co., •106-107
Ilg Electric Ventilating Co., •134, 214
Jackson & Church, Div. of York-Shipley, Inc., •119
Johnson Heater Corp., •120-121
Kritzer Products, A Div. of Peerless of America, Inc., •103
Kritzer Radiant Coils, Inc., •311
Larco Inc., •108
Lennox Industries, Inc., •122
Mammoth Furnace Co., The, •123
McQuay, Inc., •132-133
Marlo Coil Co., •86
Muckle Mfg. Co., •235
D. J. Murray Mfg. Co., •135
National Heater Co., Inc., •124-125
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325-327
Herman Nelson Div., American Air Filter Co., Inc., •128-129
John J. Nesbitt, Inc., •142
New York Blower Co., •215
Niagara Blower Co., •87
Rittling Corp., The, •192
Sheldons Engineering Ltd., •220
Thermobloc Div., Prat-Daniel Corp., •126
Trane Co., The, •136-139
Warren Webster & Co., •416-417
Western Blower Co., •284
Westinghouse Electric Corp., Sturtevant Div., •225
Edwin L. Wiegand Co., •146-147
Wilson Engineering Corp., •260
L. J. Wing Mfg. Co., •140-141
Young Radiator Co., •143

HEATERS, Unit, Electric
Electromode, Div. of Commercial Controls Corp., •144
Ilg Electric Ventilating Co., •134, 214
Weil Electric Heater Co., •148
Edwin L. Wiegand Co., •146-147
L. J. Wing Mfg. Co., •140-141

HEATERS, Unit, Gas Fired
Aerovent Fan Co., Inc., •197
Airtherm Mfg. Co., •320
American Blower, Div. of American-Standard, •66-69
American Foundry & Furnace Co., •104-105
Burnham Corp., •324
Campbell Heating Co., •114-115
Carrier Corporation, •74-75, 130
Chicago Steel Furnace Co., •117
Dravo Corp., •118
Dura-Vent Corp., Affiliate of Peerless Mfg. Div. of Dover Corp., •388
Fedders Corporation, •131
Hayes Furnace Mfg. & Supply Co., •106-107
Ilg Electric Ventilating Co., •134, 214
Jackson & Church, Div. of York-Shipley, Inc., •119
Johnson Heater Corp., •120-121
Lennox Industries, Inc., •122
Mammoth Furnace Co., The, •123
National Heater Co., Inc., •124-125
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325-327
Herman Nelson Div., American Air Filter Co., Inc., •128-129
John J. Nesbitt, Inc., •142
Thermobloc Div., Prat-Daniel Corp., •126
Trane Co., The, •136-139
Warren Webster & Co., •416-417
Westinghouse Electric Corp., Sturtevant Div., •225
L. J. Wing Mfg. Co., •140-141

HEATERS, Unit, Oil Fired
Airtherm Mfg. Co., •320
American Foundry & Furnace Co., •104-105
Campbell Heating Co., •114-115
Chicago Steel Furnace Co., •117
Dravo Corp., •118
Jackson & Church, Div. of York-Shipley, Inc., •119
Johnson Heater Corp., •120-121
Lennox Industries, Inc., •122
Mammoth Furnace Co., The, •123
National Heater Co., Inc., •124-125
Thermobloc Div., Prat-Daniel Corp., •126
Trane Co., The, •136-139

HEATING DUCTS, Under Concrete Floor Slabs (*See Ducts, Under Concrete Floor Slabs*)

HEATING EQUIPMENT, Air, Heavy Duty
American Foundry & Furnace Co., •104-105
E. K. Campbell Co., •116
Campbell Heating Co., •114-115
Chicago Steel Furnace Co., •117
Dravo Corp., •118
Johnson Heater Corp., •120-121
Lennox Industries, Inc., •109, 122
Majestic Co., Inc., The, •110
National Heater Co., Inc., •124-125
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325-327
Peerless Heater Co., The, •330-331

HEATING EQUIPMENT, Electric
Electromode, Div. of Commercial Controls Corp., •144
Industrial Engineering & Equipment Co., •145
Majestic Co., Inc., The, •110
Weil Electric Heater Co., •148
Edwin L. Wiegand Co., •146-147

HEATING EQUIPMENT, Furnace
Airtemp Div., Chrysler Corp., •68-69
Airtherm Mfg. Co., •320
Crane Co., •322-323
General Electric Co., •83-85
Johnson Heater Corp., •120-121
Kritzer Products, A Div. of Peerless of America, Inc., •103
Lennox Industries, Inc., •109, 122
Majestic Co., Inc., The, •110
National Clay Pipe Mfgrs., The, •314-317
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325-327
Peerless Heater Co., The, •330-331
Pennsylvania Furnace & Iron Co., •111, 332
Trane Co., The, •136-139
Edwin L. Wiegand Co., •146-147
York Corp., Sub. Borg-Warner Corp., •96
York-Shipley, Inc., •380

HEATING EQUIPMENT, Automatic
Airtemp Div., Chrysler Corp., •68-69
Airtherm Mfg. Co., •320
American Foundry & Furnace Co., •104-105
American Blower, Div. of American-Standard, •66-69
American-Standard, Kewanee Boiler Div., •338
American Standard, Plumbing & Heating Div., •318-319
E. K. Campbell Co., •116
Campbell Heating Co., •114-115
Carlin Co., The, •371
Chicago Steel Furnace Co., •117
Columbia Boiler Co. of Pottstown, •342-343
Crane Co., •322-323
Dravo Corp., •118
Dunham-Bush, Inc., •406-409
Federal Boiler Co., •346
Hydrotherm, Inc., •348
Illinois Engineering Co., Div. of American Air Filter Co., Inc., •418
Jackson & Church, Div. of York-Shipley, Inc., •119
Johnson Heater Corp., •120-121
S. T. Johnson Co., •374-375
Lennox Industries, Inc., •109, 122
Majestic Co., Inc., The, •110
Mammoth Furnace Co., The, •123
National Heater Co., Inc., •124-125
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325-327
Peerless Heater Co., •330-331
Pennsylvania Furnace & Iron Co., •111, 332
Petro, •378-379
Ray Burner Co., •377
Thermobloc Div., Prat-Daniel Corp., •125
Titusville Iron Works Co., The, Div. of Struthers-Wells Corp., •353
Trane Co., The, •136-139
Weil-McLain Co., •326
York-Shipley, Inc., •380

HEATING EQUIPMENT, Coal Fired
American Foundry & Furnace Co., •104-105
American Blower, Div. of American-Standard, •66-69
American-Standard, Kewanee Boiler Div., •338
American Standard, Plumbing & Heating Div., •318-319
E. K. Campbell Co., •116
Campbell Heating Co., •114-115
Columbia Boiler Co. of Pottstown, •342-343
Crane Co., •322-323
Lennox Industries, Inc., •109, 122
Majestic Co., Inc., The, •110
National Heater Co., Inc., •124-125
Peerless Heater Co., •330-331

HEATING EQUIPMENT, Gas Fired
Airtemp Div., Chrysler Corp., •68-69
Airtherm Mfg. Co., •320
American Blower, Div. of American-Standard, •66-69
American Foundry & Furnace Co., •104-105
American-Standard, Kewanee Boiler Div., •338
American Standard, Plumbing & Heating Div., •318-319
Bryan Steam Corp., •344
E. K. Campbell Co., •116
Campbell Heating Co., •114-115
Carlin Co., The, •371
Chicago Steel Furnace Co., •117
Columbia Boiler Co. of Pottstown, •342-343
Crane Co., •322-323
Dravo Corp., •118
Federal Boiler Co., •346
General Automatic Products Corp., •310
General Electric Co., •83-85
Hayes Furnace Mfg. & Supply Co., •106-107
Jackson & Church, Div. of York-Shipley, Inc., •119
Johnson Heater Corp., •120-121
Lennox Industries, Inc., •109, 122
Majestic Co., Inc., The, •110
Mammoth Furnace Co., The, •123
National Heater Co., Inc., •124-125
National-U. S. Radiator Corp., Heating & Air Conditioning Div., •325-327
Peerless Heater Co., The, •330-331
Pennsylvania Furnace & Iron Co., •111, 332
Petro, •378-379
Ray Burner Co., •377
Thermobloc Div., Prat-Daniel Corp., •125
Titusville Iron Works Co., The, Div. of Struthers-Wells Corp., •353
Trane Co., The, •136-139
Weil-McLain Co., •326
York-Shipley, Inc., •380

HEATING EQUIPMENT, High-temperature Fluid
International Boiler Works Co., The, •349
Parks-Cramer Co., •85
Titusville Iron Works Co., The, Div. of Struthers-Wells Corp., •353

HEATING EQUIPMENT, Hot Water
Air Devices, Inc., •155, 240-241
American-Standard, Kewanee Boiler Div., •338
American-Standard, Plumbing & Heating Div., •318-319
Bell & Gossett Co., •382-383
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Killebrew Engineering Corp., •387
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General Controls, •272-273
Illinois Engineering Div., American Air Filter Co., Inc., •418
Illinois Testing Laboratories, Inc., •275
Johnson Service Co., •276-277
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Jas. P. Marsh Corp., •423
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Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287
Sarco-Sarcotherm, •414-415
Spence Engineering Co., Inc., •289
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Watts Regulator Co., •426-427
Weaix Electric Heater Co., •148
White-Rodgers Co., •290
Wilcolator Co., The, •291
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Minneapolis-Honeywell Regulator Co., •170, 251
Moeller Instrument Co., •222
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Johnson Service Co., •276-277
Jas. P. Marsh Corp., •423
Minneapolis-Honeywell Regulator Co., •170, 251
Moeller Instrument Co., •222
Powers Regulator Co., The, •254-255

THERMOMETERS, Recording
Johnson Service Co., •276-277
Minneapolis-Honeywell Regulator Co., •170, 251
Moeller Instrument Co., •222
Powers Regulator Co., The, •254-255

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Combustion Control Div., Electronics Corp. of America, •294
General Controls, •272-273
Johnson Service Co., •276-277
Leslie Co., •278
Mercoid Corp., The, •280
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V. D. Anderson Co., The, Div. of International Basic Economy Corp., •404-405
Armstrong Machine Works, •71, 356
Sarco Co., Inc., •414-415
Strong, Carlisle & Hammond, •425

TRAPS, Bucket
V. D. Anderson Co., The, Div. of International Basic Economy Corp., •404-405
Armstrong Machine Works, •71, 356
Crane Co., •322-323
Hoffman Specialty Mfg. Corp., •410-411
Illinois Engineering Div., American Air Filter Co., Inc., •418
Jas. P. Marsh Corp., •423
Sarco Co., Inc., •414-415
Strong, Carlisle & Hammond, •425
Trane Co., The, •138-139

TRAPS, Float
V. D. Anderson Co., The, Div. of International Basic Economy Corp., •404-405
Armstrong Machine Works, •71, 356
Crane Co., •322-323
Dunham-Bush, Inc., •406-409
Hoffman Specialty Mfg. Corp., •410-411
Jas. P. Marsh Corp., •423
Sarco Co., Inc., •414-415
Strong, Carlisle & Hammond, •425
Trane Co., The, •138-139

TRAPS, Float and Thermostatic
V. D. Anderson Co., The, Div. of International Basic Economy Corp., •404-405
Armstrong Machine Works, •71, 356
Dunham-Bush, Inc., •406-409
Hoffman Specialty Mfg. Corp., •410-411
Illinois Engineering Div., American Air Filter Co., Inc., •418
Jas. P. Marsh Corp., •423
Sarco Co., Inc., •414-415
Sterling, Inc., •389
Strong, Carlisle & Hammond, •425
Trane Co., The, •138-139
Warren Webster & Co., •416-417

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TRAPS, Radiator
Crane Co., •322-323
Dunham-Bush, Inc., •406-409
Hoffman Specialty Mfg. Corp., •410-411
Illinois Engineering Div., American Air Filter Co., Inc., •418
Jas. P. Marsh Corp., •423
Sarco Co., Inc., •414-415
Sterling, Inc., •389
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TRAPS, Return
Crane Co., •322-323
Dunham-Bush, Inc., •406-409
Illinois Engineering Div., American Air Filter Co., Inc., •418
Jas P. Marsh Corp., •423
Trane Co., The, •138-139

TRAPS, Scale Refrigerant
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TRAPS, Steam
V. D. Anderson Co., The, Div. of International Basic Economy Corp., •404-405
Armstrong Machine Works, •71, 356
Crane Co., •322-323
Dunham-Bush, Inc., •406-409
Hoffman Specialty Mfg. Corp., •410-411
Illinois Engineering Div., American Air Filter Co., Inc., •418
Jas. P. Marsh Corp., •423
Sarco Co., Inc., •414-415
Sterling, Inc., •389
Strong, Carlisle & Hammond, •425
Trane Co., The, •138-139
Warren Webster & Co., •416-417

TRAPS, Thermostatic
V. D. Anderson Co., The, Div. of International Basic Economy Corp., •404-405
Crane Co., •322-323
Dunham-Bush, Inc., •406-409
Hoffman Specialty Mfg. Corp., •410-411
Illinois Engineering Div., American Air Filter Co., Inc., •418
Jas. P. Marsh Corp., •423
Powers Regulator Co., The, •254-255
Sarco Co., Inc., •414-415
Sterling, Inc., •389
Trane Co., The, •138-139
Warren Webster & Co., •416-417

TRAPS, Vacuum
V. D. Anderson Co., The, Div. of International Basic Economy Corp., •404-405
Armstrong Machine Works, •71, 356
Illinois Engineering Div., American Air Filter Co., Inc., •418
Jas. P. Marsh Corp., •423
Sarco Co., Inc., •414-415
Strong, Carlisle & Hamroond, •425

TUBES, Boiler
Babcock & Wilcox Tube Co., The, •339

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Mueller Brass Co., •424
Reading Tube Corp., •183-189
Wolverine Tube, •153-153

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Wolverine Tube, •153-153

TUBING, Fabricated
American Brass Co., The, American Metal Hose Div., •149
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G & O Mfg. Co., The, •190
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Reading Tube Corp., •183-189
Rittling Corp., The, •192
Wolverine Tube, •153-153

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American Brass Co., The, •150-151
American Brass Co., The, American Metal Hose Div., •149
Flexonics Corp., Expansion Joint Div., •297
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TUBING, RESTRICTOR
American Brass Co., The, •150-151
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TUBING, Unicellular Rubber
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UNITS, Air Conditioning (*See Air Conditioning Units*)

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Desomatic Products, Inc., •77
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Fairbanks Co., The, •421
Hammond Brass Works, •Insert Section
Hoffman Specialty Mfg. Corp., •410-411
Jenkins Bros., •422
Maid-O'-Mist, Inc., •413-413
Jas. P. Marsh Corp., •423

VALVES, Angle, Globe and Cross
Crane Co., •322-323
Fairbanks Co., The, •421
Frick Co., •195
Hammond Brass Works, •Insert Section
Henry Valve Co., •274
Jenkins Bros., •422
Mueller Brass Co., •424

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Barber-Colman Co., •247, 271
A. W. Cash Valve Mfg. Corp., •419
Combustion Control Div., Electronics Corp. of America, •294
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McDonnell & Miller, Inc., •358-359
Minneapolis-Honeywell Regulator Co., •170, 251
Powers Regulator Co., The, •254-255
Spence Engineering Co., Inc., •289
Watts Regulator Co., •426-427
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VALVES, Balancing
American Tube Products, Inc., •384
General Fittings Co., •386
Hammond Brass Works, •Insert Section
Maid-O'-Mist, Inc., •413-413
Sarco-Sarcotherm, •414-415
H. A. Thrush & Co., •390-391

VALVES, Blowoff
Fairbanks Co., The, •421
Jenkins Bros., •422

VALVES, By-Pass
Fairbanks Co., The, •421
Jenkins Bros., •422
Mueller Brass Co., •424

VALVES, Check
Fairbanks Co., The, •421
Hammond Brass Works, •Insert Section
Henry Valve Co., •274
Jenkins Bros., •422
Mueller Brass Co., •424

VALVES, Corrosion Resisting
Crane Co., •322-323
Jenkins Bros., •422

VALVES, Diaphragm
A. W. Cash Valve Mfg. Crop., •419
Crane Co., •322-323
General Controls, •272-273
Henry Valve Co., •274
Johnson Service Co., •276-277
Leslie Co., •278
Minneapolis-Honeywell Regulator Co., •170, 251
Mueller Brass Co., •424
Powers Regulator Co., The, •254-255
Watts Regulator Co., •426-427
White-Rodgers Co., •290

VALVES, Expansion
Alco Valve Co., •270
Crane Co., •322-323
General Controls, •272-273
Henry Valve Co., •274
Mueller Brass Co., •424
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287

VALVES, Float
Alco Valve Co., •270
Maid-O'-Mist, Inc., •413-413
McDonnell & Miller, Inc., •358-359
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •284-285
Watts Regulator Co., •426-427

VALVES, Flow Control
Bell & Gossett Co., •383-383
Crane Co., •322-323
Dole Valve Co., •420
Dunham-Bush, Inc., •406-409
Jas. P. Marsh Corp., •423
Minneapolis-Honeywell Regulator Co., •170, 251
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287
H. A. Thrush & Co., •390-391
Warren Webster & Co., •416-417

VALVES, Gas
Combustion Control Div., Electronics Corp. of America, •294
Fairbanks Co., The, •421
Hammond Brass Works, •Insert Section
Jenkins Bros., •422
Minneapolis-Honeywell Regulator Co., •170, 251
Penn Controls, Inc., •283
White-Rodgers Co., •290

VALVES, Gate
Crane Co., •322-323
Fairbanks Co., The, •421
Hammond Brass Works, •Insert Section
Jenkins Bros., •422
Mueller Brass Co., •424

VALVES, Humidifier
Maid-O'-Mist, Inc., •413-413
McDonnell & Miller, Inc., •358-359

VALVES, Magnetic
Alco Valve Co., •270
General Controls, •272-273
McDonnell & Miller, Inc., •358-359
Penn Controls, Inc., •283

VALVES, Mixing, Thermostatic
Dole Valve Co., •420
General Fittings Co., •386
Powers Regulator Co., The, •254-255
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287
Sarco Co., Inc., •414-415
Watts Regular Co., •426-427

VALVES, Motor Operated
Barber-Colman Co., •247, 271
Bell & Gossett Co., •382-383
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Desomatic Products, Inc., •77
General Controls, •272-273
Illinois Engineering Div., American Air Filter Co., Inc., •418
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Johnson Service Co., •276-277
Minneapolis-Honeywell Regulator Co., •170, 251
Powers Regulator Co., The, •254-255
Sarco Co., Inc., •414-415
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Fairbanks Co., The, •421
Hammond Brass Works, •Insert Section
Jenkins Bros., •422

VALVES, Packless
Dunham-Bush, Inc., •406-409
Henry Valve Co., •274
Hoffman Specialty Mfg. Corp., •410-411
Johnson Service Co., •276-277
Jas. P. Marsh Corp., •423
Mueller Brass Co., •424
Powers Regulator Co., The, •254-255
Watts Regulator Co., •426-427
White-Rodgers Co., •290

VALVES, Expansion
Alco Valve Co., •270
Crane Co., •322-323
General Controls, •272-273
Henry Valve Co., •274
Mueller Brass Co., •424
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287

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Hammond Brass Works, •Insert Section
Sarcotherm Controls, Inc., •414-415
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VALVES, Radiator
Crane Co., •322-323
Dole Valve Co., The, •420
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Hammond Brass Works, •Insert Section
Hoffman Specialty Mfg. Corp., •410-411
Illinois Engineering Div., American Air Filter Co., Inc., •418
Jenkins Bros., •422
Maid-O'-Mist, Inc., •413-413
Jas. P. Marsh Corp., •423
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Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287
Sarco Co., Inc., •414-415
Sterling, Inc., •389
Trane Co., The, •138-139
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VALVES, Radiator, Convector
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287

VALVES, Radiator, Electric
Barber-Colman Co., •247, 271
Minneapolis-Honeywell Regulator Co., •170, 251
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287

VALVES, Radiator, Humidifying
Maid-O'-Mist, Inc., •413-413

VALVES, Radiator Orifice
Dunham-Bush, Inc., •406-409
Hammond Brass Works, •Insert Section
Illinois Engineering Div., American Air Filter Co., Inc., •418
Warren Webster & Co., •416-417

VALVES, Radiator, Convector, Pneumatic Diaphragm
Johnson Service Co., •276-277
Minneapolis-Honeywell Regulator Co., •170, 251
Powers Regulator Co., The, •254-255

VALVES, Radiator, Thermostatic
Sterling, Inc., •389

VALVES, Reducing
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A. W. Cash Valve Mfg. Corp., •419
Crane Co., •322-323
Dunham-Bush, Inc., •406-409
Hoffman Specialty Mfg. Corp., •410-411
Leslie Co., •278
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287
Spence Engineering Co., Inc., •289
Strong, Carlisle & Hammond, •425
H. A. Thrush & Co., •390-391
Watts Regulator Co., •426-427

VALVES, Refrigerant Line
Alco Valve Co., •270
Dole Refrigerating Co., •194
Hammond Brass Works, •Insert Section
Henry Valve Co., •274
Jenkins Bros., •422
Mueller Brass Co., •424
Worthington Corp., Air Condg. & Refrigeration Div., •95

VALVES, Relief
V. D. Anderson Co., The, Div. of International Basic Economy Corp., •404-405
Bell & Gossett Co., •382-383
A. W. Cash Valve Mfg. Corp., •419
Dole Valve Co., The, •420
Dunham-Bush, Inc., •406-409
Henry Valve Co., •274
Jas. P. Marsh Corp., •423
McDonnell & Miller, Inc., •358-359
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Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287
Taco Heaters, Inc., •392
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Trane Co., The, •138-139
Tuthill Pump Co., •462
Watts Regulator Co., •426-427
York Corp., Sub. Borg-Warner Corp., •96

VALVES, Reversing, Heat Pump
Ranco Inc., •288

VALVES, Safety
A. W. Cash Valve Mfg. Corp., •419
Combustion Control Div., Electronics Corp., of America, •294
Frick Co., •195
General Controls, •272-273
Jas. P. Marsh Corp., •423
McDonnell & Miller, Inc., •358-359
Mueller Brass Co., •424
Watts Regulator Co., •426-427

VALVES, Solenoid
Alco Valve Co., •270
Combustion Control Div., Electronics Corp. of America, •294
Dole Valve Co., •420
General Controls, •272-273
Jas. P. Marsh Corp., •423
McDonnell & Miller, Inc., •358-359
Penn Controls, Inc., •283
Ranco Inc., •288
Spence Engineering Co., Inc., •289
White-Rodgers Co., •290

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American Tube Products, Inc., •384
Dole Valve Co., The, •420
General Fittings Co., •386
Powers Regulator Co., The, •254-255
Taco Heaters, Inc., •392
Watts Regulator Co., •426-427
Yuba Corp., formerly Yuba Waters, Inc. •393

VALVES, Thermostatic
Alco Valve Co., •270
Barber-Colman Co., •247, 271
Crane Co., •322-323
General Controls, •272-273
Leslie Co., •278
Jas. P. Marsh Corp., •423
Powers Regulator Co., The, •254-255
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287
Spence Engineering Co., Inc., •289
Sterling, Inc., •389
Wilcolator Co., The, •291
Yuba Corp., formerly Yuba Waters, Inc., •393

VALVES, Water Flow Regulating
Bell & Gossett Co., •382-383
Dole Valve Co., The, •420

Dunham-Bush, Inc., •406-409
General Controls, •273-275
Leslie Co., •278
Johnson Service Co., •276-277
Minneapolis-Honeywell Regulator Co., •170, 281
Powers Regulator Co., The, •284-285
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287
Spence Engineering Co., •289
Sterling, Inc., •289
Taco Heaters, Inc., •392
H. A. Thrush & Co., •390-391
Watts Regulator Co., •425-427

VALVES, Water Level, Float Control
Maid-O'-Mist, Inc., •412-413
McDonnell & Miller, Inc., •358-359
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287
Watts Regulator Co., •425-427

VALVES, Water Regulating
Alco Valve Co., •270
A. W. Cash Valve Mfg. Corp., •419
Dole Valve Co., •420
Hammond Brass Works, •Insert Section
Leslie Co., •278
Jas. P. Marsh Corp., •432
McDonnell & Miller, Inc., •358-359
Penn Controls, Inc., •282
Powers Regulator Co., The, •284-285
Robertshaw-Fulton Controls Co., Fulton-Sylphon Div., •286-287
Spence Engineering Co., •289
Sterling, Inc., •289
Watts Regulator Co., •425-427

VANES, Air Turning
Barber-Colman Co., •247, 271

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Philip Carey Mfg. Co., •446-447
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VENTILATING SKYLIGHTS (*See Skylights, Metal and Ventilating*)

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G. C. Breidert Co., •232
Buffalo Forge Co., •202
Champion Blower & Forge Co., •204
Ilg Electric Ventilating Co., •134, 214
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Reed Unit-Fans, Inc., •218
Torrington Mfg. Co., The, •221
Trade-Wind Motorfans, Inc., •222
Western Engrg. & Mfg. Co., •237
L. J. Wing Mfg. Co., •140-141

VENTILATORS, Ceiling
Buffalo Forge Co., •202
Trade-Wind Motorfans, Inc., •222

VENTILATORS, Fire
Colt Ventilation Co. of America, Inc., •228-229

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A-J Mfg. Co., •244-245
Aerovent Fan Co., Inc., •197
Air Control Products, Inc., •238
Ammerman Co., Inc., •231
Auer Register Co., The, •246
Barber-Colman Co., •247, 271
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Carnes Corp., •248-249
DeBothezat Fans Div., American Machine & Metals, Inc., •207
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Ilg Electric Ventilating Co., •134, 214
Independent Register Co., The, •255
Jenn Air Products Co., Inc., •234
Penn Ventilator Co., •236
Propellair Div., Robbins & Myers, Inc., •217
Reed Unit-Fans, Inc., •218
Titus Mfg. Corp., •Insert Section
Trane Co., The, •136-139
United States Register Co., •260-261
Universal Diffuser Corp., •264
Waterloo Register Co., Inc., •265

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G. C. Breidert Co., The, •232
Century Fan & Ventilator Co., Inc., •203
Jenn Air Products Co., Inc., •234
Muckle Mfg. Co., •235
Western Engrg. & Mfg. Co., •237

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Aerovent Fan Co., Inc., •197
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Aladdin Heating Corp., •199
Allen Cooler & Ventilator, Inc. •230
American Blower, Div. of American-Standard, •66-69
American Foundry & Furnace Co., •104-105
American Warming & Ventilating Co., •226
Ammerman Co., Inc., •231
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Hirschman-Pohle Co., Inc., •233
Ilg Electric Ventilating Co., •134, 214
Jenn Air Products Co., Inc., •234
Muckle Mfg. Co., •235
New York Blower Co., •215
Peerless Electric Co., The, •216
Penn Ventilator Co., •236
Propellair Div., Robbins & Myers, Inc., •217
Reed Unit-Fans, Inc., •218
H. H. Robertson Co., •92
Trade-Wind Motorfans, Inc., •222
Trane Co., The, •136-139
Western Engrg. & Mfg. Co., •237
Westinghouse Electric Corp., Sturtevant Div., •176, 225
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VENTILATORS, Ship
Air Devices, Inc., •155, 240-241
Ammerman Co., Inc., •231
G. C. Breidert Co., The, •233
Buffalo Forge Co., •202
Chicago Blower Corp., •205
Penn Ventilator Co., •236
Western Engrg. & Mfg. Co., •237
Westinghouse Electric Corp., Sturtevant Div., •176, 225
L. J. Wing Mfg. Co., •140-141

VENTILATORS, Smoke
Colt Ventilation Co. of America, Inc., •228-229
Propellair Div., Robbins & Myers, Inc., •217
H. H. Robertson Co., •92
Western Engrg. & Mfg. Co., •237

VENTILATORS, Unit
Aerovent Fan Co., Inc., •197
Air & Refrigeration Corp., •63
Allen Cooler & Ventilator, Inc., •230
American Blower, Div. of American-Standard, •66-69
Ammerman Co., Inc., •231
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John J. Nesbitt, Inc., •142
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Ritting Corp., The, •192
Trade-Wind Motorfans, Inc., •222
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Weir Electric Heater Co., •148
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Westinghouse Electric Corp., Sturtevant Div., •176, 225
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VENTILATORS, Wall
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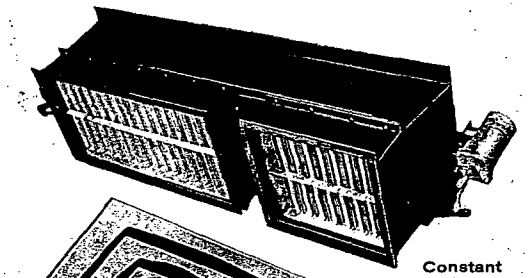
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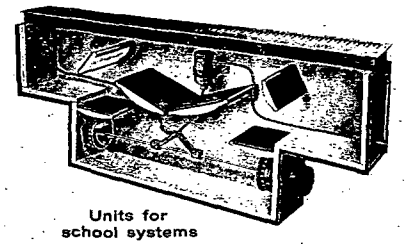
ANEMOSTAT®

presents a complete line of equipment for

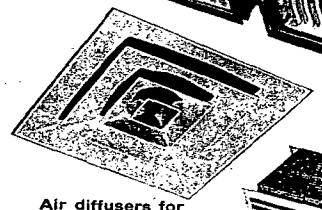
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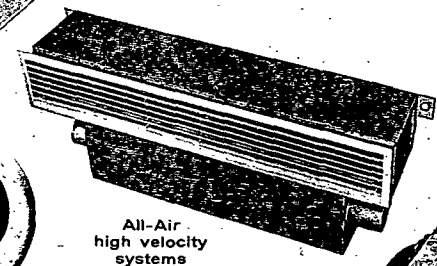
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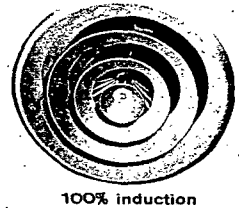
Units for school systems



Air diffusers for conventional systems



All-Air high velocity systems



100% induction air diffusers



Anemotherm Air Meter for *balancing any air system*

THE BEST WAY TO HANDLE AIR IS ANEMOSTAT®

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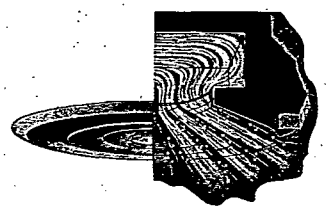
DATA ON ANEMOSTAT •

Principle of Aspiration

The Anemostat Air Diffuser splits the supply air into multiple, separate air streams and at the same time creates low pressure areas in certain parts of diverging passageways. This causes room air to be drawn into the device where it is thoroughly mixed with supply air. The mixture is discharged in a series of expanding turbulent air layers and, therefore, readily entrains a large amount of additional room air.

As a result, temperature and humidity are equalized throughout the room and a continuous air motion in the occupancy zone within low acceptable velocity limits is established. Stagnant air pockets are also avoided.

The quantity of room air drawn into Anemostat Air Diffusers depends on the specific design, size and type of the various units and is equal to as much as 35 per cent of the supply air. This effective aspiration distinguishes Anemostat Air Diffusers from all other air outlets.



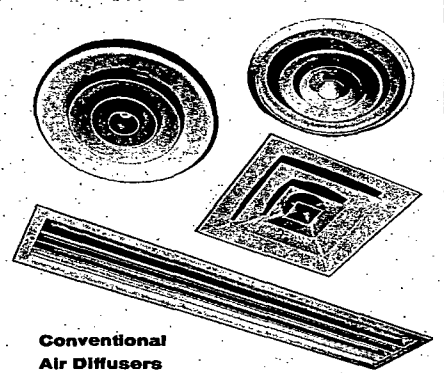
Smaller Ducts

Aspiration permits the use of higher temperature differentials and higher supply air velocities than customary, which results in savings in the initial cost of fans and ducts and in the operating cost of fans. Duct layouts may be simplified because Anemostat Air Diffusers distribute air evenly in spite of unusual room plans, columns or other obstructions.

Research and Engineering

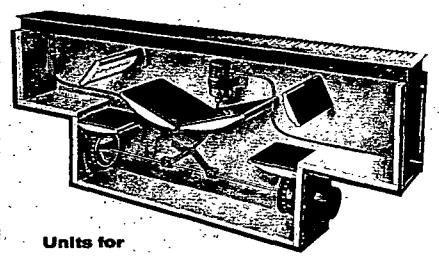
Anemostat Corporation of America maintains a large, well-equipped laboratory manned by experienced scientific and technical personnel for testing purposes and the development of new products. Engineers from all over the world have visited the Anemostat laboratory to witness demonstrations of Anemostat products and their applications to any conceivable problem pertaining to air diffusion. Anemostat has spent millions of dollars in research and engineering and will continue to make substantial contributions to the progress of the science of air distribution.

ANEMOSTAT®.....has a product for every air distribution need



Conventional Air Diffusers

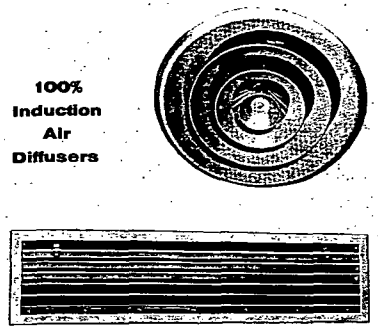
Anemostat air diffusers for conventional heating, ventilating and air conditioning systems provide thorough draftless comfort and uniform air distribution. They are the only air diffusers which have both primary (aspiration) and secondary characteristics. The cumulative effect of both primary and secondary induction assures the superior mixing and distribution of air. Only Anemostat Air Diffusers aspirate. More than a million of these units are serving in installations throughout the country.



Units for School Systems

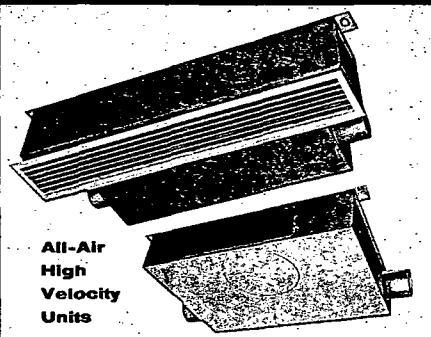
Anemostat Corporation of America has developed by research the ideal unit for heating and ventilating or cooling of classrooms from kindergarten through college and for all climates. When Anemostat Dual Duct Units are used students and teachers work in a controlled atmospheric environment, which protects their health and promotes the efficiency of teacher and student.

Metal cabinets are available where architectural designs require free standing units. Anemostat All-Air under-the-window units can also be built in on exterior walls.



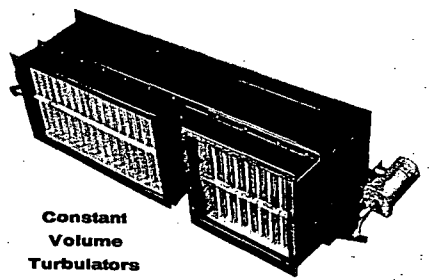
100% Induction Air Diffusers

Anemostat high induction air diffusers provide good diffusion with high temperature differential. Because supply and return ducts, fans and filters are reduced in size in direct proportion as the temperature differential of the distribution air is increased, there is a definite and logical trend to use higher temperature differentials in both low and high velocity systems. Anemostat 100% induction air diffusers meet these needs, thus help provide vital space savings in system design.



All-Air High Velocity Units

Anemostat high velocity units are being used through the country in many applications such as hospitals, schools, office buildings and plants. The Anemostat all-air high velocity draftless system offers many important architectural and engineering advantages. It permits greater flexibility of design and reduces building costs because it can be used with smaller than conventional ducts. It can be installed faster. It requires no coils, thus eliminates leakage, clogging and odors. The Anemostat all-air high velocity system has earned wide-spread acceptance because of the simplicity of its design, installation, control, operation and maintenance.



Constant Volume Turbulators

The Anemostat Constant Volume Turbulator is a standardized high capacity dual duct unit handling from 800 to 7000 CFM. It provides an economical solution to many air distribution problems in which a large volume of air at controlled volume or pressure, and temperature, is involved.

With the Anemostat system, all thermal functions—heating—cooling—ventilating—are accomplished with air. Coils and resultant coil lag are eliminated together with required piping. Pressure losses are low and so are noise levels. Quality-built Anemostat Turbulators function automatically. They are easy to install, simple to maintain.



Anemotherm Air Meter

The Model 60 Anemotherm Air Meter, developed by the Anemostat Corporation of America, gives you—in one convenient instrument—a simple, rapid method of balancing and checking any air system. It puts at your fingertips, by means of color-coded pushbuttons, air velocities, air temperature and static pressure—all vital readings you need. Model 60 Anemotherm Air Meter pays for itself on time saved on only one major job.

More data available on request

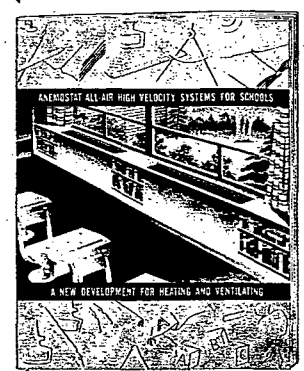
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Send for your free copies of these valuable catalogs and bulletins

•



Selection Manual #60 contains selection and application data on Anemostat Draftless Aspirating Air Diffusers for conventional, 100% induction and high velocity systems.



This new catalog describes the Anemostat Dual Duct High Velocity Air Distribution for heating and ventilating schools. You will find in these pages, typical classroom layouts, performance data, control diagrams and other essential information.



This informative bulletin gives vital data on Anemostat Constant Volume Turbulators for modern air distribution systems. It explains important design features, shows practical applications, contains selection and application data.



Anemotherm Air Meter Bulletin #55 describes the most modern way to balance heating and ventilating and air-conditioning systems. It tells how the Anemotherm Air Meter gives you all vital readings with pushbutton ease.

ANEMOSTAT®
DRAFTLESS ASPIRATING
Air Diffusers

ANEMOSTAT CORPORATION OF AMERICA
10 East 39th Street, New York 16, N. Y.
Representatives in Principal Cities

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63M-5-58

kno-draft®
air diffusers

Connor Engineering Corporation produces a complete line of Kno-Draft air diffusers and high velocity valve terminators, the result of intensive research and development. They are designed to fit every industrial, commercial and residential requirement.

Kno-Draft diffusers distribute air in an effective spiral stream pattern which eliminates drafts and equalizes room temperatures through maximum induction. All models feature the patented cobra-tail sleeve damper which permits precise air volume regulation.

Kno-Draft diffusers are precisely made, accurately assembled and rigidly inspected before shipment. Standard finish for all types is baked aluminum enamel. Oyster enamel is also available on special order to match any ceiling finish.

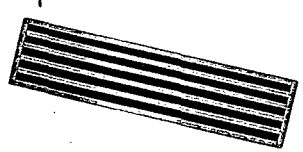
TYPE KDA



Circular Flush-Mounted Air Diffuser

Illustrated is the Kno-Draft KDA adjustable air diffuser. The sleeve damper, together with the adjustable lower cone, permits independent air volume and air direction adjustment after installation. Constructed of light, sturdy spun aluminum, Type KDA diffusers are available in a wide range of sizes from 5" to 36" neck diameter. Bulletin K-20-A.

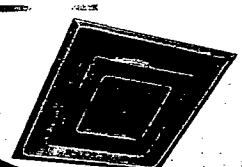
TYPE KLS



Slotted Air Diffuser

Kno-Draft Series KLS all aluminum, adjustable slotted diffusers are particularly adapted to board rooms, foyers, lounges and similar spaces. The harmonious, classic lines of the extruded aluminum vanes complement any decor. These handsome units may be installed singly or butted in series to form a continuous slot for heating or cooling. Series KLS diffusers are complete with built-in air volume and air directional control. Available in lengths from 2 to 6 feet; widths of 3½" and 6" for single direction air pattern, 12" for two-direction air pattern. Bulletin K-27.

TYPE KP



Square Air Diffuser

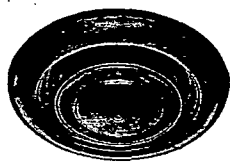
The attractive Kno-Draft square air diffuser is available in two types designed for quick, easy installation — Type KP (overlap) for flush mounting on plaster or certain acoustical ceilings — Type KPT for T-Bar ceilings where the diffuser snaps into the ceiling construction. Both models are identical in air performance characteristics. Despite their square design air is distributed in a true circumferential pattern for maximum effectiveness. Fabricated of pressed steel with baked enamel finish in sizes 5" to 14" neck diameter. Bulletin K-27.

CONNOR ENGINEERING CORPORATION Danbury, Conn.

kno·draft air diffusers

TYPE KDASM

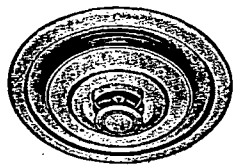
Circular Surface Mounted Diffuser



Building space is conserved with surface mounted Kno-Draft diffusers. The deep streamlined outer cone saves inches in the ever dwindling area above the ceiling by shifting the diffuser body downward. Performance characteristics are identical with the flush mounted KDA diffuser with the further advantage that the deep outer cone inhibits ceiling smudging. Fabricated in spun aluminum in sizes 5" to 18" neck diameter. Bulletin K-38.

TYPE KDSP

Circular Loudspeaker Diffuser

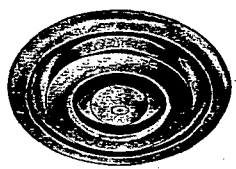


Kno-Draft single air stream, dual-cone design permits the addition of a loudspeaker to the diffuser. As shown, a loudspeaker can be mounted above the center cone allowing both the sound source and diffuser to utilize the same strategic ceiling location. Any 8" speaker may be used and a totally enclosed baffle provides extended frequency response. Sizes 14" to 24" neck diameter with performance characteristics and dimensions identical to standard KDA diffusers.

An integral light may also be provided with Kno-Draft diffusers (Type KDAL). Light fixture units are available in sizes 6" to 16" neck diameter with performance characteristics identical to standard KDA diffusers. Bulletin K-20-A.

TYPE SRD

Circular Supply-Return Diffuser



This Kno-Draft diffuser is used where air is both supplied and returned through the ceiling diffuser. It is so designed that supply air is ejected radially away from the center return air opening, precluding short-circuiting of supply air. The patented sleeve type damper and adjustable cones provide for independent air volume and air direction adjustment. Constructed of spun aluminum in sizes 6" to 24" neck diameter. Bulletin K-20-A.

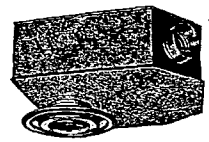
kno·draft series 45 valve attenuators

Kno-Draft Series 45 Valve Attenuators

The Series 45 Valve Attenuator is the result of ten years' experience in the design and application of high velocity units. Features incorporated in its design are all-important to the successful operation of high velocity systems . . . assuring better balanced air delivery and constantly controlled air diffusion.

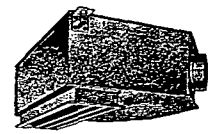
These are the only high velocity units with patented Helical Neoprene-Coated Spring Dampers to insure quiet operation and complete shutoff. Constant Volume Control insures uniform air delivery at all mixture conditions despite static pressure fluctuations. A Manual Control option is available for use when duct design permits. While all units operate quietly, a patented Sinuous Sound Baffle may be added when exceptional silencing is necessary. Series 45 Valve Attenuators are suitable for both dual and single duct systems. Complete information including selection tables, noise specifications, and duct design data in Bulletin K-33-A.

CIRCULAR OR SQUARE DIFFUSER MODEL



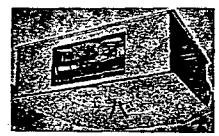
The Diffuser Model has a Kno-Draft ceiling diffuser mounted directly beneath the valve attenuator eliminating inter-connecting low pressured ducts. The linkage and pneumatic components are mounted within the unit and are removable through the diffuser ceiling opening. Six sizes, 80-1500 cfm air delivery.

LINEAR DIFFUSER MODEL

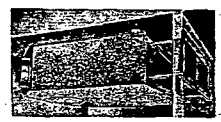


This Model has a Kno-Draft linear diffuser mounted either beneath or at the front of the valve attenuator, permitting ceiling or wall mounting of the linear diffuser. Three sizes, 80-450 cfm air delivery.

OPEN END AND GRILLE MODELS

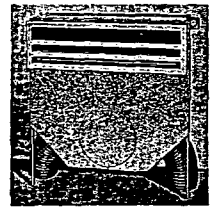


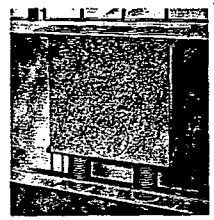
The Open End Model valve attenuator is used to supply remote diffusers or grilles through low pressure duct-work. Two types of outlet connections are available: 1. Single rectangular opening for low pressure rigid duct connection. 2. Multiple round outlets for flexible duct connections.



The Grille Model is identical with the Open End Model except for a larger discharge opening to insure quiet grille operation. Both models are supplied in six sizes, 80-1500 cfm air delivery.

UNDER-WINDOW DUAL DUCT MODELS



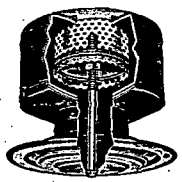


Under-window Models are provided in two standard types as shown. Models with 45° inlets provide front or top outlets to discharge from the wall beneath the sill or directly from the sill as desired. Either front or top air delivery produces the same draftless air flow pattern — upward along the window and out under the ceiling.

Models with vertical bottom inlets are available only with top discharge. The units are dropped through the grille opening and clamped to mounting plates permitting this model to be installed after ductwork and plastering is completed. Accessibility is ideal and access panels are eliminated.

Under-window Models can be equipped with internally mounted thermostats. Sheet metal cabinet enclosures are also available. Three sizes, dual-duct only, 80-450 cfm air delivery.





SERIES HPC VALVE ATTENUATORS

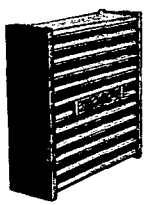
Series HPC Valve Attenuators are manually balanced, single duct units for mounting under rectangular ducts. The HPCI Model is specifically designed for exposed duct installations. Two sizes 90-180 cfm air delivery.

The HPCK Model, similar in internal construction, has a Kno-Draft air diffuser below the valve for flush ceiling installation. Six sizes, 100-1000 cfm air delivery.

dorex®

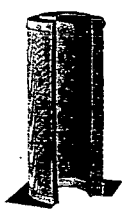
air recovery equipment

Air Recovery is the conversion of contaminated air to fresh air. Dorex Air Recovery equipment adsorbs all air entrained odorous, vaporous and gaseous impurities in the same manner as a gas mask. Its action is positive, neither adding to, subtracting from, nor otherwise altering the chemical composition of pure air. Dorex Air Recovery (air purification) can serve three functions. The purification of (1) recirculated air (2) outside air intake (3) exhaust air.



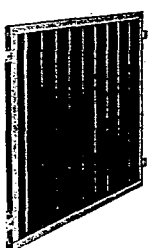
TYPE C CELLS

Dorex Type C Cells are perforated steel cells 24" x 24" x 8¾" deep filled with granular activated carbon. Three models provide different carbon bed thickness and weight for varying applications. Capacities range from 700 to 1000 cfm at 95% efficiency. Cells are installed in sturdy supporting frames and can be designed into new or existing duct systems with no more engineering than standard dust filters. Highly effective corrosion resistant coating available for industrial applications. Bulletin 108.



TYPE H CANISTERS

Type H Canisters lend themselves to a variety of arrangements. Mounted in multiple on supporting manifold plates they can be installed in practically any duct system regardless of space limitations. Supplied in two types with ½" or ¾" carbon bed thickness, each canister will purify from 25 to 30 cfm. Standard construction is steel but stainless steel and plastic are also available for highly corrosive conditions. Bulletin 108.



TYPE R PANELS

The Type R Panel is a semi-disposable activated carbon filled panel designed for installation with standard dust filters. Carbon may be easily replaced when saturated. All metal galvanized construction, four sizes, 16" x 20", 16" x 25", 20" x 20", and 20" x 25" for range of 640-1000 cfm. Bulletin 107.

CONNOR ENGINEERING CORPORATION

Connor Engineering Corp., Danbury, Connecticut

CONNOR ENGINEERING CORP. SALES REPRESENTATIVES AND ENGINEERING OFFICES

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South Africa, Johannesburg
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Airfoil

AIR CONDITIONING OUTLETS

by

TITUS

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Bulletin CGL-1R

SIDEWALL SUPPLY REGISTERS & GRILLES, VOLUME CONTROLLERS

TITUS SUPPLY GRILLES

STYLE	APPLICATION	LOUVER DESIGN	LOUVER SPACING	APPEARANCE	FRAME	DAMPER	DAMPER CONTROL	Approx. Free Area
L-271	Sidewall. 2-way directional control. Up or down.	¾" Airfoil. Solid Section. Extruded aluminum.	¾" on centers.	One set individually adjustable louvers parallel long dimension.	¼" Wide beveled edge. 18 or 20 ga. steel	None	None	80%
S-271	Sidewall. 2-way directional control. Right or left.	Same as above.	Same as above.	Same as above except louvers parallel to short dimension.	Same as above.	None	None	80%
L-270	Sidewall. 4-way control. Up, down, right or left.	Same as above.	Same as above.	Two sets louvers. Front parallel long dimension. Rear parallel short dimension. Individually adjustable.	Same as above.	None	None	80%
S-270	Sidewall. 4-way control. Right left, up or down.	Same as above.	Same as above.	Same as above except front parallel short dimension. Rear parallel long dimension.	Same as above.	None	None	80%

TITUS SUPPLY REGISTERS

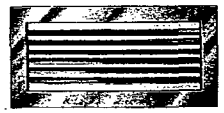
L-274	Sidewall. 2-way directional control. Up or down plus shutoff.	¾" Airfoil. Solid Section. Extruded aluminum.	Front louvers .. ¾" on centers.	Front louvers parallel long dimension. Individually adjustable.	1¼" ¼" Wide beveled edge. 18 or 20 gauge steel.	Multi-shutter .. spaced on 1" centers.	Standard lever operator. Remov. lever available at no extra cost. Specify R.L.L.	80%
S-274	Sidewall. 2-way directional control. Right or left plus shutoff.	Same as above.	Same as above.	Front louvers parallel short dimension. Individually adjustable.	Same as above.	Same as above.		80%
L-275	Sidewall. 2-way directional control. Up or down plus opposed action volume control.	Same as above.	Same as above.	Front louvers parallel long dimension. Individually adjustable.	Same as above.	AG-35 opposed action.	Removable Allen wrench operator.	80%
S-275	Sidewall. 2-way directional control. Right or left plus opposed action volume control.	Same as above.	Same as above.	Front louvers parallel short dimension. Individually adjustable.	Same as above.	Same as above.	Same as above.	80%
L-276	Sidewall. 4-way directional control. Up, down, right or left plus shutoff.	Same as above.	Front two sets .. ¾" on centers.	Two sets individually adjustable louvers. Front parallel long dimension. Second set parallel short dimension.	Same as above.	Multi-shutter .. spaced on 1" centers.	Standard lever operator. Remov. lever available at no extra cost Specify R.L.L.	80%
S-276	Same as above.	Same as above.	Same as above.	Same as above except front set parallel short dim. and second set — long dim.	Same as above.	Same as above.		80%
L-277	Sidewall. 4-way directional control. Up, down, right or left. Opposed action volume control.	Same as above.	Same as above.	Two sets individually adjustable louvers. Front set parallel long dimension. Second set parallel short dimension.	Same as above.	AG-35 opposed action.	Removable Allen wrench operator.	80%
S-277	Same as above.	Same as above.	Same as above.	Same as above except front set parallel short dim. and second set — long dim.	Same as above.	Same as above.	Same as above	80%

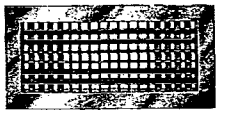
TITUS SIDEWALL & CEILING GRILLES

L-201	Sidewall or ceiling. Heating or cooling. 1-way directional air control.	Airfoil. Curved design. Extruded aluminum.	¾" on centers.	One set individually adjustable louvers parallel long dimension.	1¼" ¼" Wide beveled edge. 18 or 20 gauge steel.	Optional (See Note A)		80%
S-201	Sidewall or ceiling. Heating and cooling. 1-way directional air control.	Same as above.	Same as above.	One set individually adjustable louvers parallel short dimension.	Same as above.	Optional (See Note A)		80%
L-202	Sidewall or ceiling. Heating and cooling. 2-way directional control.	Same as above.	Same as above.	One set individually adjustable louvers parallel long dimension.	Same as above.	Optional (See Note A)		80%
S-202	Sidewall or ceiling. Heating and cooling. 2-way directional.	Same as above.	Same as above.	One set individually adjustable louvers parallel to short dimension.	Same as above.	Optional (See Note A)		80%
No. 203	Primarily where grille must be located close to wall. 3-way directional.	Same as above.	Same as above.	Left and right side louvers parallel short dimension. Center louvers parallel long dimension. Individually adjustable.	Same as above.	Optional (See Note A)		80%
No. 204	Ceiling — primarily in center of room. 4-way directional.	Same as above.	Same as above.	Left and right side louvers parallel short dimension. Center louvers parallel long dimension. Individually adjustable. (See cut.)	Same as above.	Optional (See Note A)		80%

TITUS VOLUME CONTROLLERS

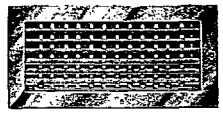
AG-25	In duct behind grille or at junction of duct and branch takeoff.	Airfoil. Solid Section. Extruded aluminum louver.	1½" on centers.	1⅛" each section. 2¼" total. Hinged for greater adjustability each section. Individually adjustable.	Channel type. 20 gauge cold rolled steel.		None	
AG-35	Attached to grille or in duct behind grille.	20 gauge steel.	1" on cent. Pivoted on center. Opposed acting	Gang operation. Full open to full closed. Blades always parallel short dimension.	Same as above.		Removable Allen wrench operator.	
AG-45	Attached to duct behind grille or at junction duct and branch takeoff.	22 gauge steel.	1" on center.	Double duty. Can move from full open to full closed. Blades always parallel short dimension.	None		Several types remote operators. Write Main OR.	


• L-271 — Supply Grille

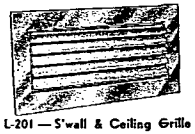

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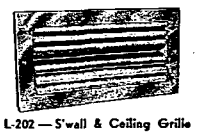

• L-274 — Supply Register


• L-276 — Supply Register

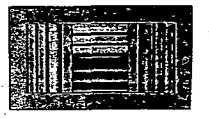

• L-275 — Supply Register

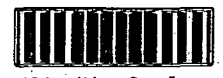

• L-277 — Supply Register

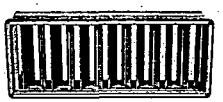

• L-201 — S'wall & Ceiling Grille

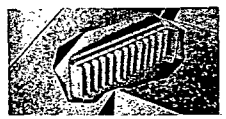

• L-202 — S'wall & Ceiling Grille


• 203 — Sidewall & Ceiling Grille


• 204 — Sidewall & Ceiling Grille


• AG-25 — Volume Controller

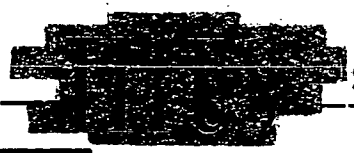

• AG-35 — Volume Controller


• AG-45 — Volume Controller

NOTE (A)

TITUS SIDEWALL & CEILING GRILLES

1. Available with opposed blade damper. Specify 2015, 2025, 2035, 2045.
2. Available with multi-shutter damper. Specify 2014, 2024, 2034, 2044.
3. Available with rear set individually adjustable aluminum louvers. Specify 2013, 2023, 2033, 2043.



SIDEWALL RETURN AIR REGISTERS, GRILLES and ACCESSORIES

TITUS RETURN AIR GRILLES

STYLE	APPLICATION	LOUVER DESIGN	LOUVER SPACING	APPEARANCE	FRAME	DAMPER	DAMPER CONTROL	Approx. Free Area
RL-230	Sidewall. To match all Titus Airfoil supplies.	Heavy steel. Curved and hemmed for added strength.	¾" on center.	Blades fixed at 45° or 0° deflection. Parallel long dimension.	¼" × 1¼" Wide beveled edge. 18 or 20 gauge steel.	None	None	82%
RS-230	Same as above.	Same as above.	Same as above.	Same as above except parallel to short dimension.	Same as above.	None	None	82%

TITUS RETURN AIR REGISTERS

RL-240	Sidewall. Return air plus multi-shutter damper.	Heavy steel. Curved and hemmed for added strength.	¾" spacing in front.	Front blades fixed at 45° or 0°. Parallel long dimension.	¼" × 1¼" Wide beveled edge. 18 or 20 gauge steel.	Multi-shutter on 1" cent. Parallel short dimen.	Standard lever operator. Remov. lever available at no extra cost. Specify R.L.L.	82%
RS-240	Same as above.	Same as above.	Same as above.	Same as above except parallel to short dimension.	Same as above	Same as above. Except parallel long dimension.		82%
RL-241	Sidewall. Return air plus opposed action damper.	Same as above.	Same as above.	Front blades at 45° or 0° deflection. Parallel long dimension.	Same as above	AG-35. opposed action.	Removable Allen wrench operator.	82%
RS-241	Same as above.	Same as above.	Same as above.	Same as above except parallel to short dimension.	Same as above	Same as above.	Same as above.	82%

TITUS DOOR GRILLES

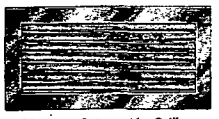
T-700A	Exhaust and return air grilles for doors, walls or partitions.	V-Shaped to prevent "see thru."	½" on centers.	Closely spaced V-shaped louvers parallel to long dimension.	Channel type. 20 ga. cold rolled steel.	None	None	56%
T-700B	Same as above.	Same as above.	Same as above.	Same as above.	Flange type. 1-5/16" beveled edge.	None	None	56%
T-800	Exhaust and return air grille for Panel Type doors.	V-Shaped to prevent "see thru."	3/16" on centers.	Closely spaced V-shaped louvers parallel to long dimension.	Core only. No frame.	None	None	66%
T-800A	Same as above.	Same as above.	Same as above.	Same as above except has frame.	Channel type. 20 ga. cold rolled steel.	None	None	66%
T-800B	Same as above.	Same as above.	Same as above.	Same as above except has flange frame.	1¼" beveled edge.	None	None	66%

TITUS GYMNASIUM GRILLES

G-1	Sidewall in school gymnasiums.	14 gauge round edged steel blades.	⅜" on centers.	Grille face only. Vertical steel support bars on 6 inch centers.	¼" × 1¼" Extra wide 16 ga. steel border.	See G-2	See G-2	80%
G-2	Same as above except has volume controller.	Same as above.	Same as above.	Grille face combined with volume controller.	Same as above.	Individually adjustable blades.	None	80%

TITUS CONVECTOR GRILLES

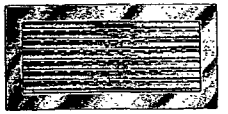
C-2100	Low sidewall.	20 gauge steel — hemmed edges.	½" on centers.	Available 1" blades deflected 45° on 12" centers or zero degrees on 6" centers.	¼" × 1¼" 16 gauge cold rolled steel.	None	None	70%
C-2200	Same as above except has multi-shutter damper blades for positive shut-off.	Same as above.	Same as above.	Same as above except provided with 1⅜" multi-shutter damper blades.	Same as above.	Multi-shutter spaced on 1" centers.	Lever knob operator on face of grille.	70%
C-2300	Low sidewall.	20 gauge steel — outside edges hemmed.	¾" on centers.	Available with 1" blades deflected 45° on 12" centers, or zero degrees on 6" centers. Specify.	Same as above.	None	None	70%
C-2400	Same as above except has multi-shutter damper blades for complete shutoff.	Same as above.	Same as above.	Same as above except provided with 1⅜" multi-shutter damper blades.	Same as above.	Multi-shutter spaced on 1" centers.	Lever knob operator on face of grille.	70%



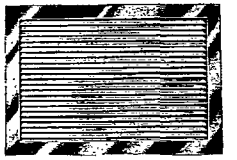
• RL-230 — Return Air Grille



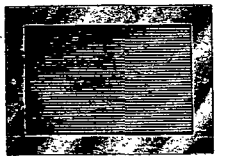
• RL-240 — Return Air Register



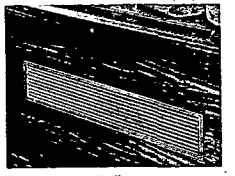
• RL-241 — Return Air Register



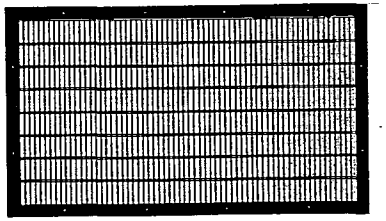
• T-700 — Door Grille



• T-800 — Door Grille



• Convector Grille



• Gymnasium Grille

NOTE THESE ADDITIONAL SPECIFICATIONS:

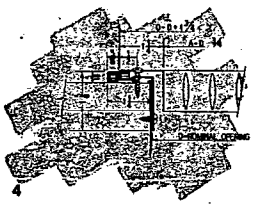
A. These added features are standard on each SUPPLY GRILLE.
1. ¼" sponge rubber gasket.
2. Countersunk screwholes.
3. Attractive softone gray lacquer finish.
4. Special mullion support where louvers exceed 20" in length.

B. VOLUME CONTROLLERS have standard black lacquer finish. Gaskets furnished on all AG-25 and AG-35s. Models are designed to fit into duct by friction.

C. RETURN AIR GRILLES are made extra heavy to withstand low sidewall abuse. Finished with standard gray softone lacquer.

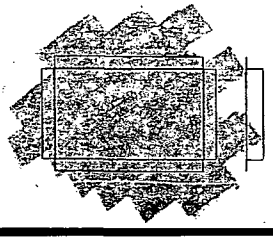
D. RETURN REGISTERS are made with extra heavy steel to withstand *low sidewall use and abuse. Provide volume control or* complete shutoff. Have gray softone prime finish.

TITUS ALL-PURPOSE FRAMES



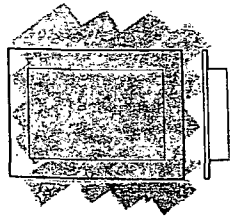
NO. 115 FRAME

For all Airfoil grilles and registers. Can be installed before or after plastering has been completed. Frame may be imbedded in plaster. Sponge rubber gasket to prevent air leakage. Designed for use with Titus Airfoil registers and grilles only.



NO. 2 STEEL ANGLE FRAME

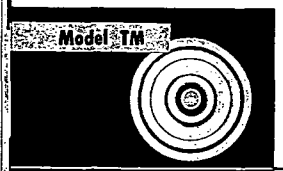
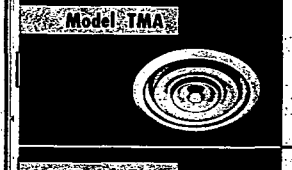
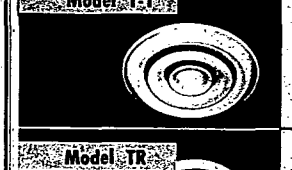
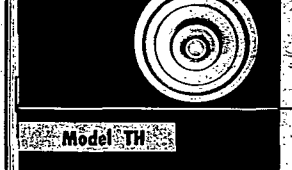
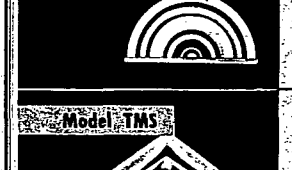
Attaches inside duct outlet with metal screws. Strengthens duct edge for easier and better plastering. Grilles fit inside frame. Grille border conceals lip of frame inside.



NO. 3 FRAME

For installation before or after plastering. No. 3 frame is built to actual listed size. Duct openings should be ⅛" larger to accommodate frame.

CEILING

STYLE	SIZES	APPLICATION	DESCRIPTION
Model TM 	6" 16" 8" 18" 10" 20" 12" 24" 14" 30" 36"	For exposed ductwork or flush mounting	A fixed cone ceiling outlet with exclusive Titus curved contour cone construction. 4 cones. Ingenious spring lock arrangement permits quick, easy removal or installation of inner cone assembly. Inner cone assembly adjustable at factory or on job to 3 different air patterns — vertical, horizontal, or intermediate.
Model TMA 	6" 16" 8" 18" 10" 20" 12" 24" 14" 30" 36"	For exposed ductwork or flush mounting	A fully adjustable outlet with Titus curved contour cone construction. Simply turn small center cone to adjust to any air pattern desired for any heating or cooling application. 3 center cones move in unison, retaining same relative position for maximum uniformity of design, minimum pressure loss. Inner cone assembly quickly, easily removed.
Model T-1 	6" 14" 8" 16" 10" 18" 12" 20" 24"	For exposed ductwork or flush mounting	A 3-cone supply outlet that combines economy with excellent diffusion performance. Inner cone assembly removes for fast, easy installation. Adjustable — on the job — to 2 air patterns by simply changing screws on inner cone assembly. Heavy-duty sponge gasket around outer edge of diffuser assures air-tight installations.
Model TR 	8" 18" 10" 20" 12" 24" 14" 30" 16" 36"	For exposed ductwork or flush mounting.	A combination air supply and exhaust diffuser for heating, ventilating and cooling. Especially designed for use where simplification of ductwork is desirable.
Model TH 	6" 14" 8" 16" 10" 18" 12" 20"	Primarily for ceiling application but can also be used on sidewall.	A half round supply air diffuser with removable inner cone assembly for quick, easy mounting. Adjustable — on the job — by simply removing inner cone assembly and adjusting screws to air pattern desired. Equally efficient for heating, cooling, ventilating.
Model TMS 	4" 12" 5" 14" 6" 15" 8" 16" 10" 18"	For exposed ductwork or flush mounting.	A square diffuser designed to deliver supply air for heating, ventilating, cooling in a 360° pattern. Blank-off baffles available to supply a wide variety of other air distribution patterns. Ingenious spring lock arrangement permits quick, easy removal and installation of inner assembly.

• Also available but not pictured is TITUS Model TL CEILING DIFFUSER (supply air diffuser combined with attractive lighting fixture). Write for literature

TITUS CEILING DIFFUSERS ARE:

A. Furnished in beautiful metalescent aluminum finish. (Unless otherwise specified)

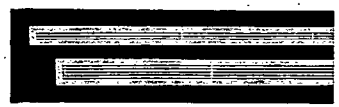
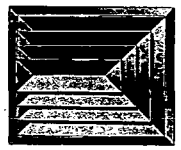
B. Individually packaged in sturdy damage-proof cartons.

DIFFUSERS AND ACCESSORIES

		FOR USE WITH DIFFUSER MODEL	DESCRIPTION
MODEL AG-65 COMBINATION DAMPER AND GRID		TM TMS TMA T-1 TL	A combination equalizing grid and damper that is quickly, easily installed in take-off. Blade opens up directly into air stream permitting precision air volume control. Installs flush with duct take-off.
MODEL AG-75 RADIAL DEFLECTOR DAMPER		TM TMS TMA T-1 TL	Four sets of gang-operated blades distribute air evenly over diffuser or duct — completely eliminating need for equalizing grids. Unique design permits the precision metering of air flow to any degree desired from full open to complete shut-off. Heavy gauge, vibration-free metal blades hold desired settings in spite of air pressure variations. Quickly, easily adjustable from face of diffuser.
MODEL AG-85 BUTTERFLY DAMPER		TM TMS TMA T-1 TL	A butterfly damper that provides the ultimate in simple, easy, economical control of air volume. Friction pivots on heavy gauge metal blades keep blades at desired setting. Can be mounted directly on diffuser. Fully adjustable from face of diffuser.
MODEL AG-85S SUPPLY DAMPER		TR	A circular type supply damper used to distribute supply air uniformly around circumference of Titus TR combination supply and return diffuser. Also used to control volume of supply air when balancing system — and as a method for baffling unit to give correct air distribution in irregular spaces. Individually adjustable blades have friction pivots to keem them at desired settings.
MODEL AG-85R EXHAUST DAMPER		TR	A butterfly type damper used on Titus TR combination supply and return diffusers. Easily and effectively controls volume of return air when balancing the system. Heavy gauge metal blades have friction pivots to maintain desired settings.
MODEL EG EQUALIZING GRID		TM TMS TMA T-1 TL	Used to equalize the flow of air into the neck of diffuser or duct — and to give directional control of air flow whenever needed. Features famous Titus solid metal Airfoil louvers designed to give maximum air control with minimum noise and turbulence. Individually adjustable louvers spring-tensioned to rigidly maintain desired settings — and provide silent, rattle-free performance.
MODEL CC CEILING CONE (Pictured with Model TMA Diffuser)		TM TR TMA TL	An attractively styled, scientifically designed accessory that not only affords maximum protection to ceiling — but actually enhances appearance of diffuser unit used with it. Adapts Titus diffusers to flush mounting on exposed ductwork — or flush with ceiling.
MODEL PMR PLASTER MOUNTING RING		TM T-1 TMA TR TL	An ingenious combination plaster mounting ring and duct ring that greatly reduces cost, time, work of flush mounting. Fastens by neck to duct and serves as plaster stop. Automatically provides correct ceiling opening and centers diffuser to it. Eliminates cutting hole in plaster. Does away with extra duct-work — extra connections. Sturdy metal construction.
MODEL AG-75½ VOLUME CONTROLLER		TH	An opposed acting damper that serves as an air flow equalizer — as well as a volume control. Positive shut-off when desired.
DUCT RING		TM TR TMA T-1 TL	Made of heavy gauge steel. Available for exposed duct work and other applications.
SECTORIZING BAFFLES		Specify side for baffle.	Blanking baffles available for special applications.

Air Foil
AIR CONDITIONING OUTLETS

OTHER TITUS DIFFUSERS

LINEAR TYPE air diffusing grilles 	Designed for continuous line use in sidewalls, window sills, floors of banks, hospitals, schools, other public buildings. Equally efficient for cooling, heating, ventilating.	Extruded aluminum, 1½", 2", 2½", 3", 3½" and 4" widths. **Model C-2500**—louvers in front fixed at 0° deflection. **Model C-2615**—louvers in front fixed at 15° deflection. Choice of borders, dampers. Any width or length on special order.
Series **TMD** SQUARE AND RECTANGULAR AIR DIFFUSERS 	Provide efficient, draftless, noiseless air distribution from any ceiling or sidewall location. For standard installations, tile ceilings, drop frame installations, surface mountings.	Square or rectangular—1, 2, 3, or 4-way patterns. Standard type TMD or type TMDC with interchangeable cores and choice of 4 styles of mounting frames.
PERIMETER DIFFUSERS  Models P-125 and P-75	New advanced design with dual adjustment feature assures plenty of capacity and the correct air patterns for both heating and cooling. For use in any size room—install on inside or outside walls.	MODEL P-125—adjustable for heating, adjustable for cooling—simply move control. Built-in damper. Neutra-Tone finish. 24" in length. MODEL P-75—same as Model P-125 above except does not have heating or cooling adjustment feature.

NOTE: Titus Perimeter Diffusers are packed 12 to a carton with each diffuser individually packaged in strong, damage-proof cardboard sleeve.

Sales Representatives  *in all Principal Cities*

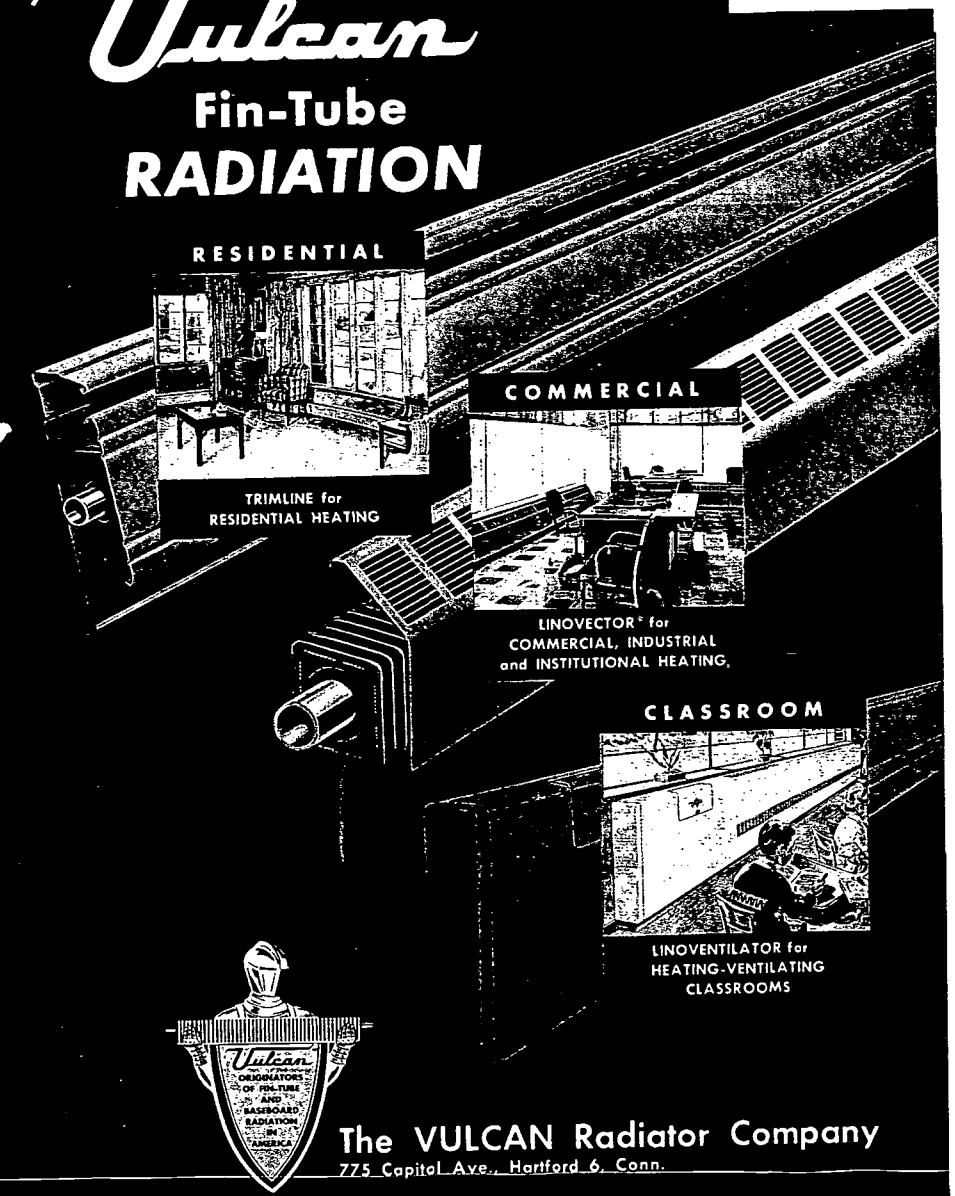
Akron, Ohio
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Atlanta, Ga.
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Baton Rouge, La.
Billings, Mont.
Birmingham, Ala.
Boise, Idaho
Boston, Mass.
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Charleston, W. Va.
Charlotte, N. C.
Chicago, Ill.
Cincinnati, Ohio
Cleveland, Ohio
Columbus, Ohio
Dallas, Texas
Davenport, Iowa
Dayton, Ohio
Denver, Colo.
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Detroit, Michigan
Fort Wayne, Ind.
Fort Worth, Texas
Grand Rapids, Mich.
Hartford, Conn.
Houston, Texas
Indianapolis, Ind.
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Jacksonville, Fla.
Kalamazoo, Mich.
Kansas City, Mo.
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Milwaukee, Wis.
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TITUS MANUFACTURING CORP. . . . . WATERLOO, IOWA

CATALOG No. 43R-3

Vulcan
Fin-Tube RADIATION

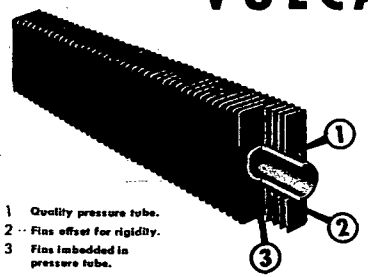


Vulcan ORIGINATORS OF FIN-TUBE AND BASEBOARD RADIATION IN AMERICA

The VULCAN Radiator Company
775 Capitol Ave., Hartford 6, Conn.

RESIDENTIAL

VULCAN TRIMLINE

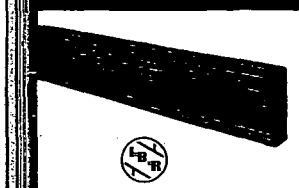


1 Quality pressure tube.
2 Fins offset for rigidity.
3 Fins imbedded in pressure tube.

TRIMLINE ELEMENTS

MATERIAL		SIZE		ENCLOSURE	
Style	Tube	Fins	Tube	Fins	Height
1000-A	Copper	Aluminum	1" Nom	2½" x 3¼"	9¼"
1000-B	Copper	Aluminum	¾" Nom	2½" x 3¼"	9¼"
97-A	Copper	Aluminum	1¼" Nom	2⅜" x 5"	12¼"
97	Steel	Steel	1¼" IPS	2⅜" x 5"	12¼"
64	Steel	Steel	1" IPS	2⅜" x 3¼"	9¼"

NOTE: Elements available in standard lengths from 2'0" to 12'0" in standard 1'0" increments.

TYPES OF RESIDENTIAL INSTALLATIONS	SPECIFICATIONS	ACCESSORIES
 TRIMLINE Hot Water or Steam Systems	Available in lengths up to 12'. One-piece back and top. Snap-on cover finished in prime gray or zinc coating. Can be curved (at factory) for special installations, at extra cost.	Rounded end enclosures, joining pieces and corner pieces. Full length positive position dampers.
 AIR-TRIM WARM AIR Baseboard Diffuser	Available in 2', 4', 6', 8' and 10' lengths with TRIMLINE design. Finished in prime gray or zinc coating.	Available with or without positive position damper.
 ELECTRIC TRIMLINE BASEBOARD	TRIMLINE enclosure design. Assembled in 3', 4', 5', 6', 8', 9', 11' and 13' lengths, with junction box. 110, 208 and 240V, 500 to 3500 watts output.	Standard TRIMLINE accessories available. For wall - to - wall installation, enclosure, without element, can be furnished to fill in.

VULCAN SERVICE IS NATIONAL . . .

Vulcan Service is available through more than 40 manufacturers' representative offices and hundreds of jobbers and contractors conveniently located throughout the nation. The Vulcan Service personel are trained heating experts. They are closely backed at the factory by Vulcan's top-flight engineering staff.

CATALOGS . . .

TRIMLINE Catalog
AIR-TRIM—AT 101
ELECTRIC TRIMLINE BULLETIN
Design MANUALS for all TRIMLINE products available upon request.

INDUSTRIAL

VULCAN LINOVECTOR

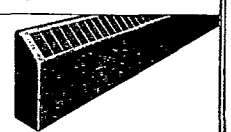
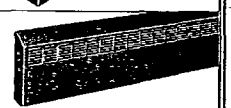
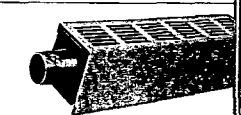
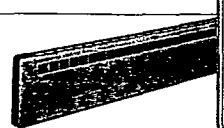
LINOVECTOR ELEMENTS

MATERIAL		SIZE	
Tube	Fins	Tube	Fins
Steel	Steel	1¼" IPS	3¼" x 3¼"
Steel	Steel	1¼" IPS	4¼" x 4¼"
Steel	Steel	2" IPS	4¼" x 4¼"
Copper	Copper or Aluminum	1¼" (1⅜" OD)	3¼" x 3¼"
Copper	Copper or Aluminum	1¼" (1⅜" OD)	4¼" x 4¼"

(Furnished in lengths up to 12 feet in increments of 1 inch)



Steel LINOVECTOR furnished threaded or chamfered for welding. Copper LINOVECTOR furnished for solder-joint fittings.

ACCESSORIES	SPECIFICATIONS	
Style X Grilles available in flexible segments for angular installations. Also, corner pieces, end enclosures, sloping and ceiling type grilles.	Expanded metal type, ½" diamond mesh (No. 16-18 gage steel). Black or ivory finish. Any lengths up to 8'.	STYLE X GRILLES 
End enclosures, "Flexi-Corner" (inside or outside), Flush Joints, Dampers, Access Panels and Enclosures for inverted mounting.	18-, 16- or 14-gage zinc-coated, rust-resistant, bonderized steel, or prime coat. Lengths up to 8'.	SLOPING TOP ENCLOSURE STYLE CS STYLE CSW DELUXE CS 
Full line available. Flush joints, "Flexi-Corners", Dampers and End Enclosures.	18-, 16- or 14-gage zinc-coated, rust-resistant, bonderized steel or prime coated. Furnished in lengths up to maximum of 8' in one piece. One unit front and top.	STYLE KS ENCLOSURES 
"Flexi-Corners", End Enclosures, and Joining Pieces (2½" wide for outside joining).	Solid front. 18-, 16- or 14-gage zinc-coated, rust-resistant, bonderized steel or prime coat finish. Lengths up to 8 feet.	STYLE F COVERS 
Full line of matching accessories, including all flush joints, rounded end enclosures and knob-operated dampers.	Available with top and front louvre outlets-or-either. 18-, 16- or 14-gage zinc-coated, rust-resistant, bonderized steel or prime coated. Lengths up to 8'.	SILL-TRIM ENCLOSURES 
Full line of matching accessories, including rounded end enclosures and knob-operated dampers.	Sturdy baseboard design. Full line of elements available. 18-, 16- and 14-gage zinc-coated, rust-resistant, bonderized steel or prime coated. Full back optional. Rugged cover support brackets.	INSTITUTIONAL BASEBOARD 

JOINTS — all enclosures shown above, except Style X and Style F, available with "H" joint (no screws or bolts needed).

LOUVRES — "Mini" (small size) louvres available only in 18- and 16-gage steel, in 2" increments. Standard louvres available in 18-, 16- and 14-gage steel in 3" increments.

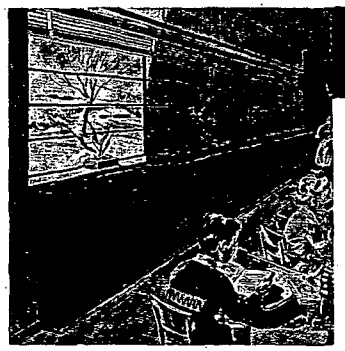
HEIGHT — all enclosures available in 1-, 2- and 3-row high, except Institutional Baseboard (only 2-row high).

BAKED ENAMEL — all enclosures, except Style X, available in 8 standard baked enamel colors (at extra cost). See Vulcan Color Guide.

LINOVECTOR CATALOG 625B available showing full line of VULCAN elements, accessories and enclosures, including complete ratings.

Vulcan LINOVENTILATOR*

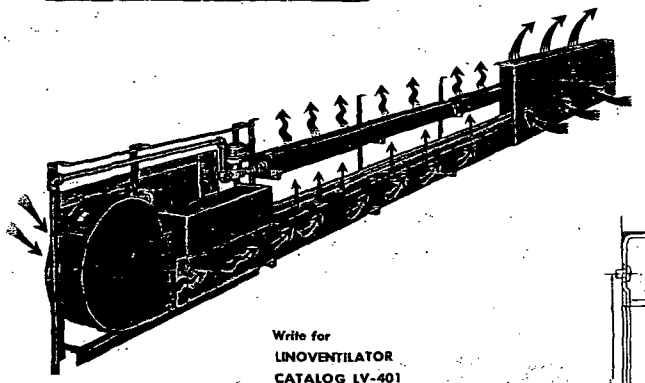
A Quality Schoolroom Heating-Ventilating System At Lowest Cost To Fit Almost Any Budget!



New Heating-Ventilating System Meets All Requirements . . .

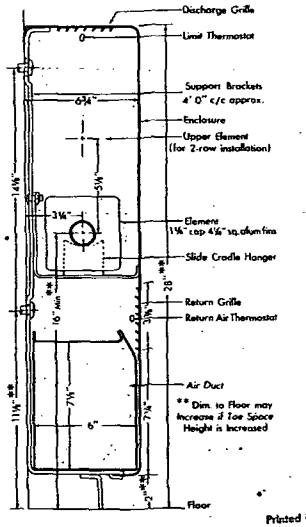
The completely new Vulcan LINOVENTILATOR SYSTEM is the latest innovation in classroom heating and ventilating. *It has been designed for economy, efficiency and functional beauty, with* remarkable savings in installation and operating costs. This has been made possible by unique engineering that introduces an entirely new concept in heating and ventilating.

The LINOVENTILATOR SYSTEM meets the exacting requirements for schoolroom heating-ventilating as established by recognized authorities. It now places heating-ventilating, heretofore impossible because of high cost, within the reach of local school budgets.



Write for
LINOVENTILATOR
CATALOG LV-401

DIMENSIONAL DATA



NEW ENGINEERING DESIGN

- NO DAMPERS REQUIRED
- LOW INSTALLATION AND OPERATING COSTS
- ELIMINATES NEED FOR AUXILIARY RADIATION
- SIMPLIFIED AUTOMATIC CONTROLS
- LOW MAINTENANCE COST
- NO WIND "BLOW-THROUGH"
- AVAILABLE IN 8 STANDARD BAKED ENAMEL COLORS

*Trademark Reg.

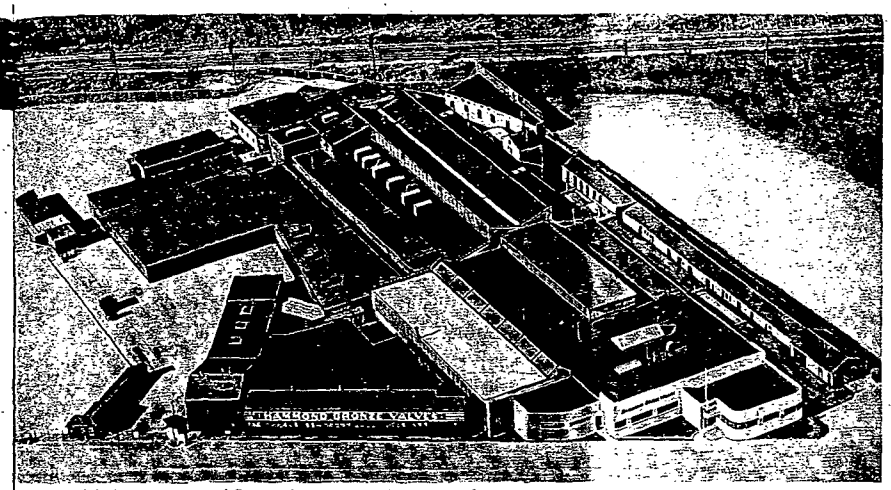
Bulletin No. 43R-3

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HAMMOND...

LEADER IN BRONZE VALVES



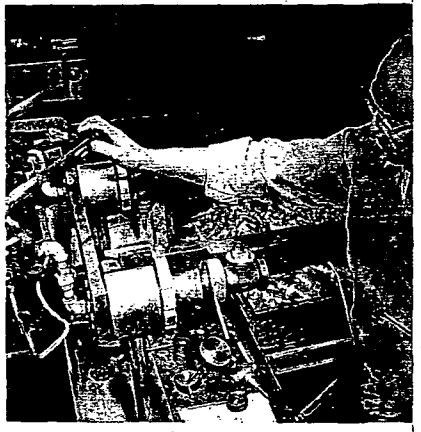
Aerial view of Hammond Brass Works . . . considered one of industry's outstanding examples of a highly efficient, completely integrated valve manufacturing plant.

Hammond Bronze Valves are produced in this modern factory building in Hammond, Indiana. Complete facilities for valve production include everything from foundries which pour the initial bronze castings down to specially developed testing equipment used to individually test each valve before it leaves the factory.

Hammond has been a leader in the bronze valve industry for over 47 years . . . and bronze valves are the only product of the Hammond Brass Works. Therefore, they receive the undivided attention of Hammond's engineering staff. This specialization in bronze valves is one of the principal reasons behind Hammond's leadership in the industry.

Hammond's production operations are distinguished by mass production, automatic methods . . . a fine example of "automation in action." These large volume automatic operations result in savings which are passed on to Hammond customers . . . allow Hammond to offer a premium in quality without any increase in price . . . for the "best valve and value in the bronze valve field."

Hammond valves are readily available from coast to coast through leading wholesalers. Forty-seven strategically located factory warehouse stocks insure up-to-date inventories.



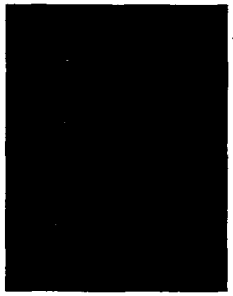
Heavy duty check valve being tested hydrostatically, the toughest test a valve can get. Every Hammond valve is tested before it leaves the factory.

HAMMOND BRASS WORKS HAMMOND, INDIANA

HAMMOND

HEAVY-DUTY VALVES

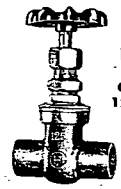
600 SERIES GATE VALVES





No. 609
N. R. S. GATE, 125 lb. WSP

With solid wedge disc. Complies with Federal Spec. No. WW-V-54, type 1, class A. May be repacked when fully open. Recommended for heavy-duty services up to 125 lb. w.s.p. and 200 lb. w.o.g. Available ¼" to 3". *Also available with drainable end. Specify on order.*



No. 613
N. R. S. GATE S. J., 125 lb. WSP

Construction features identical to No. 609, but with sweat end connections for use with copper tubing. Meets Federal Spec. No. WW-V-54, type 1, class A. Available ⅛" to 3".



No. 617 & No. 618
UNION BONNET GATE, 125 lb. WSP

Rising stem, solid wedge disc gate valve. Rated 125 lb. W.S.P. Meets MIL Spec. WW-V-54, Type 2, Class A. No. 618 features double disc, meets WW-V-54, Type 3, Class A. Sizes ½" to 3". *Both styles available in copper-to-copper as No. 617c and 618c.*



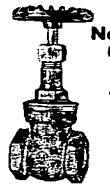
No. 619
N. R. S. GATE, 150 lb. WSP

With solid wedge disc. Meets Federal Spec. WW-V-54, type 1, class B. Large sized, generously proportioned, with extra strength for service requirements at 150 lb. w.s.p. and 300 lb. water, oil or gas. Available ¼" to 3".



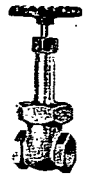
No. 622
BRAZING TYPE N.R.S. GATE, 150 lb. WSP

With construction features and ratings similar to No. 619, but with socket ends for brazing. Used for industrial construction and general piping where resistance to shock is of primary importance. Available ¼" to 3". *No. 615 available for 125 lb. w.s.p.*



No. 626
RISING STEM GATE, 150 lb. WSP

With double wedge disc. Sturdily designed for services to 150 lb. steam, 300 lb. w.o.g. Available ½" to 2".



No. 629 & No. 631
UNION BONNET GATE, 150 lb. WSP

Rising stem, solid wedge disc gate valve. Meets MIL Spec. WW-V-54, Type 2, Class B. No. 631 features double disc, meets spec. WW-V-54, Type 3, Class B. Sizes ½" to 2". *Both available in copper-to-copper as No. 629c and 631c.*

No. 611
MONEL STEM GATE, 100 lb. WSP

Non-rising monel stem solid wedge disc gate valve. Meets military specifications MIL-V-1189, type A. Rated for 100 lb. w.s.p. 200 lb. w.o.g. Available ¼" to 2".



No. 612C & No. 612
N. R. S. GATE, 200 lb. WSP

Heavy-duty, non-rising stem gate valve with solid wedge disc. Rated for 200 lb. w.s.p. Also available as Fig. 612 with monel trim to meet government specifications MIL-V-20231A (Ships). Available ¼" to 3".

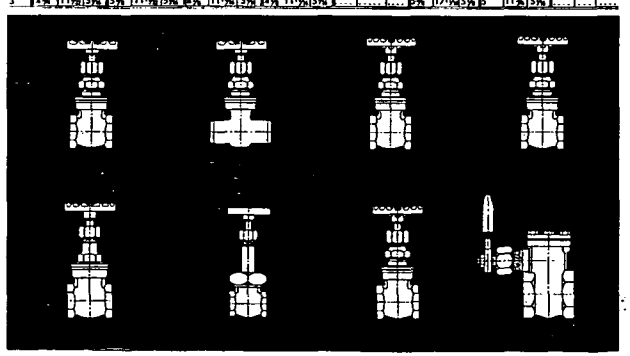


No. 610
QUICK OPENING GATE, 125 lb. WSP

Solid wedge disc is operated with quick opening lever. Meets military spec. MIL-V-12126 (CE), class 1. Rated for 125 lb. steam and 200 lb. non-shock on water, oil or gas lines. Available in ½", ¾", 1", 1¼", 1½" and 2" sizes.

ROUGHING-IN DIMENSIONS OF SERIES 600 GATE VALVES

	No. 609			No. 613			No. 619			No. 622			No. 626			No. 629 & 631			No. 612C			No. 610		
SIZE	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C
¼	1⅜	3⅜	1¾	.....	.....	.....	1⅜	3 11/16	2½	1⅜	3 11/16	2½	.....	.....	.....	1 11/16	3⅝	2½	1⅜	3 11/16	2½	.....	.....	.....
⅜	1⅜	3⅜	1⅜	2¼	3¼	1⅜	1 11/16	3⅜	2½	1 11/16	3⅜	2½	.....	.....	.....	2	5 5/16	2½	1 11/16	3⅜	2½	.....	.....	.....
½	1 13/16	3⅜	2 1/16	2¼	3¼	2 1/16	2 1/16	4⅝	2 13/16	2 1/16	4⅝	2 13/16	2⅜	5⅜	2 1/16	2 1/16	5⅝	2 1/16	2 1/16	4⅝	2 13/16	2	5 5/16	3 5/16
¾	2 5/16	4⅝	2 1/16	2 5/16	4⅝	2 1/16	2	5	2 13/16	2	5	2 13/16	2⅜	6 13/16	2 1/16	2 5/16	6	2 1/16	2 5/16	5	2 13/16	2 1/16	6	3 5/16
1	2 1/16	5⅛	2⅜	3⅜	5⅛	2⅜	2 1/16	5 1/16	3 1/16	2 1/16	5 1/16	3 1/16	2 1/16	7 1/16	3 1/16	2⅜	8	2⅜	2 1/16	5 1/16	3 1/16	3 1/16	6⅜	9 1/16
1¼	2 1/16	5⅝	3	3⅛	5⅝	3	2⅜	6 1/16	3⅜	2⅜	6 1/16	3⅜	3⅛	9 1/16	3⅜	3⅛	9½	3	2⅜	6 1/16	3⅜	3 1/16	6⅜	9 1/16
1½	2⅝	6⅜	2 15/16	3⅜	6⅝	3 15/16	2 1/16	7⅜	3⅜	3 1/16	7⅜	3⅜	3⅛	11 1/16	4⅜	3 1/16	10 5/16	3 1/16	3 1/16	7⅜	3⅜	3 1/16	6⅜	9 1/16
2	3⅛	7⅜	3 15/16	4 1/16	7⅜	3 15/16	4⅛	7⅜	4⅜	4⅛	7⅜	4⅜	3⅜	12 5/16	5	3⅜	12 5/16	3 1/16	4⅛	7⅜	4⅜	4	6	11⅞
2½	3 1/16	9 1/16	4⅜	5	9⅜	4⅜	3 1/16	9 1/16	4⅜	3 1/16	9 1/16	4⅜	.....	.....	.....	4 1/16	13⅜	4 1/16	4 1/16	9 1/16	4⅜	.....	.....	.....
3	4⅜	11 7/16	5⅜	5⅜	11 7/16	5⅜	4⅜	11 1/16	5⅜	4⅜	11 1/16	5⅜	.....	.....	.....	5⅜	17 11/16	5 1/16	5	11⅜	5⅜	.....	.....	.....



HAMMOND

HEAVY-DUTY VALVES

400 SERIES GLOBE VALVES



No. 410
BRONZE DISC GLOBE, 125 lb. WSP

Sturdily constructed. All sizes have swivel discs—¾" size and larger have renewable discs. Meets all service requirements up to 125 lb. working steam pressure, 200 lb. water, oil or gas. Available ⅛" to 2".



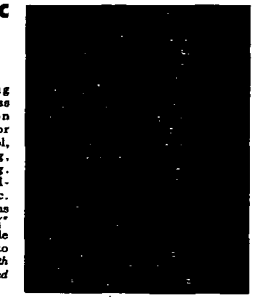
No. 410D
UNION BONNET GLOBE, 125 lb. WSP

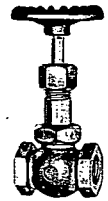
Extra strong brass disc globe valve with union bonnet. Meets Federal Spec. WW-V-51A, type 1, class A. Ratings and sizes same as No. 410.



No. 411 & 411C
REGRINDING GLOBE, 200 lb. WSP

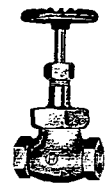
With plug type brass disc, union bonnet. For close control, throttling, regulating. Meets military spec. MIL-V-1202-A, type A. Stems of monel. Has gland follower. ¾" size and larger have renewable and swivel discs. Sizes ¼" to 2". *Available as No. 411C with high tensile bronze stem instead of monel.*





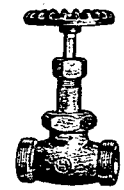
No. 412 and 412C
REGRINDING GLOBE, 300 lb. WSP

Construction features similar to No. 411 but more heavily constructed for all types of service up to 300 lb. steam, 600 lb. water, oil or gas. Monel stem and disc. Meets military spec. MIL-V-1187, type A. Sizes ¼" to 2". *Also available with high tensile bronze stem and disc instead of monel as No. 412C.*



No. 413
COMPOSITION DISC GLOBE, 150 lb. WSP

Has slip-on type disc holder for quick disc replacement, stuffing box, gland follower, union bonnet. Complies with Federal Spec. WW-V-51a, type 1, class B. For industrial steam service up to 150 lb. steam, 300 lb. w.o.g. Sizes ⅛" to 2". *Also available in copper-to-copper as No. 423.*

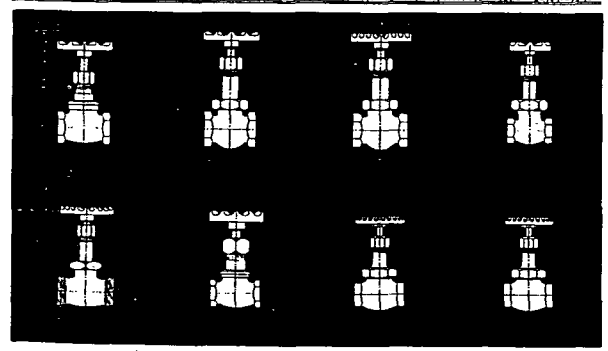


No. 414
MONEL DISC GLOBE, 200 lb. WSP

With double union ends, meets Bureau of Ships spec. B-164, alt. 7. Has monel seat ring, union bonnet. Double union ends used for silver brazing. Primarily for ship board use. Rated 200 lb. steam, 400 w.o.g. Available ¼" to ¾". *Also available in angle type, No. 460, ¼" size only.*

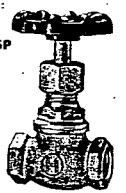
ROUGHING-IN DIMENSIONS SERIES 400 GLOBE VALVES

	No. 410			No. 411 & 411C			No. 412 & 412C			No. 413			No. 414			No. 415			No. 434			No. 435		
SIZE	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C	A	B	C
¼	1⅜	2⅜	1⅜	.....	.....	.....				.....	.....	.....	.....	.....	.....	1 15/16	2 11/16	1⅜	.....	.....	.....	.....	.....	.....
⅜	1 5/8	2⅜	1⅜	1 15/16	3 5/16	1⅜	1 15/16	4⅛	2⅜	2⅜	4⅛	2⅜	1⅜	2⅜	1⅜	2⅛	4⅛	2⅜	2⅛	4⅛	2⅜	2⅛	4⅛	2⅜
½	1 15/16	3	1⅝	2 1/16	4⅛	2⅛	2 1/16	4⅛	2⅛	2⅜	4⅛	2⅛	1⅝	2⅝	1⅝	2⅛	4⅛	2⅝	2⅛	4⅛	2⅝	2⅛	4⅛	2⅝
¾	2 5/16	3 1/16	2	2 5/16	4 5/16	2 1/16	2 5/16	4 5/16	2 1/16	2 5/16	4 5/16	2 1/16	2	2⅝	2	2 5/16	4 5/16	2 1/16	2 5/16	4 5/16	2 1/16	2 5/16	4 5/16	2 1/16
1	2⅝	4⅛	2⅜	2 5/16	4 5/16	2 1/16	2 5/16	4 5/16	2 1/16	2 5/16	4 5/16	2 1/16	2⅜	3⅛	2⅜	2⅝	4⅛	2⅜	2⅝	4⅛	2⅜	2⅝	4⅛	2⅜
1¼	3⅛	5⅛	2⅞	3 5/16	5 5/16	2 13/16	3 5/16	5 5/16	2 13/16	3 5/16	5 5/16	2 13/16	2⅞	3⅞	2⅞	3⅛	5⅛	2⅞	3⅛	5⅛	2⅞	3⅛	5⅛	2⅞
1½	3 1/16	5 1/16	3⅛	3 5/16	7 1/16	4⅛	3 5/16	7 1/16	4⅛	3 5/16	7 1/16	4⅛	3⅛	4⅝	3⅛	3 1/16	7 1/16	4⅛	3 1/16	7 1/16	4⅛	3 1/16	7 1/16	4⅛
2	4 1/16	6 1/16	3 15/16	4⅛	7 1/16	4⅛	4 1/16	7 1/16	4⅛	4 1/16	7 1/16	4⅛	4 1/16	5⅝	4 1/16	4 1/16	7 1/16	4⅛	4 1/16	7 1/16	4⅛	4 1/16	7 1/16	4⅛
2½	4⅝	6 7/16	4⅝	5⅜	9⅛	4⅝	5⅜	9⅛	4⅝	5⅜	9⅛	4⅝	5⅜	7⅛	4⅝	.....	.....	.....	5	9⅛	5⅛	5	9⅛	5⅛
3	5⅜	7 7/16	5⅜	6	10⅛	5⅛	6	10⅛	5⅛	6	10⅛	5⅛	6	8⅛	5⅛	.....	.....	.....	6 1/16	10⅛	6	6	10⅛	6



No. 415
NEEDLE VALVE, 200 lb. WSP

Produced to Federal Specification, meets requirements of 200 lb. working steam pressure. Available in ¼", ⅛", ⅜", ½", ¾" sizes.



No. 435C
"500 BRINELL" GLOBE VALVE PLUG TYPE

With "500 Brinell" hardened stainless steel disc and renewable seat. Rated 300 lb. steam, 600 lb. w.o.g. *Available ¼" to 2". No. 433 and 434, 150 lb., 200 lb., designs of No. 435C.*

No. 435, "AAR" globe, meets MIL-V-1187B, type 2.

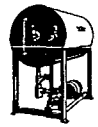


HAMMOND INDUSTRIAL VALVE COMPARATIVE FIGURE NUMBER REFERENCE CHART

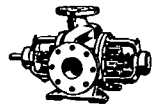
FIG. NO.	DESCRIPTION									
107	150# Radiator Angle JAN-V-189 Type C	67-G	—	—	168	—	—	95	—	—	—
404	125# Globe, Renewable Steam, Composition Disc	1240	—	—	—	—	—	97	—	—	—
410	125# Globe, Brass Disc WW-V-51A, Type I, Class A	1	58	2140	—	234	650	150	011	B-16	045
411-C	200# Globe, Regrinding Mil-V-1202A Type A	70 Globe	160	407	750	270	110	134	409	B-37	029
412-C	300# Globe, Regrinding Mil-V-1187, Type A	362-E	205	409	1130	370	120	—	427	B-66	033
413	150# Globe, Renewable Composition Disc WW-V-51A, Type I, Class B	7 Globe	95	123	106-A	520	150	89	1A	B-22	U-01
414	200# Globe, Regrinding Double Union Bushing Dwg. B-164 Alt. 7	—	154 W.S.	—	—	—	—	—	—	—	—
415	200# Globe, Needle Point	88	120	906	743-G	238	180	—	—	B-64	041
431	200# Globe, Convector	4270	—	—	—	—	—	—	—	—	—
433	150# Globe, Plug Type	14½P	—	—	546P	—	—	—	—	B-29	0525
434	200# Globe, Plug Type	212P	245P	73PS	556P	558	1708	138P	409P	B-62	0501P
435	300# Globe, AAR Plug Type Mil-V-11678, Type II	376P	256P	1690P	—	700P	—	5445	—	B-78	—
435-C	300# Globe, Plug Type	382P	260P	16PS	576P	670	1710	544P	427P	B-74	0545
453	125# Angle, Brass Disc WW-V-51A, Type II, Class A	2	59	2141	—	235	651	151	033	B-216	046
454	150# Angle, Renewable Composition Disc WW-V-51A, Type II, Class B	7 Angle	96	214	108-A	521	151	90	3A	B-222	U-03
455-C	300# Angle, Regrinding Mil-V-1187, Type B	364-E	206	557	1132	371	122	—	429	B-266	035
458-C	200# Angle, Regrinding Mil-V-1202A, Type B	70 Angle	161	408	752	271	112	135	411	B-237	031
600	125# Gate, Double Disc Rising Stem	440	—	—	—	95	—	—	—	B-114	—
609	125# Gate, Solid Disc NRS — WW-V-54, Type I, Class A	438	4	2129	370	99	507	27	86	B-115	0250
610	125# Gate, Quick Opening Mil-V-12126 (CE) Class I	432	9	—	302	106	362	—	—	B-116	—
612-C	200# Gate, Solid Disc, NRS Mil-V-20231A (Ships) Class II	458	—	2230	270	241	380	37	—	B-138	—
613	125# Gate, Solid Disc, NRS Solder Ends WW-V-54, Type I, Class	1320	—	—	1240	—	1822	427S.J.	90	—	0280
615	125# Gate, Solid Disc, NRS, Brazing	—	4 W.S.	—	1273	—	—	427B.S.	—	—	0250F.B.
617	125# Gate, Solid Disc, Rising Stem Union Bonnet WW-V-54, Type II, Class	428	2	2127	47-U	97	500	25	86Y	B-105	0252
618	125# Gate, Double Disc, Rising Stem Union Bonnet WW-V-54, Type III, Class	430	3	2125	62-U	100	500	—	86X	B-106	0253
619	150# Gate, Solid Disc, NRS WW-V-54, Class B, Type I	437	14	2153	670	98	512	23	103	B-130	0224
622	150# Gate, Solid Disc, NRS, Brazing	—	4 W.S.	—	1273	—	—	23 B.S.	—	—	0224F.B.
626	150# Gate, Double Disc, RS	431	12	2150	64	96	514	45	103A	B-121	0228
629	150# Gate, Solid Disc, Union Bonnet, RS WW-V-54, Type II, Class B	431	11	2151	49-U	750	514	—	784	B-120	U-0226
631	150# Gate, Double Disc, Rising Stem Union Bonnet Mil-WW-V-54, Type III, Class	—	12	—	64-U	7152	514	—	705	B-121	0228
901	125# Check, Lift	20	73	2142	—	237	658	133	—	B-316	0607
902	125# Check, Vertical	24	75	418	119	—	—	490	—	—	—
904	125# Check, Swing WW-V-51A Type IV, Class A	34	406	2144	92	134	578	103	198	B-319	0600
906	125# Check, Swing, Brazing Mil-V-17547 (Navy) Type A, Class I	—	—	—	1272	—	—	—	—	—	0601F.B.
909	150# Swing Check, Composition Disc WW-V-51A, Type IV, Class B	34½	412	—	352	—	596	—	205	—	—

HAMMOND BRITE-KOTE® BRONZE VALVES

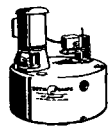
HAMMOND BRASS WORKS HAMMOND, INDIANA



PACKAGED BOILER FEED UNITS — PAGES 2-3-4-5



2-STAGE POWER PLANT PUMPS — PAGES 6-7



VERTICAL CONDENSATE UNITS — PAGE 8

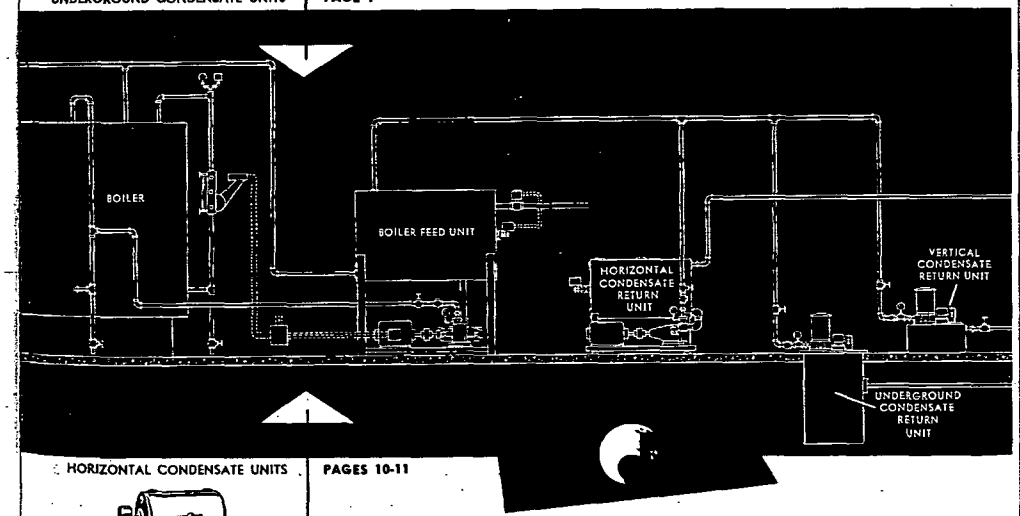


UNDERGROUND CONDENSATE UNITS — PAGE 9

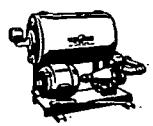
ENGINEERS' GUIDE TO

ROTH

PACKAGED BOILER FEED & CONDENSATE RETURN UNITS FOR INDUSTRIAL & COMMERCIAL INSTALLATIONS COMPLETE WITH SELECTION DATA & SPECIFICATIONS



HORIZONTAL CONDENSATE UNITS — PAGES 10-11



ROY E. ROTH COMPANY TURBINE PUMP DIVISION ROCK ISLAND, ILLINOIS

2U58

ROTH

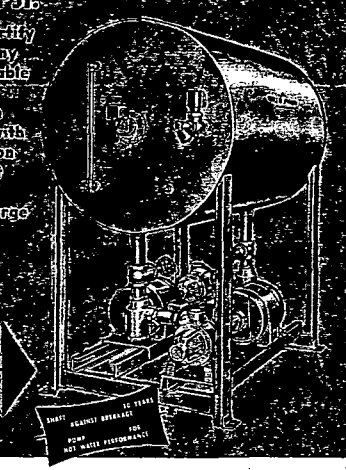
PACKAGED BOILER FEED UNITS

DUPLEX MODELS

Certified Hot Water Performance in Factory Assembled Systems for Boilers of 40 to 500 HP Operating at Pressures up to 200 PSI.

The Roy E. Roth Company will certify the hot water performance of any model taken from the selection table (page 4) at the stated capacity and pressure listed therein when such certification is requested with the purchase order. Certification will be based upon a special hot water test run upon the pump ordered. A small additional charge will be made to cover the cost of the hot water test.

All pumps on these units will deliver at least 150% of rated evaporation at 210°F.



ROTH turbine boiler feed pumping units are a line of factory assembled systems for boilers from 40 to 500 HP operating at pressures up to 200 PSI. *Factory assembly reduces costly erection labor on the job site and insures factory responsibility for the proper selection and function of the entire unit.*

ROTH boiler feed units consist of one or more motor driven pumps combined with a hot water storage tank or receiver and an automatic make-up valve to replace system losses. These units are used with most fire tube and scotch marine boilers when operated for process steam at pressures above 20 PSI. They are also used in modified form with water tube boilers.

The motor driven pump operates intermittently and is controlled by a special float switch or pump controller mounted on the boiler water column, starting the pump at low water level and stopping it at high water level.

Water pumped from the vented receiver is usually at least 70% hot condensate and varies from 180°F up to boiling point, depending on the amount of flash steam returned.

Boiler feed pumps with low suction head tend to lose capacity as water temperature approaches 212°F. By lowering the suction head to zero and raising the water temperature to 212°F it is possible to vapor lock any turbine or centrifugal pump. For this reason the buyer should be protected against hot water fall-off by performance guarantees in which suction head and temperatures are clearly stated as in the selection table (see page 4). *All pump models furnished with ROTH Boiler Feed Units are thoroughly charted for performance in handling water close to boiling point and are rated for their capacity at 206°F.*

In addition to certified hot water performance, ROTH engineering has resulted in many other design improvements assuring long, dependable operation. By eliminating ball bearing end-play, impeller position is permanently maintained. *The new ROTH pumps have heavier shafts than any previously manufactured equipment and are covered by a written guarantee (see page 12).*

Other components of ROTH Boiler Feed Units are manufactured with equal care. Welded steel receivers are of 3/16 inch copper-bearing steel and internally coated with a rust-resistant phenolic resin lining. Large diameter piping between receiver and pump eliminates pressure drop and includes a large strainer especially made for use on these units. Mechanical make-up valve for receivers up to 100 gallon capacity is float operated. A solenoid valve controlled by a float switch is supplied for units with 180, 250, 350, and 500 gallon receivers.

Available at additional cost are the following items:

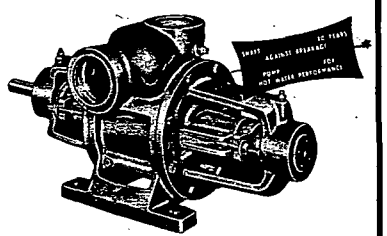
(1) Separable tank cradles instead of welded legs available on all sizes.

(2) Dished head receivers made to 75 lb. ASME specifications, with or without code stamp can be substituted for 3/16 inch steel flat head receivers.

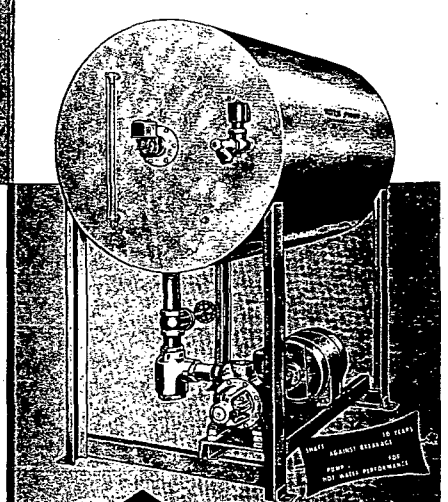
(3) Perforated tube heaters with manual or automatic temperature-controlled steam valves.

(4) Magnetic starters, alarm bells, pressure gauges, and signal lights.

(5) Motorized valve in place of solenoid valve.



ROTH TWO-STAGE TURBINE POWER PLANT PUMPS delivering up to 200 PSI at 1750 RPM are standard on this line of equipment. Large passages between stages, special suction entrance modification, and the inherent vapor handling characteristics of the ROTH turbine impeller contribute to this pump's ability to efficiently handle hot water at high pressure differential and low suction head.



SIMPLEX MODELS

ROTH Boiler Feed Units may be had in simplex or duplex models either as packaged or erected down units. Both are engineered and manufactured with the full and precision characteristics of all ROTH equipment.

SPECIFICATION FORM

Furnish and install as called for on the drawings one Roth Model ..... (specify simplex or duplex) boiler feed unit suitable for ..... GPM at a discharge pressure of ..... PSI. The unit shall consist of the following components:

(1) One (substitute two for duplex) Roth Model ..... standard fitted power plant pump certified by the manufacturer for a minimum of ..... GPM of 206°F water at ..... PSI with 4 ft. suction head. The pump shall be guaranteed by the manufacturer for one year against defects in workmanship and material and shall be guaranteed for ten years by the manufacturer against shaft breakage when flexible coupled to a motor of recommended rating. The pump shall contain an internal system for relief of pressure on the packing to insure maximum packing life.

(2) One non-code welded steel receiver made of 3/16" thick copper-bearing steel with phenolic resin lining, integral supports, and connection for inlet, outlet, drain, vent, make-up valve, thermometer, and gauge glass. (Substitute if desired one dished head receiver built to ASME code for unfired pressure vessels rated 75 lb. working pressure with or without ASME inspection stamp but with separable cradle supports and connections for inlet, outlet, vent, drain, thermometer, gauge glass and make-up valve also with hand hole or manhole for cleanout per ASME code depending on the diameter of the receiver.)

(3) One fresh water make-up valve consisting of a ¾ inch solenoid valve (specify 1 inch on 500 gallon receivers) controlled by a float switch mounted on the receiver for ..... phase ..... volt current, (substitute ¾" float operated mechanical valve with pressure reducing valve set for 15 PSI for units with 60 and 100 gallon receivers).

(4) Piping between the receiver and pump to consist of one (substitute two for duplex) vertical section containing a gate valve and one (substitute two for duplex) horizontal section connected by Roth L-Type strainer to insure minimum pressure drop between receiver and pump.

(5) One (substitute two for duplex) 1750 RPM drip proof motor for ..... phase ..... volt current. Motor shall not be loaded in excess of NEMA service factor for integral drip proof motors. Make shall be General Electric, Louis Allis, Howell or equal.

(6) One (substitute two for duplex) magnetic starter of approved type suitable for the above motor, with holding coil for ..... volts.

(7) Gauge glass of proper size.

(8) Tank thermometer.

(9) When provision for heating water in the receiver is required add, "One brass perforated tube heater installed in the receiver." When automatic temperature regulated heater is required specify, "One brass perforated tube heater installed in the receiver connected to one temperature controlled regulating valve mounted on the heater connection."

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*SEE PAGE TWELVE

*SEE PAGE TWELVE

ROTH SELECTION TABLE

Boiler Feed Units with 1750 RPM Power Plant Pumps

Boiler H.P.	Evap. GPM	Evap. Lbs./Hr.		80	100	125	150	200	Receiver Data Cap.	Receiver Data Dia.	Receiver Data Lgth.	Make-Up Valve Mech.	Make-Up Valve Elec.	Unit Data Pipe	Unit Data Width	Unit Data Length	Unit Data Height
41 to 50	3.5	1725	Unit No. Pkg. Motor HP GPM Cert. 206 @	XE51 1 11.0 9.1	XE52 1½ 9.9 8.0	XE52 1½ 8.6 6.7	XE53 2 7.5 5.5	XE55 5 10.5 9.3	60	22½	36	¾		1½	24	41	60
51 to 60	4.2	2070	Unit No. Pkg. Motor HP GPM Cert. 205 @	XE51 1 11.0 9.1	XE52 1½ 9.9 8.0	XE52 1½ 8.6 6.7	XE54 3 13.5 12.3	XE55 5 10.5 9.3	60	22½	36	¾		1½	24	41	60
61 to 80	5.5	2760	Unit No. Pkg. Motor HP GPM Cert. 206 @	XF51 1½ 11.5 10.8	XF51 1½ 10.4 9.7	XF54 3 12.0 9.9	XF57 3 13.5 12.3	XF58 5 10.5 9.3	100	27	40	¾		1½	29½	45	78
81 to 90	6.2	3105	Unit No. Pkg. Motor HP GPM Cert. 206 @	XF52 1½ 15.7 13.9	XF53 2 13.8 12.0	XF56 3 15.0 14.2	XF57 3 13.5 12.3	XF60 7½ 17.0 16.5	100	27	40	¾		1½	29½	45	78
91 to 100	6.9	3450	Unit No. Pkg. Motor HP GPM Cert. 206 @	XF52 1½ 15.7 13.9	XF53 2 13.8 12.0	XF56 3 15.0 14.2	XF57 3 13.5 12.3	XF60 7½ 17.0 16.5	100	27	40	¾		1½	29½	45	78
101 to 125	8.7	4313	Unit No. Pkg. Motor HP GPM Cert. 206 @	XF55 2 20.2 18.8	XF56 3 18.0 16.9	XF57 3 16.5 15.3	XF59 5 21.7 21.4	XF60 7½ 17.0 16.5	100	27	40	¾		1½	29½	45	78
126 to 150	10.4	5175	Unit No. Pkg. Motor HP GPM Cert. 206 @	XG51 3 29.4 28.8	XG52 3 25.2 23.6	XG53 5 23.4 22.8	XG53 5 21.7 21.4	XG56 7½ 21.0 21.0	180	30	60		¾	2	30¼	66½	78
151 to 200	13.9	6900	Unit No. Pkg. Motor HP GPM Cert. 206 @	XG51 3 29.4 28.8	XG54 5 36.5 32.6	XG54 5 31.7 27.0	XG55 7½ 35.0 32.0	XG52 10 32.2 30.7	180	30	60		¾	2	30¼	66½	78
201 to 250	17.3	8625	Unit No. Pkg. Motor HP GPM Cert. 206 @	XH51 5 48.5 45.2	XH51 5 36.5 32.6	XH51 5 31.7 27.0	XH52 7½ 35.0 32.0	XH53 10 32.2 30.7	250	36	60		¾	2	36½	66½	78
251 to 300	20.7	10350	Unit No. Pkg. Motor HP GPM Cert. 206 @	XJ51 5 48.5 45.2	XJ52 5 45.6 41.6	XJ54 7½ 44.0 36.0	XJ54 7½ 42.0 33.0	XJ56 15 40.6 38.7	350	42	60		¾	2½	42¼	66½	78
301 to 350	24.2	12075	Unit No. Pkg. Motor HP GPM Cert. 206 @	XJ51 5 48.5 45.2	XJ53 7½ 50.5 46.5	XJ55 10 59.0 50.5	XJ57 15 68.0 62.0	XJ58 20 49.0 43.5	350	42	60		¾	2½	42¼	66½	78
351 to 400	27.7	13800	Unit No. Pkg. Motor HP GPM Cert. 206 @	XK51 7½ 57.0 53.5	XK51 7½ 50.5 46.5	XK53 10 59.0 50.5	XK54 15 68.0 62.0	XK55 20 62.5 60.0	500	42	84		1	2½	42¼	80½	78
401 to 500	34.6	17250	Unit No. Pkg. Motor HP GPM Cert. 206 @	XK52 7½ 74.0 59.5	XK53 10 68.0 56.5	XK54 15 76.0 69.0	XK54 15 68.0 62.0	XK55 20 62.5 60.0	500	42	84		1	2½	42¼	90½	78

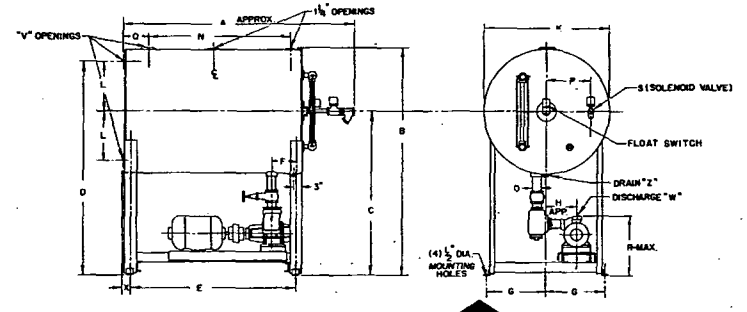
FOR SMALL BOILERS— in a range of 1 to 40 horsepower, Roth End Mounted Single Stage Pumps are recommended. These pumps are described on page [illegible] of this catalog.

For a more complete line, provides selections for factory assembled units for small boilers.

Boiler HP		Pump Discharge Pressure — PSI 30	80	100	125	150
1-12	UNIT NO. MOTOR HP	XB75 1/4	XB67 ½	XB67 ½	XB70 ¾	XB74 1½
13-20	UNIT NO. MOTOR HP	XB75 1/4	XB67 ½	XB70 ¾	XB73 1½	XB74 1½
21-30	UNIT NO. MOTOR HP	XC76 1/3	XC78 ¾	XC71 1	XC68 1½	XC73 2
31-40	UNIT NO. MOTOR HP	XC76 1/3	XC71 1	XC67 1	XC69 1½	XC73 2

DIMENSIONAL DATA

PACKAGED BOILER FEED UNIT (SIMPLEX)



DIMENSIONS IN INCHES (SIMPLEX)

Receiver Capacity	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	Q	R	S	T	V	W	Z
60 Gal.	74⅜	70	47⅜	53⅜	62	36	11⅝	6⅝	27⅛	24⅛	23⅛	1½	6	10"	6	12¼	5	20½	4	8	1¼	2
100 Gal.	98⅜	94	71⅜	77⅜	86	60	11⅝	6⅝	27⅛	24⅛	23⅛	1½	6	10"	6	12¼	5	20½	4	8	1¼	2
120 Gal.	98⅜	94	71⅜	77⅜	86	60	11⅝	6⅝	27⅛	24⅛	23⅛	1½	6	10"	6	12¼	5	20½	4	8	1¼	2
200 Gal.	74⅜	70	47⅜	53⅜	62	36	11⅝	6⅝	27⅛	24⅛	23⅛	1½	6	10"	6	12¼	5	20½	4	8	1¼	2
250 Gal.	98⅜	94	71⅜	77⅜	86	60	11⅝	6⅝	27⅛	24⅛	23⅛	1½	6	10"	6	12¼	5	20½	4	8	1¼	2
300 Gal.	98⅜	94	71⅜	77⅜	86	60	11⅝	6⅝	27⅛	24⅛	23⅛	1½	6	10"	6	12¼	5	20½	4	8	1¼	2

For 150 lb. ASA flanges substitute following dimensions for units listed below:

Unit No.	A	B	C
XE51-XE55, XF51-XF60, XG51-XG53, XG56, XH51-XH53, XJ51-XJ56, XK51-XK53, XJ57-XJ58, XK54-XK55	74⅜	70	47⅜
XG54-XG57, XH56-XH58, XK56-XK57	98⅜	94	71⅜

DIMENSIONS IN INCHES (DUPLEX)

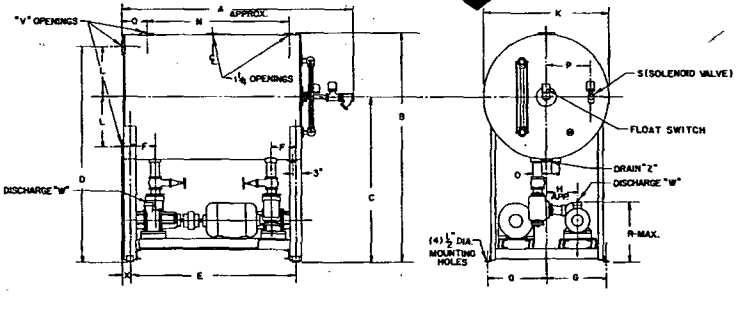
Receiver Capacity	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	Q	R	S	T	V	W	Z
120 Gal.	74⅜	70	47⅜	53⅜	62	36	11⅝	6⅝	27⅛	24⅛	23⅛	1½	6	10"	6	12¼	5	20½	4	8	1¼	2
200 Gal.	74⅜	70	47⅜	53⅜	62	36	11⅝	6⅝	27⅛	24⅛	23⅛	1½	6	10"	6	12¼	5	20½	4	8	1¼	2
250 Gal.	98⅜	94	71⅜	77⅜	86	60	11⅝	6⅝	27⅛	24⅛	23⅛	1½	6	10"	6	12¼	5	20½	4	8	1¼	2
300 Gal.	98⅜	94	71⅜	77⅜	86	60	11⅝	6⅝	27⅛	24⅛	23⅛	1½	6	10"	6	12¼	5	20½	4	8	1¼	2

For 150 lb. ASA flanges substitute following dimensions for units listed below:

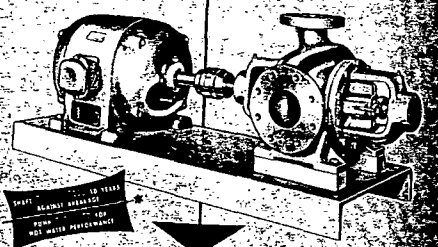
Unit No.	A	B	C
XE51D-XE55D, XF51D-XF60D, XG51D-XG53D, XG56D, XH51D-XH53D, XJ51D-XJ56D, XK51D-XK53D, XJ57D-XJ58D, XK54D-XK55D	74⅜	70	47⅜
XG54D-XG57D, XH56D-XH58D, XK56D-XK57D	98⅜	94	71⅜

DIMENSIONAL DATA

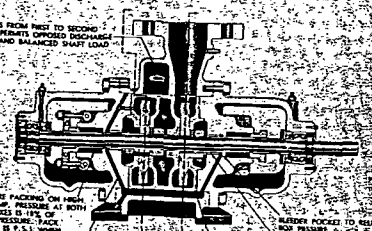
PACKAGED BOILER FEED UNIT (DUPLEX)



ROTH TWO-STAGE POWER PLANT PUMPS



SERIES 3000

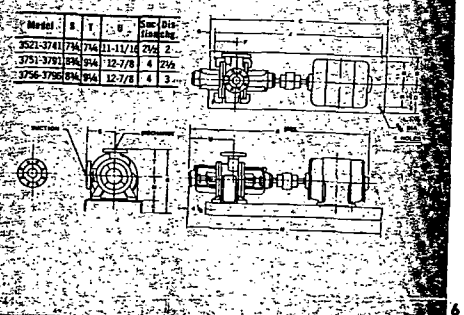


INSTALLATION DIMENSIONS — SERIES 3000

	Motor Fr. NEMA Std.	A	A*	B	C	D	F	G	H	J	E	M	N
1750 RPM	213, 215	43-7/16	45	16-9/16	40	6¾	6-11/16	⅞	5	3⅜	3-1/16	6¼	44⅛
	254U, 256U	48-3/8	48-5/8	15-15/16	46	7½	6-5/16	1¼	6	4½	3-7/16	6¼	50⅛
	284U	50	51-1/8	17-11/16	46	7½	6-5/16	1¼	6	4½	3-7/16	7	50⅛
	286U	51-1/2	52-5/8	18-3/16	52	9	6-5/16	1¼	7½	4¾	3-15/16	7	56⅛
	324U, 326U	54-1/4	55-3/8	19-3/16	52	9	6-5/16	1¼	7½	4¾	3-15/16	8	56⅛
3500 RPM	Motor Fr. NEMA Std.	A	A*	B	C	D	F	G	H	J	E	M	N
	213, 215	44-5/8	46-3/16	18-9/16	40	6¾	6	⅞	5	3⅜	3-1/16	6¼	44⅛
	254U, 256U	49-9/16	50-13/16	18-15/16	46	7½	5-5/8	1¼	6	4½	3-7/16	6¼	50⅛
	284U	51-3/16	52-5/16	19-11/16	46	7½	5-5/8	1¼	6	4½	3-7/16	7	50⅛
	286U	52-11/16	53-13/16	20-3/16	52	9	5-5/8	1¼	7½	4¾	3-15/16	7	56⅛
	324U, 326U	55-7/16	56-9/16	21-3/16	52	9	5-5/8	1¼	7½	4¾	3-15/16	8	56⅛

A: Covers standard drip-proof motors.
A* Covers totally enclosed motors.

Model	S	T	U	Suc. Flanging	Dis. Flanging
3521-3741	7¼	7¼	11-11/16	2½	2
3751-3791	8¾	9¼	12-7/8	4	2½
3756-3796	8¾	9¼	12-7/8	4	3



SERIES 3000 and 2000

ROTH Two-Stage Power Pumps serve a variety of applications found in steam power service such as feeding water to steam boilers in a range of 100 to 500 PSI as well as condensate return, autoclave, desuperheater, and vacuum condenser service.

Series 3000 and 2000 have excellent hot water handling characteristics due to the large passages between stages, the special suction entrance modifications, and the inherent vapor handling characteristics of the exclusive ROTH turbine impeller design.

Both series are ruggedly built with extra heavy duty, flexible coupled shafts guaranteed for 10 years against breakage (see ROTH Guarantee, page 12). End-play has been completely eliminated and pressure lubricated ball bearings maintain precision performance and assure long life under hardest usage.

Steam engineers understand the proper use and maintenance of packing far better than the harder-to-replace mechanical seal. To cater to this preference, ROTH Power Plant Pumps are designed to handle hot water at high pressures with ordinary packing. Pressure on the packing has been reduced by approximately 85% through the installation of a throttle bushing between the impeller and stuffing box in each side. Packing maintenance is minimized and adjustment made simple and easy.

SERIES 3000 two-stage turbine power plant pumps consist of ten models divided into two general frame sizes.

Balanced design is one of the several factors resulting in the better performance and long life of ROTH series 3000. In these pumps, the two stages are arranged so that loads

(continued on next page)

SELECTION TABLE — SERIES 3000 — 1750 RPM

Model No.	Sect.	Disch.	RPSH		Total Head in Feet 100	150	200	250	300	350	400	450	500	600	700
3521	2½	2	30'	GPM	18	17	15	14	13	12	11	10	9	7	5
			7'	GPM	17	16	14	13	12	11	10	9	8	6	4
			3'	GPM	15	14	13	13	12	11	10	9	8	6	4
				BHP	3.9	4.1	4.4	4.6	4.9	5.1	5.4	5.7	6.0	6.5	7.1
3711	2½	2	30'	GPM	30	29	28	26	25	24	22	21	20	17	14
			7'	GPM	30	29	28	26	25	24	22	21	20	17	14
			3'	GPM	25	24	24	23	22	21	21	20	19	17	14
				BHP	4.2	4.6	4.9	5.2	5.6	6.1	6.6	7.3	7.8	8.8	9.8
3721	2½	2	30'	GPM	46	44	42	40	39	37	35	33	31	26	22
			7'	GPM	40	39	38	37	36	34	33	31	29		
			3'	GPM	33	33	32	31	30	29	28	27	26		
				BHP	4.0	4.7	5.4	6.1	6.8	7.5	8.3	9.1	9.8	11.3	12.8
3741	2½	2	30'	GPM	66	61	57	54	50	47	44	41	39	34	30
			7'	GPM	53	52	51	50	47	45	42	39	37	32	28
			3'	GPM	42	41	40	38	36	34	32	31	29	27	26
				BHP	5.0	6.2	7.3	8.5	9.5	10.5	11.7	12.8	13.8	16.0	18.2
3751	4	2½	30'	GPM	102	96	90	84	78	72	66	60	51	41	
			7'	GPM	91	87	82	77	69	62	54	45	36		
			3'	GPM	66	65	64	63	60	56	50	42	32		
				BHP	5.0	7.8	10.5	13.0	15.5	17.5	19.0	20.5	21.5		
3756	4	3	30'	GPM	180	154	124	86							
			9'	GPM	138	137	121	88							
			6'	GPM	108	105	99	82							
				BHP	10.0	13.0	16.0	18.0							
3771	4	2½	30'	GPM	127	113	104	95	86	78	71	64	57		
			7'	GPM	106	103	99	92	83	75	68	61			
			3'	GPM	69	68	67	66	65	63	60	55			
				BHP	12.0	13.0	14.3	16.0	17.5	19.2	21.0	23.0	25.7		
3776	4	3	30'	GPM	202	166	131								
			9'	GPM	142	128	100								
			6'	GPM	90	88	76								
				BHP	11.0	14.0	17.5								
3791	4	2½	30'	GPM	160	149	138	128	118	108	99	90	81		
			8'	GPM	128	128	127	123	115	104	95	86			
			4'	GPM	102	99	95	92	88	85	81	77			
				BHP	12.0	14.0	16.0	17.6	19.6	21.5	24.0	26.0	28.5		
3796	4	3	30'	GPM	266	230	197	159	122						
			8'	GPM	242	222	195	159	122						
			6'	GPM	179	175	168	154	122						
				BHP	14.0	18.0	22.0	26.0	32.0						

MECHANICAL DATA — SERIES 3000

Model No.		Hydrostatic Test Press. — PSI C.I.	Brz.	Max. Water, Oil or Gas — PSI C.I.	Brz.
3521-3741	Case 2½" Suction 2" Discharge	500	600	400 400	500 500
3751-3791	Case 4" Suction 2½" Discharge	500	600	175 400	225 500
3756-3796	Case 4" Suction 3" Discharge	500	600	175 400	225 500

*SEE PAGE TWELVE

are balanced between the two stages. This is achieved by opposing the point of discharge of the two stages and connecting discharge of first stage with suction of second stage by an external channel.

Pumps are standard fitted, bronze fitted, and all bronze and are supplied with ASA flanged connections.

Models 3521, 3711, 3721, and 3741 have the by-pass between stages located under the pump. Two impellers are arranged in series.

Models 3751, 3756, 3771, 3776, 3791, and 3796 have the by-pass between stages located at top of pump. Models with numbers ending in digit (1) contain impellers arranged in series. Models with numbers ending in digit (6) contain impellers arranged in parallel.

SERIES 2000 two-stage turbine power plant pumps consist of fifteen models divided into two general frame sizes.

Standard fitted, bronze fitted, and all bronze pumps are supplied with ASA screwed connections.

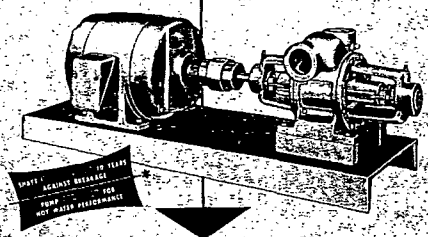
SELECTION TABLE — SERIES 2000 — 1750 RPM

Model No.	Suct.	Disch.	KPSH		Total Head in Feet									
					100	150	200	250	300	350	400	450	500	600
2511	1½	1¼	30' 7' 3'	GPM	13.2	11.8	10.6	9.3	8.4	7.4	6.4	5.4	4.5	2.7
				GPM	11.2	10.0	8.7	7.6	6.5	5.5	4.6	3.6		
				GPM	11.2	10.0	8.7	7.6	6.5	5.5	4.6	3.6		
				BHP	1.1	1.3	1.5	1.7	2.0	2.2	2.5	2.7	2.9	3.5
2521	1½	1¼	30' 7' 3'	GPM	12.7	12.4	11.1	10.0	9.0	8.0	7.1	6.3	5.6	4.2
				GPM	12.6	11.6	10.4	9.2	8.2	7.2	6.3	5.4		
				GPM	12.6	11.6	10.4	9.2	8.2	7.2	6.3	5.4		
				BHP	1.1	1.3	1.5	1.8	2.0	2.3	2.5	2.8	3.0	3.5
2531	1½	1¼	30' 7' 3'	GPM	15.1	13.5	12.1	10.8	9.6	8.4	7.2	6.0	4.6	1.8
				GPM	13.7	12.4	11.1	9.8	8.5	7.3	6.1	4.9		
				GPM	13.7	12.4	11.1	9.8	8.5	7.3	6.1	4.9		
				BHP	1.4	1.6	1.8	2.1	2.4	2.7	3.0	3.4	3.9	4.9
2531A	1½	1¼	30' 7' 3'	GPM	19.1	16.9	15.0	13.2	11.6	10.0	8.5	7.0	5.5	2.8
				GPM	15.3	15	12.9	12.1	10.4	8.8	7.3	5.8		
				GPM	14.1	13.9	12.7	11.1	9.5	7.9	6.4	5.0		
				BHP	1.5	1.8	2.1	2.5	2.8	3.2	3.6	4.1	4.5	5.5
2541	1½	1¼	30' 7' 3'	GPM	15.8	14.8	13.4	12.3	11.2	10.2	9.2	8.2	7.2	5.4
				GPM	14.6	13.5	12.6	11.6	10.6	9.6	8.5	7.4		
				GPM	14.2	13.4	12.5	11.6	10.6	9.6	8.5	7.4		
				BHP	1.9	2.2	2.5	2.8	3.1	3.5	3.8	4.2	4.5	5.2
2551	1½	1¼	30' 7' 3'	GPM	25	22	19.5	17	14.8	12	9.6	7.2	5.0	
				GPM	21	20	18.2	16	13.8	11.4	9.0	6.5		
				GPM	21	20	18.2	16	13.8	11.4	9.0	6.5		
				BHP	1.7	2.0	2.4	3.0	3.4	3.9	4.5	5.0	5.0	
2561	1½	1¼	30' 7' 3'	GPM	25.8	23.7	21.6	19.7	17.8	16	14.5	12.8	11.3	
				GPM	23.6	21.9	20	18.1	16.4	14.7	13.2	11.7		
				GPM	17.7	17.4	17.1	16.6	15.7	14	12.4	10.9		
				BHP	1.8	2.2	2.6	3.0	3.5	4.0	4.5	5.0	5.5	
2571	1½	1¼	30' 7' 3'	GPM	34	32	29	25	21	17.2	13.5	9.7	6.0	
				GPM	33	31	28	24	20	16.4	12.4	8.3		
				GPM	22	22	22	21	19.6	15.7	11.8	7.7		
				BHP	2.6	3.0	3.3	3.8	4.4	5.1	5.8	6.5	7.4	
2581	1½	1¼	30' 7' 3'	GPM	29.9	28.0	26.3	24.6	23.1	21.6	20.0	18.6	17.2	
				GPM	24.0	23.9	23.8	23.5	22.5	21.2	19.8	18.6		
				GPM	17.0	17.0	16.9	16.7	16.4	16.0	15.6	15.1		
				BHP	2.1	2.7	3.2	3.8	4.3	4.9	5.5	6.1	6.6	
2591	1½	1¼	30' 7' 3'	GPM	37.0	34.6	31.5	27.2	21.8	17.0	12.7	8.5	4.7	
				GPM	32	31	28.5	24.5	20	14.8	9.5	4.2		
				GPM	21.0	21.0	20.8	20.2	18.7	16.8	9.5	4.2		
				BHP	3.2	3.5	4.0	4.5	5.4	6.2	7.0	7.8	8.5	
2631	2½	2	30' 7' 3'	GPM	46	43	39	35	31	27	23	19	15	
				GPM	41	38	35	32	28	25	21	17		
				GPM	34	33	32	29	27	23	20	16		
				BHP	3.2	3.8	4.5	5.3	6.1	7.0	8.2	9.4	10.6	
2651	2½	2	30' 7' 3'	GPM	56	52	48	44	40	35	31	27		
				GPM	51	48	44	40	36	32	27			
				GPM	36	36	35	34	32	30	26			
				BHP	3.6	4.4	5.3	6.2	7.2	8.3	9.4			
2671	2½	2	30' 7' 3'	GPM	69	62	55	48	41	34				
				GPM	61	58	51	44	36	28				
				GPM	54	50	45	41	35	27				
				BHP	4.8	5.9	7.0	8.4	9.6	11.0				
2681	2½	2	30' 7' 3'	GPM	57.8	54.9	51.8	48.5	45.0	41.5				
				GPM	47	45	43	40	36	33				
				GPM	30.0	30.0	30.0	29.6	29.0	28.2				
				BHP	5.0	5.2	5.8	6.4	7.2	8.3				
2691	2½	2	30' 7' 3'	GPM	86	79	72	65	58					
				GPM	61	61	59	55	49					
				GPM	43	40	36	32	27					
				BHP	5.3	6.8	8.3	10.0	11.6					

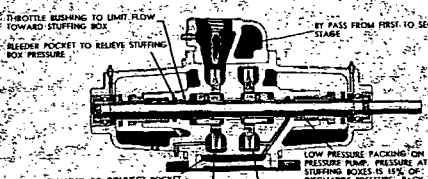
MECHANICAL DATA — SERIES 2000

Model No.		Hydrostatic Test Press. — PSI		Max. Water, Oil or Gas — PSI	
		C.I.	Brz.	C.I.	Brz.
2511-2591	Case 1½" Suction 1¼" Discharge	500	600	400 400	500 500
2631-2691	Case 2½" Suction 2" Discharge	500	600	175 175	400 400

ROTH TWO-STAGE POWER PLANT PUMPS



SERIES 2000

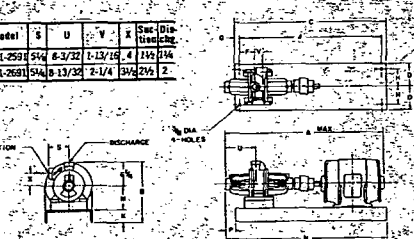


INSTALLATION DIMENSIONS — SERIES 2000

	Motor Fr. NEMA Std*	A_1	A_2	B	C	D	E	F	G	H	J	K	M	N	P
2511 thru 2591	182, 184	34-5/16	36-1/8	12-3/4	30	6¼	3-21/32	⅞	5	2¼	4	2	4½	33-9/16	3-9/16
	213, 215	38-5/16	39-7/8	14-9/16	40	6¼	4-21/32	⅞	5	3⅜	4	3-1/16	5¼	42-9/16	2-9/16
	254U, 256U	41-1/2	42-3/4	15-9/16	40	6¾	4-21/32	⅞	5	3⅜	4	3-1/16	6¼	42-9/16	2-9/16
2631 thru 2691	Motor Fr. NEMA Std*	A_1	A_2	B	C	D	E	F	G	H	J	K	M	N	P
	182, 184	36-5/8	38-7/16	13-13/16	40	6¼	5	⅞	5	3⅜	4	3-1/16	4½	42-9/16	2-9/16
	213, 215	40-5/8	42-3/16	14-9/16	40	6¼	5	⅞	5	3⅜	4	3-1/16	5¼	42-9/16	2-9/16
	254U, 256U	45-9/16	46-13/16	15-9/16	40	6¼	2-23/32	⅞	5	3⅜	4	3-1/16	6¼	44-13/16	4-13/16
	284U, 286U	48-11/16	49-13/16	16-11/16	48	7½	3-7/8	1¼	6	4⅜	4½	3-7/16	7	49-5/16	3-5/16

*A_1 Covers standard drip-proof motors
A_2 Covers totally enclosed motors

Model	S	U	V	X	Suction	Discharge
2511-2591	5/16	6-3/32	1-13/16	4	1½	1¼
2631-2691	5/16	8-13/32	2-1/4	3½	2½	2

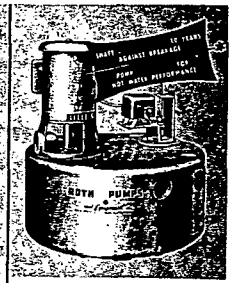


*SEE PAGE TWELVE

ROTH

VERTICAL CONDENSATE UNITS

(with cast iron or welded steel receivers)



SPECIFICATION FORM

Furnish and install as shown on the drawings one vertical condensate unit, Roth Model ..... or equal for ..... sq. ft. EDR at ..... PSI.

Pump shall be 1750 RPM vertical submerged type with corrosion resistant liners and stainless steel shaft.

Motor shall be General Electric or equal 1750 RPM, all angle, drip proof construction, mounted above the high water level for ..... volt (specify actual operating voltage) ..... cycle ..... phase current. Motor shall be non-overloading at any working pressure below design pressure.

Float switch shall be 2-pole Square D or equal, direct actuated by all-copper float and float rod with visibly adjustable float rod collars and suitable for across-the-line starts on single phase current up to 1 HP load.

A magnetic starter shall be provided suitable for the motor supplied. (Specify for 3 phase current only.)

The receiver shall be of high quality close grain cast iron with 2 inch inlet 7 inches above floor level (specify 9-7/16 inches for 23 gal. rec.) and provided with vent, overflow, and drain connections.

DESCRIPTION

ROTH Vertical Condensate Units are rugged, dependable, long-lived equipment engineered to include the most desirable features needed in pumps for local condensate collection or small steam heating systems.

These units are especially useful where condensate must be drained from low return piping. (Floor to inlet distance on 10 and 15 gallon receivers is 7 inches.) A minimum of floor space is required due to location of motor and switch on top of receiver which also permits shallow pit installation without hazard of flooding the electrical parts.

All models listed in the selection data below operate at 1750 RPM providing a full range of pressures up to 60 PSI for radiation up to 20,000 sq. ft EDR. This low operating speed plus special bronze bearing makes for unusually long, trouble-free service.

Up to 20 PSI working pressure, the pump operates on centrifugal phase. It will not vapor bind at 210°F. All working parts are of bronze or stainless steel.

A fully guaranteed General Electric or equal drip-proof, all-angle motor with thrust load carried on ball bearings is standard equipment. The unit is controlled by a heavy Square D, 2-pole float switch adequate for across-the-line starts on single phase current up to 1 HP load.

All cast iron receivers are equipped with 2 inch inlet plus drain, overflow, and vent connections.

SIMPLEX UNITS are float switch operated and provided with cast iron receivers of 10, 15, or 23 gallon capacity.

DUPLEX UNITS are standard with two float switches, one connected to each pump separately. Floats are set at separate levels to start second pump at peak returns. They are furnished with cast iron receivers of 15 or 23 gallon capacity. (Mechanical alternator can be substituted for the two float switches at slight additional cost.)

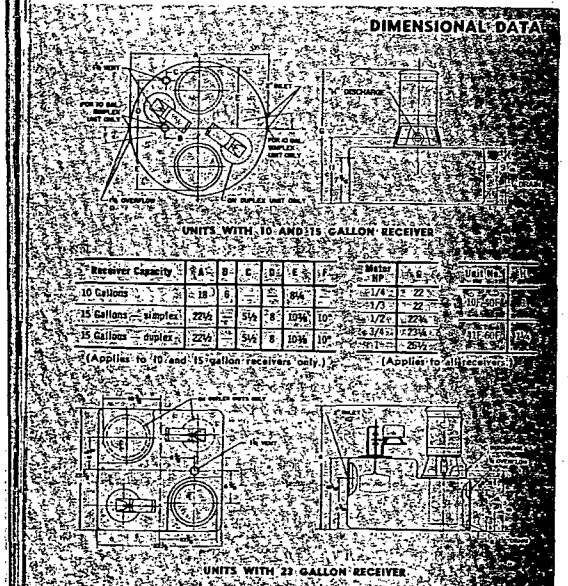
Units also available with heavy gauge welded steel receivers made from copper-bearing steel having .20% copper content and lined with phenolic resin.

SELECTION DATA — VERTICAL UNITS — 1750 RPM

Sq.Ft. E.D.R.	GPM		Pump Discharge Pressure — PSI							Receiver Data	
			10	15	20	33	40	50	60	Simplex	Duplex
2,000	3.0	Unit No.	10F	10F	10F	10F	11F	29F	34F	Cap.	15
		RPM	1750	1750	1750	1750	1750	1750	1750	Dia.	22½"
		HP	1/4	1/4	1/4	1/4	1/3	1/3	1/2	In. Ht.	7"
3,000	4.5	Unit No.	10F	10F	10F	12F	21F	40F	40F	Cap.	15
		RPM	1750	1750	1750	1750	1750	1750	1750	Dia.	22½"
		HP	1/4	1/4	1/4	1/3	1/3	1/2	1/2	In. Ht.	7"
4,000	6.0	Unit No.	10F	10F	10F	12F	40F	40F	40F	Cap.	15
		RPM	1750	1750	1750	1750	1750	1750	1750	Dia.	22½"
		HP	1/4	1/4	1/4	1/3	1/2	1/2	1/2	In. Ht.	7"
6,000	9.0	Unit No.	13F	17F	17F	19F	43F	43F	45F	Cap.	15
		RPM	1750	1750	1750	1750	1750	1750	1750	Dia.	22½"
		HP	1/4	1/3	1/3	1/2	3/4	3/4	1	In. Ht.	7"
8,000	12.0	Unit No.	15F	18F	18F	41F	44F	50F		Cap.	15
		RPM	1750	1750	1750	1750	1750	1750		Dia.	22½"
		HP	1/4	1/3	1/3	1/2	3/4	1		In. Ht.	7"
10,000	15.0	Unit No.	23F	25F	25F	48F	50F			Cap.	15
		RPM	1750	1750	1750	1750	1750			Dia.	22½"
		HP	1/3	1/2	1/2	3/4	1			In. Ht.	7"
15,000	22.0	Unit No.	52F	52F	54F	60F				Cap.	23
		RPM	1750	1750	1750	1750				L.W.	22½"x22½"
		HP	1/2	1/2	3/4	1				In. Ht.	9-7/16"
20,000	30.0	Unit No.	56F	56F	58F					Cap.	23
		RPM	1750	1750	1750					L.W.	22½"x22½"
		HP	1/2	3/4	3/4					In. Ht.	9-7/16"

1. All pump capacities shown are delivered at temperatures up to 200°F.
2. Will not vapor bind at 210°F.
3. Consult factory for selections involving enclosed motors or switches.
4. Select the unit according to design pressure and maximum expected square feet of direct radiation on the location.
5. Specify simplex or duplex.
6. Specify current and voltage. A specific operating voltage should be specified rather than dual voltage to assist in the proper selection and installation of the equipment.

DIMENSIONAL DATA

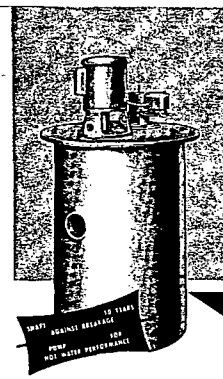


Receiver Capacity	A	B	C	D	E	F	Motor HP	G	Unit Wt.	Rec. Wt.
10 Gallons	18	6	-	-	8¼	-	1/4	22	30	24
15 Gallons — simplex	22½	-	5½	6	10⅜	10"	1/3	22	30	24
15 Gallons — duplex	22½	-	5½	8	10⅜	10"	1/2	22¾	32	26
							3/4	23¼	41	60
							1	25½	41	60

(Applies to 10 and 15 gallon receivers only.) (Applies to all receivers)

ROY E. ROTH COMPANY TURBINE PUMP DIVISION ROCK ISLAND, ILLINOIS

*SEE PAGE TWELVE



ROTH

UNDERGROUND CONDENSATE UNITS

(with cast iron or welded steel receivers)

DESCRIPTION

ROTH Underground Condensate Units are indicated where condensate must be drained from underground piping. The floor-to-inlet distance is 12 inches to make possible a pipe slope for draining purposes. Low silhouette design permits flush floor installation, locating under storage tanks, etc.

Motor, motor mounts, and pump are essentially the same as used on the vertical condensate unit shown on preceeding page.

All underground condensate units operate at 1750 RPM providing a full range of pressures up to 60 PSI for radiation up to 20,000 sq. ft. EDR. This unusually low operating speed plus special bronze bearing makes for unusually long, trouble-free service.

The 1750 RPM pump operates on centrifugal phase up to 20 PSI working pressure. It will not vapor bind at 210°F. All working parts are of bronze or stainless steel. The impeller is precision-located between the adjacent faces in a fixed position to prevent it "riding" the lower face.

Motor is a fully guaranteed, General Electric or equal drip-proof, all-angle model with thrust load carried on ball bearings and controlled by a heavy Square D, 2-pole float switch adequate for across-the-line starts on single phase current up to 1 HP load.

All receivers are of heavy cast iron or welded steel construction flanged at top for bolt-down cover and equipped with 3 inch inlet plus vent connection. Interior is coated with a rust resistant, phenolic lining. Receivers are for atmospheric vented service only.

SIMPLEX UNITS are float switch operated with switches set to start at high level and stop at low level.

DUPLEX UNITS are standard with two float switches, one connected to each pump separately. Floats are set at separate levels to start second pump at peak returns. (One mechanical alternator can be substituted for the two float switches at slight additional cost.)

SELECTION DATA — UNDERGROUND UNITS — 1750 RPM

Unit No.		Sq. Ft. E.D.R.	Pump Dis. Pres.	GPM	Motor HP	Pump Dis. Size	Receiver Data		
CI Rec'r.	Std. Rec'r.							Simpl.	Dupl.
RU11-210	SU11-210	2,000	10	3.0	1/4	1"	Cap. Dia. Depth Inlet In. Dpt.	26 16" 3" 12"	38 30" 3" 12"
RU11-215	SU11-215		15	3.0	1/4	1"			
RU11-220	SU11-220		20	3.0	1/4	1"			
RU11-230	SU11-230		30	3.0	1/4	1"			
RU10-240	SU10-240		40	3.0	1/3	1"			
RU63-250	SU63-250		50	3.0	1/2	1¼"			
RU63-260	SU63-260		60	3.0	1/2	1¼"			
RU11-310	SU11-310	3,000	10	4.5	1/4	1"	Cap. Dia. Depth Inlet In. Dpt.	26 16" 3" 12"	38 30" 3" 12"
RU11-315	SU11-315		15	4.5	1/4	1"			
RU11-320	SU11-320		20	4.5	1/4	1"			
RU13-330	SU13-330		30	4.5	1/3	1"			
RU22-340	SU22-340		40	4.5	1/3	1"			
RU63-350	SU63-350		50	4.5	1/2	1¼"			
RU63-360	SU63-360		60	4.5	1/2	1¼"			
RU11-410	SU11-410	4,000	10	6.0	1/4	1"	Cap. Dia. Depth Inlet In. Dpt.	26 16" 3" 12"	38 30" 3" 12"
RU12-415	SU12-415		15	6.0	1/4	1"			
RU12-420	SU12-420		20	6.0	1/4	1"			
RU13-430	SU13-430		30	6.0	1/3	1"			
RU63-440	SU63-440		40	6.0	1/2	1¼"			
RU63-450	SU63-450		50	6.0	1/2	1¼"			
RU45-460	SU45-460		60	6.0	1	1¼"			
RU12-610	SU12-610	6,000	10	9.0	1/4	1"	Cap. Dia. Depth Inlet In. Dpt.	26 16" 3" 12"	38 30" 3" 12"
RU19-615	SU19-615		15	9.0	1/3	1"			
RU19-620	SU19-620		20	9.0	1/3	1"			
RU21-630	SU21-630		30	9.0	1/2	1"			
RU43-640	SU43-640		40	9.0	3/4	1¼"			
RU43-650	SU43-650		50	9.0	3/4	1¼"			
RU45-660	SU45-660		60	9.0	1	1¼"			
RU18-810	SU18-810	8,000	15	12.0	1/4	1"	Cap. Dia. Depth Inlet In. Dpt.	40 16" 3" 12"	38 30" 3" 12"
RU18-815	SU18-815		15	12.0	1/3	1"			
RU18-820	SU18-820		20	12.0	1/3	1"			
RU70-830	SU70-830		30	12.0	1/2	1¼"			
RU44-840	SU44-840		40	12.0	3/4	1¼"			
RU50-850	SU50-850		50	12.0	1	1¼"			
RU23-1010	SU23-1010	10,000	10	15.0	1/3	1"	Cap. Dia. Depth Inlet In. Dpt.	40 16" 3" 12"	38 30" 3" 12"
RU25-1015	SU25-1015		15	15.0	1/2	1"			
RU25-1020	SU25-1020		20	15.0	1/2	1"			
RU48-1030	SU48-1030		30	15.0	3/4	1¼"			
RU50-1040	SU50-1040		40	15.0	1	1¼"			
RU52-1510	SU52-1510	15,000	10	22.5	1/2	1¼"	Cap. Dia. Depth Inlet In. Dpt.	40 16" 3" 12"	38 30" 3" 12"
RU52-1515	SU52-1515		15	22.5	1/2	1¼"			
RU54-1520	SU54-1520		20	22.5	3/4	1¼"			
RU60-1530	SU60-1530		30	22.5	1	1¼"			
RU56-2010	SU56-2010	20,000	10	30.0	1/2	1¼"	Cap. Dia. Depth Inlet In. Dpt.	40 16" 3" 12"	38 30" 3" 12"
RU58-2015	SU58-2015		15	30.0	3/4	1¼"			
RU58-2020	SU58-2020		20	30.0	3/4	1¼"			

SPECIFICATION FORM

Furnish and install as shown on the drawings one underground condensate unit, Roth Model ..... or equal for ..... sq. ft. EDR at ..... PSI.

Pump shall be 1750 RPM vertical submerged type with corrosion resistant liners and stainless steel shaft mounted inside the receiver.

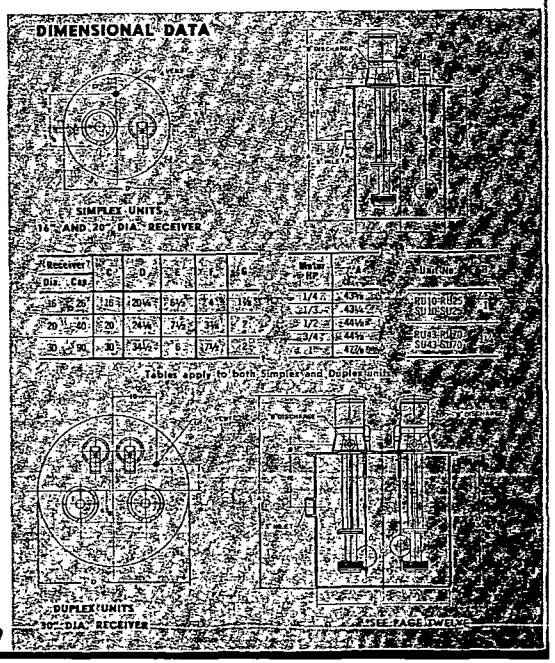
Motor shall be General Electric or equal 1750 RPM, all angle, drip proof construction, mounted above the high water level for ..... volt (specify actual operating voltage) ..... cycle ..... phase current. Motor shall be non-overloading at any working pressure below design pressure.

Float switch shall be 2-pole Square D or equal, direct actuated by all copper float and float rod with visibly adjustable float rod collars suitable for across-the-line starts on single phase current up to 1 HP load.

A magnetic starter shall be provided suitable for the motor supplied. (Specify for 3 phase current only.)

The receiver shall be of high quality close grain cast iron (or 3/16" welded steel with phenolic resin lining) with 3 inch inlet 12 inches below floor level.

DIMENSIONAL DATA

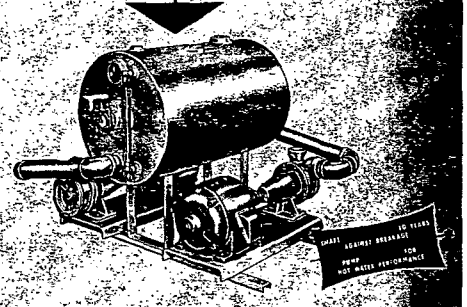


Receiver Dia.	Cap.	A	B	C	D	E	F	Motor HP	G	Unit Wt.	Rec. Wt.
16"	26"	16"	20¼"	6½"	4"	1¼"	1"	1/4	43¾	RU11-210 to RU11-260	38
								1/3	43¾		
30"	90"	30"	34½"	6"	17½"	2½"	1"	1/2	44¼	RU43-640 to RU43-660	38
								3/4	44¾		
								1	44¾		

*SEE PAGE TWELVE

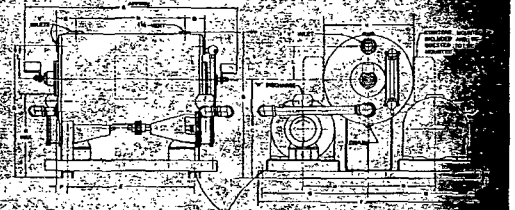
ROTH

HORIZONTAL CONDENSATE RETURN UNITS

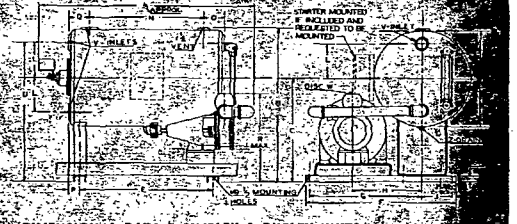


PACKAGE SIMPLEX or DUPLEX UNITS

DUPLEX



SIMPLEX



DIMENSIONAL DATA — SIMPLEX & DUPLEX UNITS

UNIT NO.	W
UB89-UB88, UB110-UB118	1½
UB86, UB119-UB123, UC107-UC122, UE51-UE55, UE98	1¼
UC123-UC125, UE59-UE96, UE99-UE101, UF84-UF90, UF94-UF100, UF105, UG101	2
UE-97, UF92, UF102, UF106, UF107, UG102-UG105	2½*

*250 lb. ASA flange.

DIMENSIONAL DATA — SIMPLEX & DUPLEX UNITS

Tank Size	A		B	C	D	E	F		G	H	J	K	L	N	O	P	Q	R	S	V	Z
	Simp.	Dup.					Simp.	Dup.													
20 gal.	36	39	25⅝	18⅛	23⅜	22	24⅛	38⅛	18⅝	11½	4⅜	16	5¾	19	3⅞	2¼	3⅞	15	1⅛	2	½
35 gal.	44	46½	23½	19	25½	22	24⅞	38⅛	18⅝	11½	4⅜	18	6½	25½	3⅛	6⅛	3⅜	15	1⅛	2½	½
60 gal.	50	52	36¼	24⅛	32⅛	22	32⅛	51⅛	24⅜	18⅛	6¼	22½	7¾	23	5	7½	9	19⅛	1⅛	3	1
100 gal.	60	61	41¼	27	37⅛	23	35¼	53	25⅜	17⅛	9	27	10⅛	28	5	8½	9	20¼	1¾	4	1¼
200 gal.	75	76	45	29⅛	41⅜	30	39⅛	56½	27½	19	10⅛	32	12½	48	5	16	9	20¼	1¾	4	1½

ROY E. ROTH COMPANY TURBINE PUMP DIVISION ROCK ISLAND, ILLINOIS

*SEE PAGE TWELVE

DESCRIPTION

These units are essentially collection stations for condensed steam in heating or process systems. The package unit consists of a receiver tank and a motor driven pump controlled by a float switch mounted in the receiver.

Successful pumping of water at temperatures in the 190°F to 200°F range from low leg receivers requires special pumps designed for those temperatures.

High temperature returns are often caused by defective traps or local dumping of process steam and cause ordinary pumps to vapor bind. ROTH protects you from such occurrences by thoroughly testing and rating for hot water performance the pump models selected and guaranteeing them to perform as rated in the table. If desired, pump selection will be certified for 200°F temperature if request accompanies order. Such certification is based upon special test data obtained from hot water tests run upon the pump of your selection. (A slight charge is made to cover cost of test.)

Since many of these units will be located at points distant from the boiler room, maximum attention has been given to providing pumps that will give dependable service without constant maintenance. For this reason, these units are equipped with ROTH End Mounted Single Stage Pumps operating at

(continued next page)

SPECIFICATION FORM

Furnish and install as called for on the drawings one Roth model ..... simplex (or duplex) horizontal condensate unit or equal for ..... GPM at ..... PSI at 200°F to consist of the following:

One (substitute two for duplex) Roth 1750 RPM single stage end mounted pump with mechanical seal and sealed ball bearings certified for ..... GPM at ..... PSI at 200°F. Pump shall contain stainless steel shaft and bronze impeller and be guaranteed for 10 years against shaft breakage when driven by motor supplied.

One welded non-code vented receiver made from high quality copper-bearing steel sheets and internally lined with phenolic resin lining. (Substitute 75 lb. ASME code receiver with or without stamp on cradle legs if desired.)

One (substitute two for duplex) General Electric, Louis Allis, Howell or equal 1750 RPM drip proof motor selected within recommended service factors for intermittent service and guaranteed by the manufacturer at the point of selection. Current characteristics shall be ..... phase ..... volts.

One (substitute two for duplex) Square D two pole float switch or equal suitable for direct connecting to single phase motors up to 1 HP and for pilot service on larger single phase and all three phase motors. (Substitute mechanical alternator for two float switches if desired on duplex units.)

For units with three phase motors or large single phase motors one (or two) magnetic starter is required.) One (substitute two for duplex) magnetic starter of approved type suitable for the above motor, with holding coil for ..... volts.

The entire unit to be factory assembled and factory guaranteed.

1750 RPM and furnished with sealed factory-lubricated bearings and dependable mechanical seals, to eliminate the need of periodic lubrication and re-packing. (Grease fittings are provided to coat the bearing with a water-resistant perservative once or twice a year.) Packed pumps available if specified.
Because of the steep performance curve characteristic of ROTH turbine pumps, these units will maintain a fairly constant capacity over a considerable range of discharge pressures — a distinct advantage over comparable centrifugal-type pumps.

At operating pressures lower than design pressure the pump will not slug the boiler in heating systems. Also there is no increase in motor load at under-design-pressure conditions.

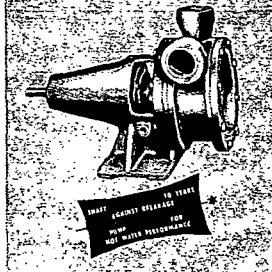
SELECTION DATA — HORIZONTAL CONDENSATE UNITS — 1750 RPM

Sq. Ft. E.D.R.	Boil'r HP	Lbs. Per Hour	GPM		Pump Discharge Pressure — PSI								Receiver Data					Unit Data — Simplex			
					10	15	20	30	40	50	60	75	Cap.	Dia.	L*th	Inlet Size	Inlet Rt.	Pkg.	Wt.	L*th	Rt.
1,000	7½	250	1.5	Unit No. Pkg.	UB110	UB110	UB110	UB110	UB110	UB115	UB88		20	16	25	2	23⅝	114	27⅛	36	25⅛
				Motor HP	⅛	⅛	⅛	⅛	⅛	⅛	½										
2,000	15	500	3	Unit No. Pkg.	UB110	UB110	UB110	UB110	UB110	UB115	UB88		20	16	25	2	23⅝	114	27⅛	36	25⅛
				Motor HP	⅛	⅛	⅛	⅛	⅛	⅛	½										
3,000	22½	750	4.5	Unit No. Pkg.	UB110	UB110	UB110	UB111	UB112	UB88	UB88		20	16	25	2	23⅝	114	27⅛	36	25⅛
				Motor HP	⅛	⅛	⅛	⅛	½	½	½										
4,000	30	1,000	6	Unit No. Pkg.	UB110	UB111	UB111	UB111	UB88	UB88	UB118	UB120	20	16	25	2	23⅝	114	27⅛	36	25⅛
				Motor Hp	⅛	¼	⅛	⅛	½	½	1	1									
6,000	45	1,500	9	Unit No. Pkg.	UB111	UB113	UB114	UB70	UB71	UB71	UB118	UB120	20	16	25	2	23⅝	114	27⅛	36	25⅛
				Motor HP	¼	¼	½	½	¾	¾	1	1									
8,000	60	2,000	12	Unit No. Pkg.	UB113	UB113	UB70	UB69	UB71	UB97	UB121	UB122	20	16	25	2	23⅝	114	27⅛	36	25⅛
				Motor Hp	¼	½	½	½	¾	1	1	1½									
10,000	75	2,500	15	Unit No. Pkg.	UB117	UB117	UB117	UB69	UB97	UB119	UB112	UB123	20	16	25	2	23⅝	114	27⅛	36	25⅛
				Motor HP	½	½	½	½	1	1½	1½	2									
15,000	112½	3,750	22.5	Unit No. Pkg.	UC118	UC108	UC109	UC110	UC110	UC122	UC112	UC123	35	18	32	2	25½	2	28⅛	43½	28½
				Motor HP	½	½	¾	1	1	1½	2	3									
20,000	150	5,000	30	Unit No. Pkg.	UC107	UC120	UC120	UC115	UC115	UC124	UC125	UC125	35	18	32	2	25½	2	28⅛	43½	28½
				Motor HP	½	¾	¾	1½	1½	2	3	3									
25,000	187½	6,250	37.5	Unit No. Pkg.	UE51	UE51	UE55	UE95	UE96	UE96	UE97	UE97	60	22½	36	3	32¼	2½	36⅞	50	36¼
				Motor HP	1	1	1½	1½	2	2	3	3									
30,000	225	7,500	45	Unit No. Pkg.	UE71	UE71	UE71	UE95	UE96	UE96	UE97	UE100	60	22½	36	3	32¼	2½	36⅞	50	36¼
				Motor HP	1½	1½	1½	1½	2	2	3	5									
35,000	262½	8,750	52.5	Unit No. Pkg.	UE69	UE69	UE95	UE95	UE96	UE79	UE85	UE101	60	22½	36	3	32¼	2½	36⅞	50	36¼
				Motor HP	1½	1½	1½	1½	2	3	5	7½									
40,000	300	10,000	60	Unit No. Pkg.	UE69	UE69	UE95	UE79	UE79	UE85	UE85	UE101	60	22½	36	3	32¼	2½	36⅞	50	36¼
				Motor HP	1½	1½	1½	3	3	5	5	7½									
50,000	375	12,500	75	Unit No. Pkg.	UF84	UF84	UF94	UF90	UF85	UF91	UF105	UF107	100	27	40	4	37	3	40	56	40½
				Motor HP	1½	1½	2	3	5	5	7½	10									
65,000	487½	16,250	97.5	Unit No. Pkg.	UF100	UF100	UF100	UF101	UF102	UF102	UF106	UF107	100	27	40	4	37	3	40	56	40½
				Motor HP	2	2	2	5	5	5	7½	10									
100,000	750	25,000	150	Unit No. Pkg.	UG102	UG102	UG102	UG106	UG105	UG105	UG105		200	30	60	4	41½	3	44¼	74	45¼
				Motor HP	5	5	5	7½	10	10	10										

*SEE PAGE TWELVE

ROTH

TURBINE END MOUNTED SINGLE STAGE PUMPS



Check these Features:

- ✓ End mounted construction insures complete separation of liquid and mechanical end.
- ✓ Renewable liners provided in all models.
- ✓ Mechanical seal can be replaced without disturbing piping or motor.
- ✓ Factory lubricated sealed bearings eliminate need for periodic lubrication and protect against danger of overgreasing.
- ✓ Heavy stainless steel shaft and large bearings prevent coupling vibrations from being transmitted to impeller.

A PARTIAL LIST of ROTH BOILER FEED and CONDENSATE RETURN UNIT USERS

This list of ROTH customers indicates the nation-wide acceptance of ROTH Boiler Feed and Condensate Return Units, and Power Plant Pumps. Should you care to contact users in your area, please write for list of names.

ROTH GUARANTEES AGAINST SHAFT BREAKAGE!

The pump shaft is the keystone to good pump design. A rigid shaft properly supported by well protected ball bearings will prolong pump life by many years.

All Roth pumps are equipped with unusually heavy shafts or protected by balanced loads or both. Because of superior pump design, cases of shaft failure are extremely rare.

To emphasize this important quality the Roy E. Roth Company now offers this unprecedented guarantee:

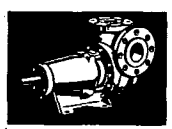
Guarantee

All Roth pumps sold after September 1, 1957 will be guaranteed by the company against shaft breakage for ten years from the date of shipment when flexible coupled to electric motors of sizes recommended by the factory . . . The company's liability under this guarantee will be limited to repair at the factory of any pump with a broken shaft when such breakage occurs within the specified period. Shipping cost to and from the factory to be borne by the owner.

CERTIFIED HOT WATER PERFORMANCE

The Roy E. Roth Company will certify the hot water performance of any model recommended in the foregoing tables upon receipt of request for certification with the purchase order. A slight charge will be made to cover the cost of a special hot water test.

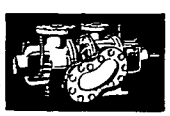
TWO OTHER ROTH TURBINE-PUMPS AVAILABLE FOR SPECIALIZED APPLICATIONS



USAGE: Hot and cold water, acids, caustic solutions, liquified gases, solvents, and hydrocarbons.

RANGE: Head, 20 to 1000 ft.; differential, 0-450 PSI; capacity, 1-140; temperature, —60°F to +600°F; HP, ¾ to 30; NPSH, as low as 3 ft.

ROTH TURBINE CHEMICAL PUMPS—are a special adaptation of the turbine principle specifically designed for continuous, high pressure service under adverse conditions of corrosion or temperature. These pumps have a record of continuous service equalling and often surpassing that of other, more costly types of high pressure chemical pumps and are recommended for clear liquids to 100 cp viscosity.



USAGE: Ammonia, freons, methyl chloride, propane, butane, boiling liquids.

RANGE: Head, 150 to 1600 ft.; differential, 50-700 PSI; capacity, 2 to 100 GPM; temperature, —80°F to +320°F; HP, 5-40 BHP; NPSH, 1, 2, or 3 ft.

ROTH TURBINE LOW NPSH PUMPS—are designed to meet suction-loss problems in liquids having low suction heads with viscosities to 100 cp and temperatures to 300°F. This ability derives from their extremely low NPSH characteristic which is due to the low disturbance at the suction and a gradual pressure build-up throughout the pumping cycle. Available in single or 2-stage design with exclusive booster stage.

ROY E. ROTH COMPANY TURBINE PUMP DIVISION ROCK ISLAND, ILLINOIS

DEFENSE INSTALLATIONS

Major units installed in well over 100 air force bases, army posts and proving grounds, armories, naval stations, and radar stations on the North American continent.

HOSPITAL INSTALLATIONS

Ashland Hospital
Ashland, Kansas
Charity Hospital
New Orleans, Louisiana
Friona Hospital
Friona, Texas
Huntsville Hospital
Huntsville, Alabama
McPherson County Hospital
McPherson, Kansas
Mercy Hospital
Oklahoma City, Oklahoma
Mid-Island Hospital
Bethpage, L. I., New York
New Mexico State Hospital
Las Vegas, New Mexico
P. H. S. Indian Hospital
Tahlequah, Oklahoma
St. Margaret's Hospital
Fredonia, Kansas
St. Mary's Hospital
Galesburg, Illinois
Sedgwick County Hospital
Wichita, Kansas
State Hospital
Milledgeville, Georgia
Wausau Memorial Hospital
Wausau, Wisconsin

INDUSTRIAL INSTALLATIONS

American Bridge Division
Pittsburgh, Penna.
Atlantic Refining Co.
Port Arthur, Texas
Blue Star Produce Co.
Wahoo, Nebraska
Boeing Airplane Co.
Wichita, Kansas
The Borden Co.
Rock Island, Ill.
Fayetteville, Tenn.
Cincinnatus, N. Y.
Leominster, Massachusetts
The Carnation Co.
Oakland, Md.
Cameron, W. Va.
Sullivan, Wisc.
Belleville, Ohio
Glasgow, Ky.
Mt. Vernon, Wash.
Watertown, Tenn.
Rushville, Ohio
Chevrolet
Flint, Mich.
Otterburn, Mich.
Chicago, Burlington & Quincy R.R.
St. Joseph, Mo.
Alliance, Nebraska
Brookfield, Mo.
Aurora, Ill.
Graybull, Wyoming
Chicago, Milwaukee, St. Paul & Pacific R. R. Co.
Tacoma, Washington
Madison, S. D.
Chicago, Illinois
Sioux Falls, S. D.
Mason City, Iowa
St. Paul, Minn.
Green Bay, Wisc.
Curtiss-Wright Corp.
North Hollywood, Calif.
Eagle Food
Milan, Illinois
Federal Crushed Stone Corp.
Cheektowaga, N. Y.
Firestone Plantations
Ford Motor Co.
Northville, Michigan
Warren, Michigan
Dearborn, Michigan
Livonia, Michigan
Somerville, Massachusetts
Goodyear Tire & Rubber Co.
Des Moines, Iowa
Los Angeles, Calif.
Akron, Ohio
Port Newark, New Jersey
Greenwich Telephone Bldg.
Greenwich, Connecticut
Haxton Foods
Wyoming, N. Y.
Hughes Aircraft
Culver City, Calif.
Hawthorne, Calif.
Lake Success Shopping Center
Lake Success, L. I., N. Y.
Lincoln Assembly Plant
Div. of Ford Motor Co.
Novi, Michigan
New Mexico State Penitentiary
Sante Fe, New Mexico
N. Y. International Airport
New York, N. Y.
New York Telephone Co.
Tarrytown, N. Y.
Larchmont, N. Y.
Brooklyn, N. Y.
New York, N. Y.
Northwestern Bell Tele. Co. Bldg.
Mt. Clinton, Iowa
Peoples Shopping Center
Lincoln Park, Michigan
J. C. Penney Co.
256 Units in stores in major cities of 41 states.
St. Lawrence Seaway Project
Staten Island Plaza Shopping Center
Port Richmond, Staten Island, N. Y.
Wausau Iron Works
Wausau, Wisconsin

SCHOOL INSTALLATIONS

A & M College
Las Cruces, New Mexico
Chaplain Kapaun Memorial Jr. High School
Wichita, Kansas
Clinton St. Elementary School
New Hartford, New York
Delavan-Darien High School
Delavan, Wisconsin
East Littleton School
Littleton, Colorado
Geo. Washington Jr. High School
Cedar Rapids, Iowa
Granite City Community High School
Granite City, Illinois
Great Bend High School
Great Bend, Kansas
Indiana University
Bloomington, Indiana
Lowville Central School
Lowville, New York
Mayberry School
Wichita, Kansas
Minidoka County High School
Rupert, Idaho
Rochelle Township High School
Rochelle, Illinois
Salina High School
Salina, Kansas
Stillwater Central School Bldg.
Stillwater, New York
B. W. Truesdell School
Wichita, Kansas
Valley Center High School
Valley Center, Kansas
Whitworth College
Spokane, Washington

Printed in U. S. A.

MANUFACTURERS' CATALOG DATA

PAGES •63—•476

HEATING VENTILATING AIR CONDITIONING GUIDE 1959

On pages ●63–●476 will be found the Catalog Data of 309 manufacturers whose products are described and illustrated.

For the convenience of the user of THE GUIDE 1959 there are ten main divisions:

Air Conditioning	●63–●148
Air Conditioning and Heating Piping	●149–●154
Air System Equipment	●155–●269
Controls and Instruments	●270–●295
Expansion Joints	●296–●303
Fittings and Flanges	●304–●307
Floats	●308
Heating Systems	●309–●427
Insulation	●428–●459
Publications and Exposition	●460–●469

On pages ●7–●24, under each of the index headings—Air Cleaning Equipment, Fans, Humidifiers, Ventilators, etc., will be found a list of manufacturers of any desired products, fully cross-indexed, and the page numbers in the Catalog Data Section where the products are described.

By reference to these indices, the manufacturers' names and the page numbers, any item of equipment or materials, and the producer's address, may be located quickly.

AIR & REFRIGERATION CORP.

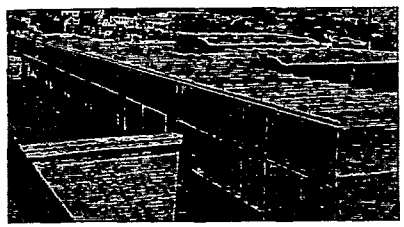
439 Madison Ave., New York-22-N.Y. 271 Milton Ave., S.E., Atlanta, Ga.

CAPILLARY

Sales Representatives in Principal Cities

Foreign Representative—Rocke International Corp.
13 East 40th St., New York-16-N. Y.

A&R TRADE MARK



ARTICULATED UNITS

Units with unlimited capacities are furnished for standard, double duct or multizone distribution systems. Designed for each particular application, they incorporate coordinated equipment to fit within the space available. All units are factory test-assembled to assure fit of parts prior to shipment. We call them ARTICULATED UNITS.

COMPONENTS

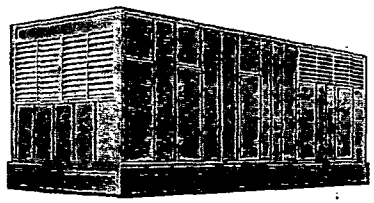
A typical system often combines air mixing, filtration, washing, cooling and heating, humidifying or dehumidifying and air circulating equipment. More complete systems may also include the refrigeration compressor, water chiller, evaporative condenser, pumps and boiler with all being enclosed in a factory insulated casing. Blow-through multizone systems also require outlet air dampers for mixing hot and cold air streams for each zone.

HOUSINGS

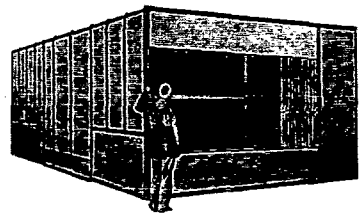
Unit housings are assembled using A & R factory insulated panels. Available in indoor or outdoor weatherproofed construction, the panels consist of an interior galvanized steel sheet to which is applied the insulation, aluminum vapor seal and outer protective sheet of tempered hardboard. For outdoor use an additional weatherproof skin of aluminum is added. To save additional cost, the machine room floor may be used as the bottom of each enclosure or an insulated drip pan may be included.

ADVANTAGES

Articulated units are custom engineered to suit the performance requirements and space limitations of each particular application. All parts are selected to work in co-ordination with the rest of the system.



EQUIPMENT ENCLOSURE—INSULATION IN ONE PREFAB PACKAGE



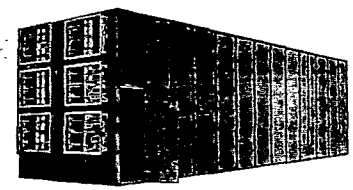
Office time and effort by the purchaser are reduced in adopting articulated unit construction. One purchase order covers materials and services normally requiring a number of separate contracts.

The factory insulated housings do away with the usual mess of field insulation and plastering. This may be an important factor, particularly in older buildings such as department stores, etc., where valuable stocks of merchandise cannot be removed. Prime coated before leaving the factory, the exterior presents a continuity and neatness of appearance which needs no field painting unless scratched or marred in handling and erection.

The panel type construction permits sections of units to be dismounted quickly and easily where it is necessary to remove bulky internal parts for repair or replacement.

CATALOGS

For further information on A & R Articulated Units or Factory Insulated Panels, write direct to main office in New York, or contact factory representative in your area.



Acme INDUSTRIES, INC. 600 N. Mechanic St. Jackson, Mich.

✱ COMPLETE AIR CONDITIONING AND REFRIGERATION SYSTEMS

✱ ADVANCED DESIGN COMPONENTS

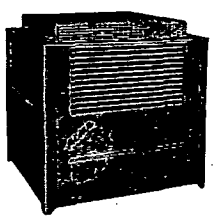
Look to Acme for all types of air conditioning systems. Acme supplies a complete line of air conditioning and process cooling equipment including packaged liquid chillers; and remote room conditioners, air handlers and multi-zone units for use in direct expansion or chilled water systems. Acme water saving equipment includes evaporative condensers, indoor-outdoor cooling towers and outdoor cooling towers.

In addition, Acme offers a complete range of self-contained air conditioners.

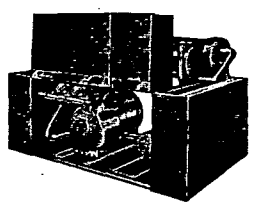
Acme components for built-up systems have long been recognized for excellent quality. All major manufacturers of air conditioning and refrigeration equipment have used Acme components for many years.

Call your Acme sales engineer for details on how he can help you with your air conditioning or process cooling problems.

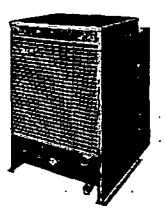
MANUFACTURERS OF QUALITY AIR CONDITIONING AND REFRIGERATION EQUIPMENT SINCE 1919



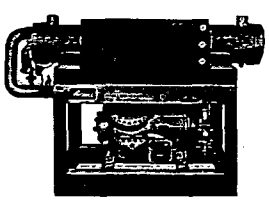
Flow Pac Packaged Chillers— 1¾ through 5 hp



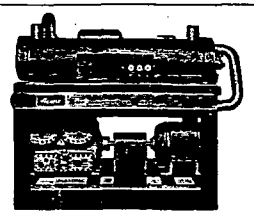
Flow Cold Packaged Water Chillers—3 through 30 tons



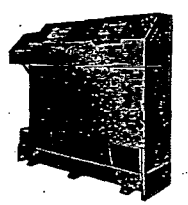
Flow Cold Cooling Towers— 3 through 20 tons



Model HE Flow Therm Packaged Water Chillers— 20 through 60 tons



Model DE Flow Therm Packaged Water Chillers— 20 through 125 tons



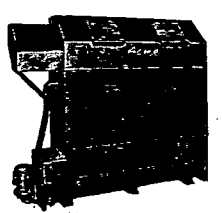
Flow Mixer Cooling Towers— 20 through 175 tons

Acme INDUSTRIES, INC. 600 N. Mechanic St. Jackson, Mich.

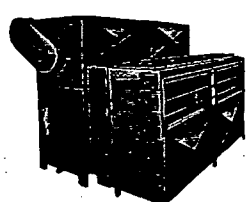
FOR COORDINATED BUILT-UP SYSTEMS . . . USE THESE Acme MATCHED COMPONENTS

- **DIRECT EXPANSION COOLERS** *1 to 300 tons*
- **SHELL AND TUBE CONDENSERS** *1½ to 700 tons*
- **SHELL AND COIL CONDENSERS** *¾ to 10 tons*
- **HEAT EXCHANGERS** *3 to 150 tons*
- **LIQUID RECEIVERS** *3 to 150 tons*
- **OIL SEPARATORS** *5 to 100 horsepower*
- **AIR COOLED CONDENSERS** *3 to 100 tons*
- **DIRECT EXPANSION COILS**
- **WATER AND STEAM COILS**

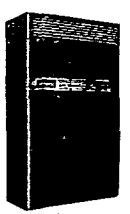
A complete range of sizes and capacities for almost every OEM, built-up system and replacement need.



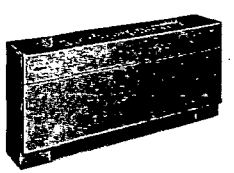
Flow Mixer Evaporative Condensers— Capacities to 200 tons



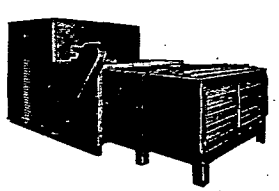
Air handlers—4 types in 40 models from 665 to 19,200 cfm



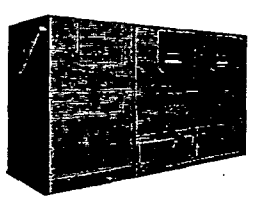
Self contained packaged air conditioners—5 models from 3 to 15 tons, air or water cooled



Remote room conditioners— 4 types in 16 models from 200-600 cfm



Multi-zone air conditioners—5 models from 4,060 to 19,200 cfm



Self contained packaged air conditioners—24 models from 20 to 60 tons, air or water cooled

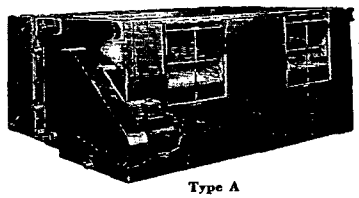
AMERICAN-Standard

AMERICAN BLOWER DIVISION

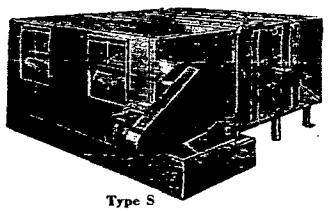
Detroit 32, Michigan

AMERICAN-STANDARD PRODUCTS (CANADA) LTD., Toronto, Ontario
CANADIAN SIROCCO PRODUCTS Branch Offices in Principal Cities

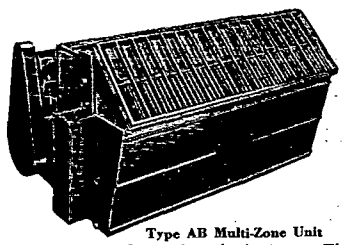
AIR CONDITIONING—HUMIDIFYING—DEHUMIDIFYING—VENTILATING—HEATING—VAPOR-ABSORPTION—DRYING—AIR WASHING AND PURIFICATION—EXHAUSTING EQUIPMENT AND MECHANICAL DRAFT APPARATUS



Type A

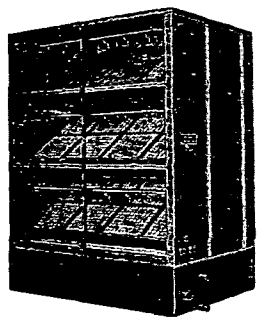


Type S

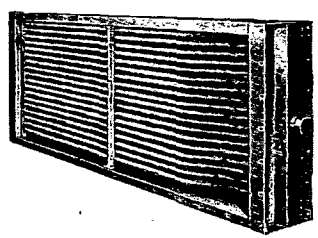


Type AB Multi-Zone Unit

Air Conditioning Units—four basic types. Eleven sizes. Capacities from 800 to 40,000 cfm, and for operation up to 9 in. static pressure. Type A for unitary-type applications: Summer cooling and dehumidifying, using chilled water, direct-expanded refrigerant, or brines; winter heating, using steam or hot water. Type S is the same basic unit as Type A, plus spray system, water recirculating pump and eliminators; combines the washing, air cleaning, humidifying, and evaporative cooling with the cooling and heating of the basic unit conditioner. Type AB Multi-Zone Unit for either horizontal or vertical air flow. Plenum and diffusion device provides even air distribution over coils. Type C combines the features of the capillary air washer with the basic Type A unit. Request *Bulletin 9227.*



Capillary Air Washers—for high efficiency in cleaning, humidification, cooling and dehumidification of air. Air is forced at low resistance through long, irregular passages of small size formed by a large amount of thoroughly wetted glass surface. Write for *Bulletin 4125.*

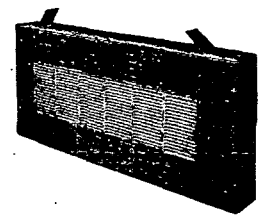


Heating and Cooling Coils—American Blower heating and cooling coils offer a number of improvements in design and construction. Available in a complete range of sizes and types, including:

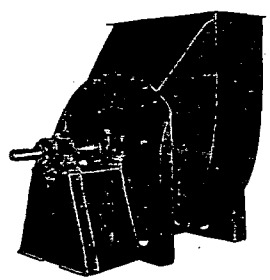
Bulletin B-1218	Type S steam coils
	Type D double tube coils
	Type U return bend coils
	Type B booster coils
Bulletin 1521	Type W water coils
	Type C cleanable water coils
	Type X direct-expansion coils
Bulletin B-1318	Type H heavy-duty coils
Bulletin B-1418	Type D-1 double tube coils



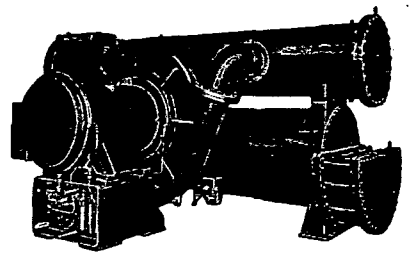
Diaflo® Conditioners—fan-coil type air-conditioning units for cooling and heating multi-room buildings with chilled or hot water. Fan draws fresh and recirculated air across coil, cooling or heating or dehumidifying it. Write for *Bulletin 9327.*



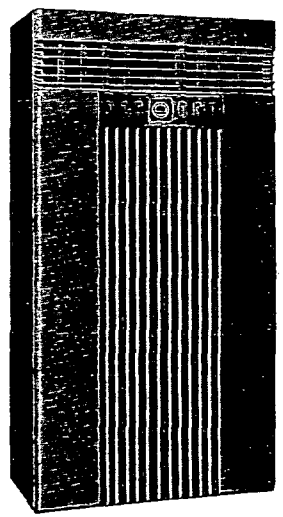
Inductor Units—small room air conditioners for cooling and heating multi-room buildings. Preconditioned primary air, supplied under pressure, induces a flow of room air through a coil in the unit, then into the primary air stream. Mixed air is discharged through top of unit. Write for *Bulletin 8827.*



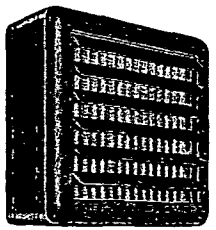
Sirocco High-Pressure Fans—especially adaptable to high-pressure air-conditioning systems. The multiple forward-curved blade design gives it a relatively high static pressure characteristic, so for a given duty it runs at about half the speed of backward-inclined blade fans. Data on request.



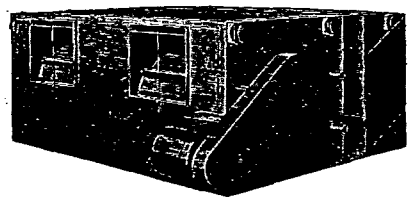
Tonrac®—single-stage hermetic centrifugal refrigerating machine. Maintains constant chilled-water temperature from 0 percent to 100 percent load. No cycling is required. Completely automatic load control—pneumatic or electronic. Induction motor suitable for any type of starting equipment with network and primary voltage. Bearings arranged for positive thrust and forced-feed lubrication. Single-level construction simplifies installation and maintenance. Request *Bulletin 1426.*



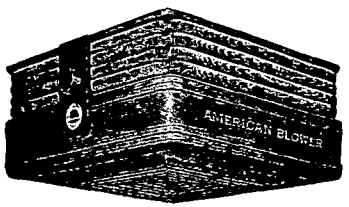
Packaged Air Conditioners—completely self-contained units that cool, filter, dehumidify, circulate air; heat, when equipped with heating coils. Hermetically sealed motor-compressor assemblies. Six sizes; 3 ton to 20 ton capacities. Exceptionally quiet. Request *Bulletin 6125.* Also air-cooled models, *Bulletin 8025.*



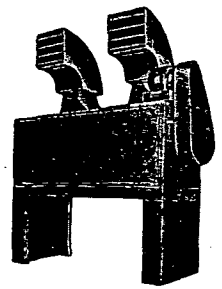
Venturafin® Horizontal Unit Heaters—available with one- or two-row standard steam and water coils, or serpentine hot-water coils (results in greater heating capacities at lower water volumes). These propeller-fan-type units heat quickly, quietly. Have adjustable-direction louvers. Request *Bulletin 9017.*



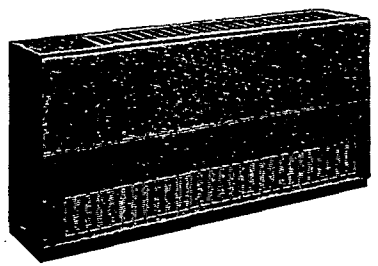
Heating, Ventilating and Multi-zone Units—with air filters and Aileron Control. For use where attractive, quiet, and economical heating and ventilating units are required. Wall, floor, or ceiling mounting. Units offer great flexibility of design and arrangement to meet specific needs. Request *Bulletin 8927.*



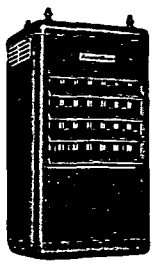
Venturafin Vertical Unit Heaters—can be mounted in minimum space at ceiling above craneways; has, as standard equipment, Equitemp Air Diffuser for maintaining desired even temperature. Capacities: 42,000 to 560,000 Btu at 2 lb steam pressure and 60 deg entering air. Standard steam and hot-water coils. Request *Bulletin 9017.*



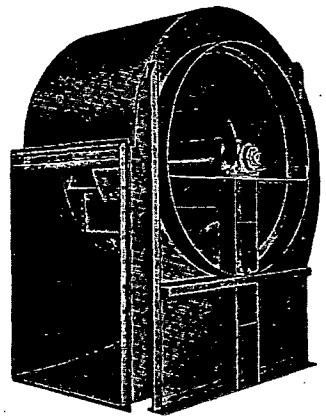
Industrial Heaters—feature quiet, low-speed centrifugal fans—V-belt driven to provide easy speed adjustments. Universal cowls adjust to direct air and heat where required. Four arrangements: wall, horizontal, inverted, and floor (shown). Request *Bulletin 7727.*



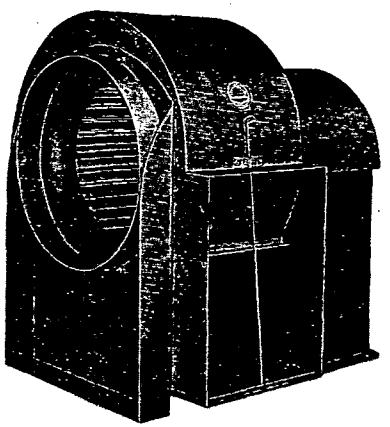
Cabinet Heaters—for offices, lobbies, showrooms, corridors—any application where appearance is important. Heavy-gauge steel cabinet houses high capacity steam or hot-water heating element and two or more centrifugal fans. Flexibility of mounting and inlet-discharge locations make numerous applications possible. Write for *Bulletin 9617.*



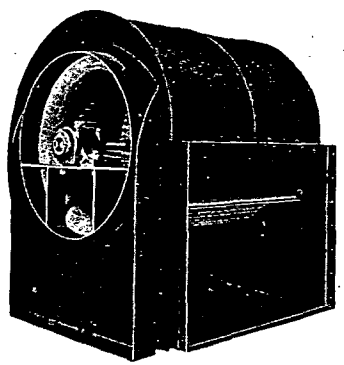
Gas Fired Heaters—self-contained units for clean, automatic, instantaneous heating. Adjustable louvers assure efficient heat distribution throughout the work area. *A. G. A.* approved. Write for *Bulletin 8517.*



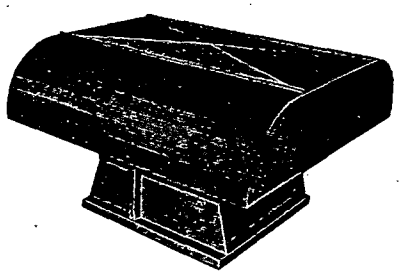
HS Fans—used for ventilating, air conditioning, and in commercial and industrial applications such as: drying, heating, fume removal, and processing systems. They are available in both single- and double-inlet designs and may be specified for clockwise or counter-clockwise rotation in any of eight standard discharges. Single-inlet fans are built in seven standard arrangements, double-inlet fans in two arrangements. Write for *Bulletin A-605.*



Utility Sets—complete packaged units, directly connected or with V-belt overhung drive, for duct applications. Belt driven units available with Sirocco forward curved blade wheels or HS backwardly inclined blade wheels having nonoverloading power characteristics. Sizes for a wide variety of ventilating problems. Quiet, compact. Complete technical information in *Bulletins 8314, 8414.*



Sirocco Multiblade Fans—operate at lower rotative and peripheral speeds for a given duty. They operate quietly at low speeds, and deliver more air per revolution. (Double-inlet model shown.) Request *Bulletin A-801.*



Power Roof Ventilators—weatherproof ventilators with V-belt drive and access door in fan housing. Model HRV exhaust ventilators (shown) are hood type, and have automatic shutters and 4-way discharge. Model VRV exhaust ventilators are upblast type and expel air high above roof, minimizing roof deterioration and preventing re-entry of air into supply ventilators. Model SSV supply ventilators, hood type, have motor-operated dampers and perimeter intake. Write for *Bulletin 3904.*

American Moistening Company

Cleveland, N. C.

ATLANTA, GA. . PROVIDENCE, R.I. . TORONTO, ONT.

AMCO

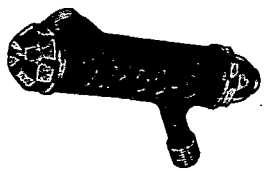
TEXTILE and INDUSTRIAL HUMIDIFICATION AIR CONDITIONING SYSTEMS

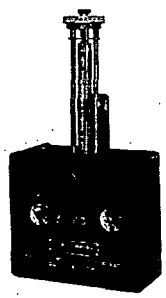
The American Moistening Company has been designing and manufacturing humidifying equipment for the textile industry since 1888. During this time, Amco has developed and introduced new equipment and control devices to solve atmospheric problems created by motorization, high speeds, new fibers, new machines, new techniques, and operation in varied climates.

Four basic advantages are to be found in all Amco products: superior performance, economy, durability, and simplicity with consequent ease of maintenance. The combination of all four is found throughout the complete Amco line.

AMCO ATOMIZER NO. 6

The automatically self-cleaning Amco No. 6 micro-spray atomizer delivers a smoke-like spray to a great distance for balanced distribution of humidity. Its fine performance and ease of maintenance make this atomizer popular for new installations and replacements. These units require compressed air for operation and obtain their water supply from a constant level, float controlled tank and piping.



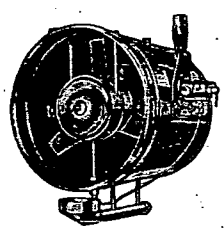


AMCO HUMIDITY CONTROL

Model K-2, improved through redesign, maintains room humidity within the closest practical limits under both normal and unusual conditions. An Electro-Pneumatic mechanism, operating in cushion-mounted jeweled bearings controlled by two humidity-sensing membranes, regulates atomizer operating periods. A variable change anticipating mechanism compensates for mechanical lag and the time required for water vapor to mix with and change the relative humidity of the room atmosphere. This results in humidity control within limits not previously attained in a rugged, dependable device.

AMCO MODEL 35 HUMIDIFIER

Ideally suited for areas of light heat concentration, especially where cost of other types of humidification might be prohibitive. Propeller-type fan draws room air into rear of unit and forces it forward around periphery of atomizer disc—thus carrying atomized water into room where it quickly evaporates into surrounding atmosphere. Housed in stainless steel. Requires only simple water and electrical connections. May be adjusted to evaporate from zero to 22 pounds of water per hour while circulating 1200 cu ft of air per minute. Simple and reliable, it requires a minimum of maintenance.



ARMSTRONG MACHINE WORKS

851 Maple St., Three Rivers, Mich.

GUARANTEED: SATISFACTION OR YOUR MONEY BACK

Available from Factory Representatives in Major Cities, or Direct from Factory

Automatic Steam Humidifiers—Air-Modulating or Electric Operated

For Comfort Air Conditioning or Protection of Hygroscopic Materials in Industrial Plants

Armstrong Humidifiers are large capacity, automatic units which discharge dry steam into duct systems or directly into space humidified. Installation is simple. Maintenance is negligible.

Advantages of Steam Humidification

1—Steam is ready-made vapor which merely has to be mixed with the air.
2—Steam deposits no mineral dust such as comes from evaporated water droplets.
3—Steam can be shut off as soon as the control is satisfied—no over-run.
4—Steam comes on immediately when control calls for humidity.
5—Steam creates no sanitation or odor problems. Duct system and air-handling units are always dry.
6—With steam there is no accumulation of moisture to promote growth of algae or bacteria.
7—Steam eliminates corrosion problems in ducts or fan housing.

Armstrong Humidifier Advantages

1—Low cost and large capacity.
2—Minimum maintenance, proved by thousands of installations.
3—Quiet operation.
4—No carry-over or drip.
5—Immediate response to control
6—Low operating expense.
7—Guaranteed tabulated capacities.
8—Selection is simple and easy.
9—Small space required makes duct design easy.
10—Easy installation.

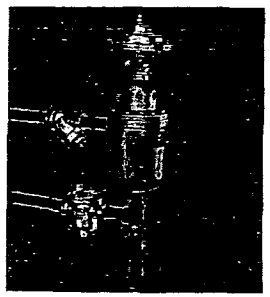
Size and List Price Range

Air-Operated Units

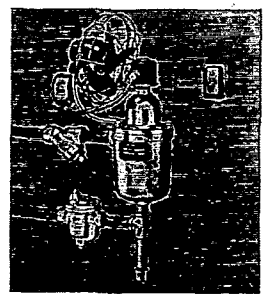
Model	List Price	Capacity, lb/hr
AM-32	229.25	31 to 150
AM-33	316.00	60 to 320
DA-34	593.00	145 to 1425

Electric Units

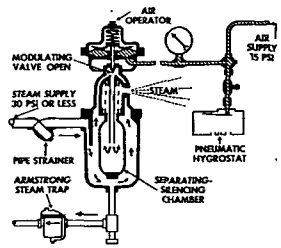
Model	List Price	Capacity, lb/hr
CL-2	170.00	25 to 76
CL-3	295.00	60 to 170
CK-4	545.00	145 to 1280



AM-32 Air-Modulating Humidifier complete with strainer, trap and hygrostat. AM-33 similar.



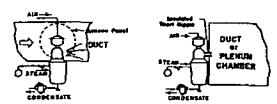
CL-2 Humidifier complete with fan and motor, humidistat, fine mesh strainer and trap. CL-3 similar.



Operation

Air-Operated Units—When relative humidity drops slightly below desired level, the pneumatic hygrostat increases air pressure in operator to open modulating valve in proportion to demand. When demand is satisfied air bleed closes the valve. Air-operated units usually discharge to ducts or are installed in plenums.

Electric Units—When relative humidity drops slightly below desired level, humidistat energizes solenoid to open steam valve and start fan. Steam flows through muffler to atmosphere and is dispersed by fan. When desired humidity is reached, humidistat closes valve and stops fan. Electrically operated units normally discharge directly into atmosphere.



Simple duct installations of air operated units

Automatic Evaporative-Type Humidifiers

For Residential, Commercial and Industrial Service where the direct use of steam is impractical. Advantages:

BIG CAPACITY—up to 16 pints/hr.

AUTOMATIC CONTROL AND OPERATION—set it and forget it.

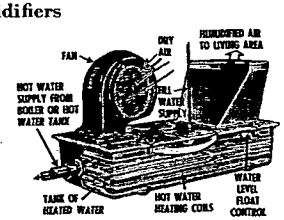
USE WITH ANY HEATING SYSTEM—hot water, steam or warm air.

FLEXIBLE INSTALLATION—Can be suspended from ceiling, mounted on floor stand, or attached to warm air duct. Enclosure available.

DUST FREE HUMIDIFICATION—"distilled" water vapor is used.

LIFE-TIME CONSTRUCTION—Fiberglass tank and corrosion resistant parts. List prices start at $195. *Bulletin No. 902* gives further information.

How It Works—When relative humidity drops slightly below humidistat setting, humidistat starts fan, blowing air across surface of heated water and thence to area to be humidified, or into a furnace plenum. Water level is kept constant by automatic float control.



Bahnson Company

ENGINEERS AND MANUFACTURERS

WINSTON-SALEM, N. C.

AIR CONDITIONING · HUMIDIFICATION · CLEANING · VACUUM COLLECTION · CREELS

1. AIR WASHER—Bahnson Type Y and Type DC Air Washers cleanse, cool, and provide humidity control. Available in standard sizes from *6,000 to 250,000 cfm*, they can be provided with spray systems best suited for the specific application. Write for *Catalog 1A*

2. TYPE E HUMIDIFIER—A self-contained unit with high evaporative capacity for installation in all types of industrial and commercial spaces requiring automatically controlled humidification. Evaporates up to *3 gph* with 360 degree radial distribution of moisture. Requires only water supply and electrical connection. Write for *Catalog 16A*

3. CENTRIFUGAL HUMIDIFIER—A self-contained unit for industrial use. Evaporation up to *12 gph* plus directional air flow with fractional horsepower motor. Installed singly or in groups with individual or group automatic control. Requires only water supply, drain, and electrical connection. Write for *Catalog 2A*

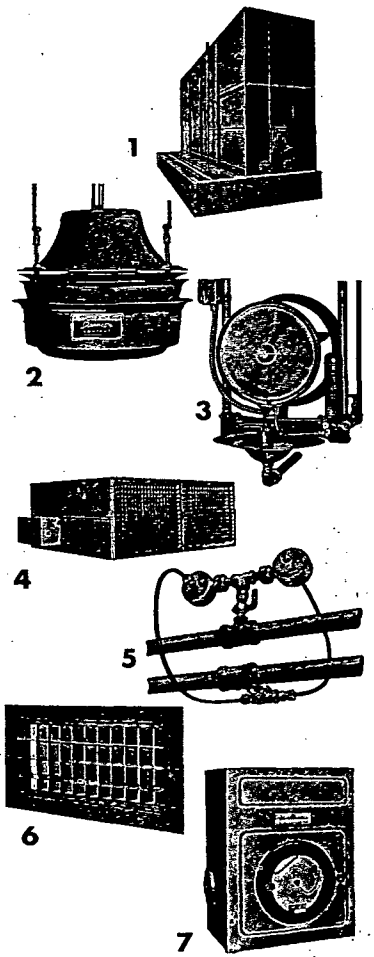
4. TYPE BA-2 HUMIDIFIER—A high-capacity unit for overhead suspension in commercial and industrial applications. Adjustable grilles give complete directional control of moist air delivered. Automatically regulated, the unit provides air flow, humidification and filtering. Requires only water supply, drain, and electrical connection; capacity up to *18 gph*. Write for *Catalog 17A*

5. TYPE ESC-2 ATOMIZER—A self-cleaning pneumatic atomizer of quality design for direct humidification or supplementary evaporation. Uses air and water under pressure to provide high evaporation and fine spray quality, with minimum air requirements. Modulating control regulates evaporation in direct proportion to the room requirements. Write for *Catalog 5A*

6. GRILLES—Bahnson Type C and D Grilles are designed for air discharge with removable core for easier cleaning and access to distribution duct. The air-foil type blades in the grilles are wide-spaced for both horizontal and vertical blade deflection and are individually adjustable by hand for air flow control. Constructed of extruded aluminum, blades are mounted on 1½ in. centers and are 1½ in. in width. Type C Grille has flush type face mounting; Type D Grille blade section projects forward permitting unobstructed air flow in distribution duct. Write for *Catalog 6A*

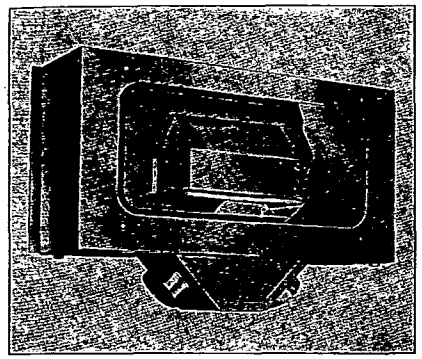
7. HUMIDITY CONTROLS—Bahnson Controls are offered in either electric or pneumatic design. The control consists of a highly sensitive humidistat which actuates an electric or pneumatic relay in response to humidity changes in the conditioned area. The humidistat is mounted in an aspirated cabinet which constantly draws a sample of room air over the hygroscopic element. Manual selector switches in the cabinet provide automatic functioning of motor-operated valves supplying water or steam to humidifying units. Write for *Catalog 9A*

OTHER BAHNSON MANUFACTURED EQUIPMENT: portable humidifiers, steam humidifiers, air washer suction strainers, spray nozzles, and complete industrial air conditioning systems. In addition Bahnson manufactures vacuum collection and cleaning equipment and creels. Write for descriptive literature on your requirements.

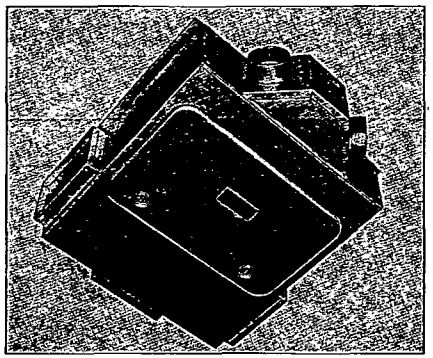


BUENSOD-STACEY, INCORPORATED

45 WEST 18th STREET, NEW YORK 11, NEW YORK • CHARLOTTE, NORTH CAROLINA · RICHMOND, VIRGINIA



Type V3. Window-Wall Dual Duct unit (showing automatic volume control).



Type H. Overhead Dual Duct unit.

BUENSOD-STACEY AIR HANDLING UNITS WITH SELF-CONTAINED AUTOMATIC VOLUME REGULATION

Automatic volume control is a self-contained feature of every Buensod-Stacey air handling unit. By simple mechanical means it maintains a constant flow of air through each unit in the system within plus or minus 5% of the desired amount even though static pressure at the unit's inlet may vary as much as 8" wg. This makes it impossible for any zone in the system to become unbalanced. Automatic volume control operates *without* the need of any outside power source, sensing instrument or complicated arrangement of linkage.

SIMPLE TO INSTALL AND ADJUST

A screw setting of spring tension at the time of installation and the unit is permanently set for the specified volume. If, after installation, a change of air volume is desired, it can be made quickly by adjusting one nut. No linkage. The mixing valve is designed for direct, straight-line movement. On dual-duct units, compressed air from a local thermostat controls the mixing valve to provide the desired room temperature.

DUAL DUCT AIR CONDITIONING SYSTEMS

This method of air conditioning utilizes hot and cold air in parallel conduits. The two streams of air are carried throughout a building separately and are blended in each conditioned zone by an air mixing valve. Temperature is controlled locally by a thermostat. The occupant may select the temperature at which he is most comfortable. Without adequate automatic volume control in each unit zones become unbalanced very easily. It is much like what happens when a shower bath valve is opened on the ground floor of a building and pressure drops on the top floor. Automatic volume control prevents this.

TYPE V3 WINDOW-WALL UNIT

Designed for installation under windows. It discharges vertically upward . . . quietly and without draft.

TYPE H OVERHEAD UNIT

Intended for use above hung ceilings, it supplies air through conventional side wall outlets or overhead diffusers.

SINGLE DUCT AIR CONDITIONING

This is Buensod-Stacey's newest air handling unit. It has been designed for use with central systems where the air temperature is pre-determined by zone thermostat and a single stream of air is carried to each zone for distribution. When such an arrangement is used, the Type HSD unit, by using a volume regulating device as the valve, saves much field labor normally required for adjustment and balancing of the system.

All Buensod-Stacey Air Handling Units, both Single Duct and Dual Duct, are now available in aluminum. Where weight and handling are factors, this construction shows economy, and where corrosion is a factor, aluminum construction has many advantages.

HIGH PRESSURE DEHUMIDIFIERS

The development of a series of dehumidifiers for use with high pressure Dual Duct and Single Duct systems has proven to be essential. These factory insulated, specially braced, sprayed coil dehumidifiers are designed for pressures up to 10" wg.

BUENSOD-STACEY INCORPORATED

Literature describing the equipment mentioned here is available. To obtain information about it, write Buensod-Stacey or communicate with our nearest sales representative.

Carrier

Carrier Corporation

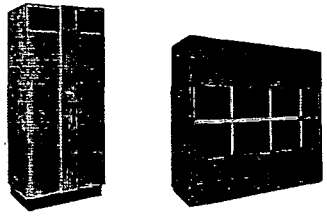
Carrier Parkway · Syracuse 1, New York

AIR CONDITIONING
REFRIGERATION
INDUSTRIAL HEATING

MARINE DEPARTMENT
385 Madison Avenue
New York 17, N. Y.

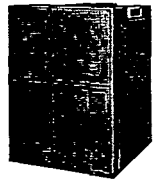
INTERNATIONAL DIVISION
385 Madison Avenue
New York 17, N. Y.

Branch Offices and Dealers in all principal cities



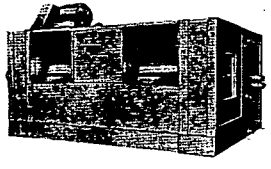
Weathermakers

These versatile self-contained air conditioners are excellent for use either singly or in multiples in all types of commercial and industrial buildings. Compact, attractively styled for use in or out of conditioned space, with or without ducts. Heating coils optional. Available in eight sizes, air or water-cooled models with capacities of 3, 5, 7½, 10, 15, 20, 25 and 30 tons.



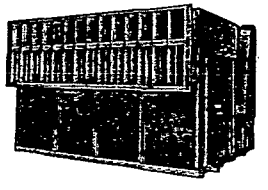
Year-Round Weathermakers

Complete year round air conditioning units for homes and smaller commercial and industrial buildings. All the elements for cooling, heating, filtering, circulation, humidifying and dehumidifying in one compact package. Air or water cooled models available with cooling capacities from 5 to 7½ tons; gas heating capacities are from 100,000 to 190,000 Btuh input.



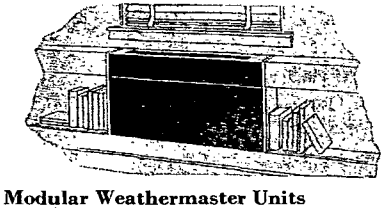
Single Zone Weathermakers

Central station fan-coil air conditioning units for cooling and heating systems using remote sources of refrigeration and heat. *Building-block design offers extreme* flexibility of application. Adaptable to floor or overhead mounting in horizontal or vertical arrangements. In eight sizes with direct expansion, brine or chilled water coils. Capacities from 2400 to 40,750 cfm.



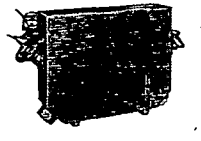
Multi-Zone Weathermakers

For air conditioning several zones of a large building with a single unit, using remote sources of refrigeration and heat. The multiple mixing dampers connect to hot and cold air plenums for independent temperature control of each zone. Eight sizes with coils for direct expansion, chilled water or brine and steam or hot water. Capacities range from 2400 to 40,750 cfm.



Modular Weathermaster Units

The high induction Weathermaster System uses these room terminals with primary air for cooling and ventilating and chilled water for supplemental cooling or heating. In four base unit sizes for furred-in application or with metal enclosures for floor-mounted or wall-hung installation. Adaptable to all types of fenestration, stool heights and wall construction.



All-Air Weathermaster Units

These high velocity room terminals combine efficient sound attenuation with effective control for every type of all-air system currently popular, including dual duct, single duct (with or without reheat), and dual conduit. Five models in six sizes from 50 to 1700 cfm offer choice for under-window or ceiling installation for both exterior and interior zone application.

Carrier

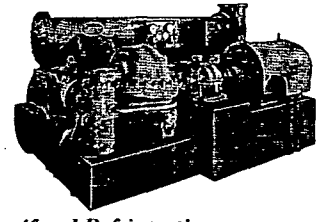
Carrier Corporation

Carrier Parkway · Syracuse 1, New York

Branch Offices and Dealers in all principal cities

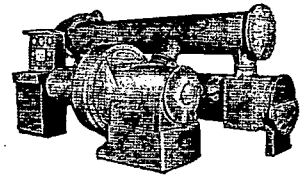
AIR CONDITIONING
REFRIGERATION
INDUSTRIAL HEATING

Refer to page •130 for Carrier Unit Heaters
See Sweet's Architectural, Plant Engineering or Industrial Construction Files or write Carrier Corporation, Syracuse 1, New York for specific literature.



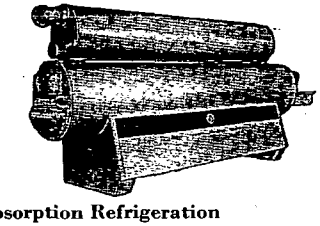
Centrifugal Refrigeration

For large scale air conditioning, process refrigeration, direct or indirect cooling of water, brines, hydrocarbons and other liquids. This compact refrigerating machine is comprised of an evaporator, condenser, and multi-stage centrifugal compressor. Available in a wide range of sizes with cooling capacities from 100 to 4,000 tons, for use with motor or steam turbine drive.



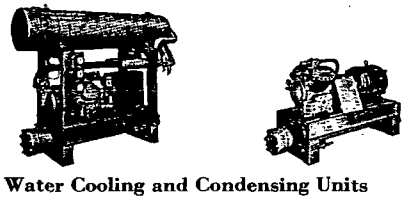
Hermetic Centrifugal Refrigeration

A compact, completely integrated water cooling machine designed especially for air conditioning or process liquid cooling applications. Features include a sealed, refrigerant cooled motor-compressor unit, automatic electronic controls, low demand motor circuiting and hydraulically powered capacity regulating vanes. In cooling capacities from 90 to 1500 tons.



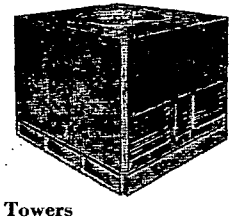
Absorption Refrigeration

A heat operated refrigerating machine, using low-pressure steam or high-temperature water to chill water for air conditioning or process liquid cooling applications. Uses plain water as the refrigerant and lithium bromide as the absorbent. Operation can be fully automatic from full load to zero capacity. Thirteen sizes in cooling capacity range from 60 to 700 tons.



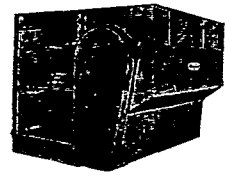
Water Cooling and Condensing Units

Completely packaged water-cooling machines include compact assembly of cooler, condenser, motor, compressor, refrigerant piping and control center all pre-engineered, assembled and factory tested. Capacities range from 5 to 139 tons. Compressors and condensing units for medium and low temperature applications range from 5 to 150 hp; direct, belt drive or hermetic.



Cooling Towers

Carrier induced-draft towers present an extremely compact design to reduce operating weight, structural support and maintenance. Galvanized steel construction with plastic fortified cellulose fibre fill insures long service life. Fan has adjustable pitch cast aluminum blades with close coupled, in-line motor drive. Single or multiple cell towers in sizes from 100 tons up.



Evaporative Condensers

Fan-coil type refrigerant condensers for use with refrigerating compressors to supplant water-cooled condensers or cooling towers. Designed for outdoor or indoor installation, they provide an efficient, economical means of heat disposal where problems exist on water supply and disposal. The complete range of sizes covers refrigeration capacities from 5 to 250 tons.

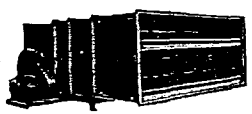
CLARAGE FAN COMPANY

Kalamazoo, Michigan

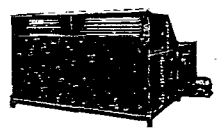
Air Handling and Conditioning Equipment

Sales Engineering Offices in All Principal Cities

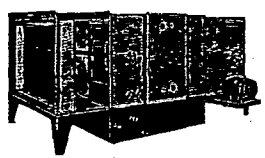
Since 1913 Clarage has been a leading manufacturer of air handling and conditioning equipment. By specializing—building a complete line of this equipment exclusively—Clarage has a concentrated "know how" that means important dividends to the user. (See "Air System Equipment—Fans and Blowers" for other Clarage products.)



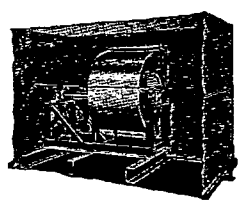
DRAW-THRU MULTITHERMS (Design 1). For both comfort and process applications. Can be arranged to handle applications ranging from simple ventilation to multi-function air conditioning. Horizontal and vertical arrangements. 12 sizes. Capacities 500 to 35,000 cfm.



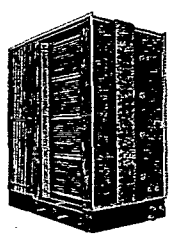
BLOW-THRU MULTITHERMS (Design 1). For central station zone control air conditioning. A single unit can handle different needs in as many as 8 different areas simultaneously. Hot and cold air is mixed to prevent stratification. Horizontal arrangement only. 12 sizes. Capacities 500 to 35,000 cfm.



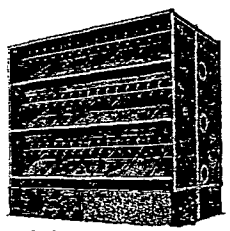
SPRAYED COIL MULTITHERMS (Design 1). Draw-Thru type for applications requiring extremely close control of both temperature and humidity, or where continuous flushing of coil surfaces is desired. Capacities 500 to 25,000 cfm. Unit shown with sprayed coil section exposed.



HI-STATIC MULTITHERMS (Design 3). Principally for conduit type systems where high duct velocities and pressures to 8 in. sp are encountered. Horizontal arrangement only. Cut-away view shown. Capacities 2,500 to 22,000 cfm. Sprayed coil units also available.



UNICOIL UNITS. Sprayed coil equipment principally designed for cooling and dehumidifying service in central station air conditioning systems. Capacities, based on 500 fpm air velocity, range from 3,000 to 41,600 cfm.



AIR WASHERS. Capillary® washer (shown) primarily for air cleaning efficiency in comfort and process central station air conditioning systems. Equipped with capillary cells. Spray type for comfort air conditioning, dust removal, and product reclamation.

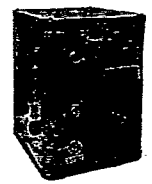
Desomatic Products inc.

1109 West Broad Street

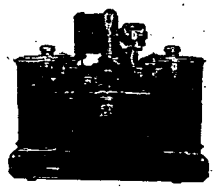
Falls Church, Virginia

Engineers and Manufacturers

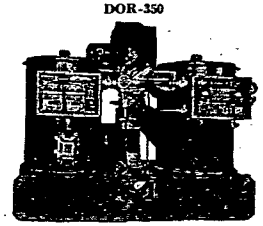
Representatives in Principal U. S. Cities and Abroad



SOR-8



DOR-350



DOR-800

Desomatic Products, Inc. manufactures industrial dehumidification equipment for commercial and Government use.

Refrigeration as well as adsorption units are manufactured for storage rooms, underground caves and mines, cargo holds on ships, laboratories, packaging rooms and for process control. Some applications require combination refrigeration and adsorption systems, with refrigeration for moisture removal from warm humid air and adsorption for dehumidification of medium to low dew-point air.

Equipment is built for atmospheric as well as elevated pressures. Standard stock models available as well as custom engineered equipment.

	SOR-8	DOR-38	DOR-100	DOR-350	DOR-800	AC-3000	AC-5000
Air Flow cfm	25	100	175	500	1500	3000	5000
External Pressure	⅛" W.G.	½" W.G.	½" W.G.	1" W.G.	1" W.G.	1" W.G.	1" W.G.
Voltage	110 V, 1 Ph	110 V, 1 Ph	220 V, 3 Ph	440 V, 3 Ph	440 V, 3 Ph	440 V, 3 Ph	440 V, 3 Ph
Load Kw	1	1.4	3	9	25	60*	100*
Reactivation Energy	Electric	Electric	Electric	Electric	Electric	Electric Gas Steam	Electric Gas Steam
Capacity 70 deg—35 per-cent lb/hr	¼	1	2.5	7.5	22	52	87
Weight, Lbs.	48	375	425	1150	4000	6000	9500
Length, Inches	15	54	54	70	108	148	177
Width, Inches	13	20	20	25	61.5	81	81
Height, Inches	18	42.5	42.5	60	92.5	91	91

* Electric reactivation

drayer hanson

(Division of National-U. S. Radiator Corporation)

Factory and General Sales Office

3301 Medford St. **Los Angeles 63, California**

Representatives in Principal Cities—Domestic and Abroad

Manufacturers of Air Conditioning and Refrigeration Equipment

REFRIGERATION AND AIR CONDITIONING

THE B-I-G LINE-UP . . . 23 FULL LINES!

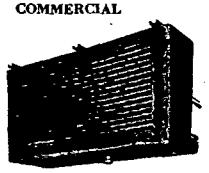
d-h AIR CONDITIONING:

SPOTAIRE ROOM - by - ROOM AIR CONDITIONERS: #1 d-h LRC's: basic unit, concealed or deluxe cabinet; 4 models, 200 through 600 cfm. #2. VRC's: concealed or deluxe consoles; 4 models, 200 through 600 cfm. #3. d-h HRC's: 3 suspended types, 19 models, 300 through 1750 cfm.
AIR HANDLING UNITS: #4. d-h HH-Series: ceiling suspended. #5. d-h HHV-Series: floor mounted. Both: 14 models, 624 through 28,000 cfm.
HIGH PRESSURE BLOW THROUGH: #6. d-h HCD and HPF: Hot & Cold Deck, 10 models, 1752 through 32,250 cfm.
VENTILATING UNITS: #7. d-h AM: 1752 through 32,250 cfm. 9 sizes.
MULTIZONE TYPES: #8. d-h **FLEXAZONE:** for simultaneous, independent, variable heating, cooling, ventilating; 1752 through 32,250 cfm. 9 sizes.
PACKAGED AIR CONDITIONERS: #9. d-h AECR: with built-in evaporative condenser. #10 d-h SCR: with water-cooled condenser. Both, 7½ through 75 hp.
PACKAGED STORE COOLERS: #11. d-h DYNA-PAC & ROYAL-AIRE: 2 through 15 ton.
PACKAGED WATER CHILLERS: #12. d-h CWG: 7½ through 75 hp. #13. d-h CWG-E: attached evaporative condenser. Both 7½ through 75 hp.
EVAPORATIVE CONDENSER: #14. d-h **PERMA-FAN:** 13 models; 5 through 110 ton.
COILS: #15. d-h Extended surface; steam, water, DX, Ammonia, Brine. #16. TYPE "H"; small applications, DX or chilled water.
COOLING TOWERS: #17. d-h WMT: 13 models, 5 through 100 ton.
AIR-COOLED CONDENSERS: #18. d-h ACC: 13 models, 3 through 80 ton.

d-h COMMERCIAL REFRIGERATION:

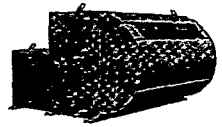
UNIT COOLERS: #19. d-h **FLOCOLD UNIT COOLERS:** over 34 F; 15 models; 750 through 11,000 cfm. #20. d-h **FLOCOLD:** under 34 F, water defrost, 11 models, 750 through 11,000 cfm. #21. d-h **FLOCOLD:** over and under 34 F water-defrost ammonia, 7 models, 2000 through 13,200 cfm. #22. d-h **FLOCOLD HOT GAS:** 11 models, 750 through 11,000 cfm. #23. d-h SPASAVER: over 34 F. 9 models, 730 through 4400 cfm. #24. d-h **HOT SHOT: AUTOMATIC ELECTRIC DEFROST:** under 34 F, 6 models, 700 through 3700 cfm. #25. d-h HRC; **MEAT CUTTING & PACKAGING ROOM UNITS:** 14 models, 500 through 1750 cfm. #26. d-h **FLOCOLD HOT PAN; FRUIT/VEG./MEAT ROOMS:** 11 models, 750 through 11000 cfm.
PRODUCT COOLERS: #27. d-h **FLOCOLD (FT & CT; FTWD & CTWD; FTAF & CTAF):** 9 models; 1314 to 24,000.
REPLACEMENT HOUSING: #28. d-h **SPASAVER:** kit to modernize on-job unit coolers.

COMMERCIAL



"FLOCOLD"

INDUSTRIAL

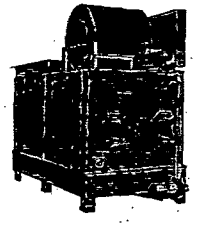


"HCD" HIGH PRESSURE BLOW THRU

RESIDENTIAL



"SPOTAIRE LRC"



"PERMA-FAN" EVAPORATIVE CONDENSER

Dryomatic DRY CONDITIONER — DRYOMATIC CORPORATION

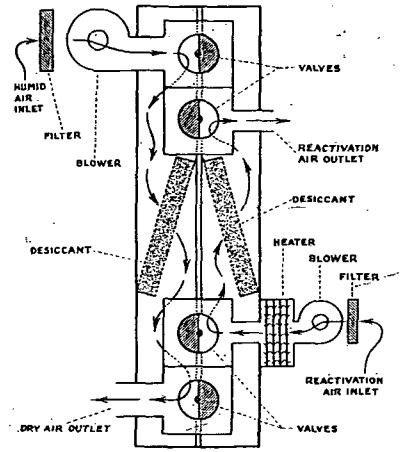
806 N. FAIRFAX STREET, ALEXANDRIA, VIRGINA

MANUFACTURERS OF COMMERCIAL, INDUSTRIAL AND HOME DEHUMIDIFIERS

Representatives in Principal U. S. Cities, Canada, & Abroad

DRYOMATIC manufactures a complete line of automatic dehumidifiers for accurate control of humidity in laboratory, processing, and storage operations. Desiccant type industrial and commercial air dryers are produced in a variety of standard models, and household dehumidifiers of both the mechanical and desiccant type are stocked in various capacities.

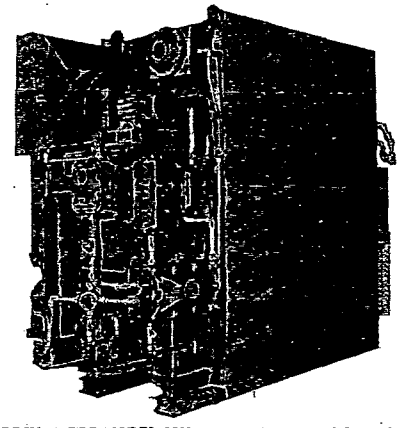
SCHEMATIC DIAGRAM



MODEL 1500—Schematic Diagram showing basic principle of design of all Dryomatic Dry Conditioners

DRYOMATIC dehumidifiers are sturdy, time-tested package units which automatically and continuously remove moisture from the air through the use of stable, solid adsorbents such as silica gel and activated alumina. These desiccants are self-reactivated upon saturation by electric heating elements. Regeneration by steam or gas is available as optional equipment on larger models. Air handling capacities range from 25 cfm to 1500 cfm with water removal rates from 1 to 45 lb/hr. Units in other capacities can be built to meet individual specifications.

DRYOMATIC desiccant dehumidifiers operate over a temperature range from −40 F to plus 120 F. Their particular efficiency at low temperatures enables them to maintain low dew points without the use of reheat, making them suited for dehumidification of refrigerated areas in conjunction with existing air conditioning and cooling systems. Special units containing molecular sieve desiccant, for operating conditions above 120 F, are available. Machines may be installed either outside or inside the controlled area, and require only electrical power and two simple flexible hose or galvanized duct connections.
Write for further details, requesting Inquiry Data Sheet for engineering recommendations or consultation regarding any phase of humidity control.



DRYOMATIC MODEL 1500—adsorption type dehumidifier with steam reactivation, explosion-proof electrical fittings, and water-cooled after-cooler

MODEL NUMBER	TYPE	POWER	TYPICAL MOISTURE REMOVAL	ADSORPTION AIR FLOW	EFFECTIVE AREA (vapor sealed)
25 CT	Single Bed, Solid Adsorption	115V, 60 Cycle, Single Phase	11.9 lb Water/24 hrs @ 90 F & 75% R. H.	27 cfm	10,000 cu ft
100	Three Bed, Solid Adsorption	115V, 60 Cycle, Single Phase	40 lb Water/24 hrs @ 90 F & 75% R. H.	90 cfm	25,000 cu ft
105	Dual-Bed, Solid Adsorption	115V, 60 Cycle, Single Phase	48 lb Water/24 hrs @ 90 F & 75% R. H.	100 cfm	35,000 cu ft
150	Dual-Bed, Solid Adsorption	230 or 208V, Single or 3-Phase	104 lb Water/24 hrs @ 90 F & 75% R. H.	150 cfm	75,000 cu ft
L-400	Dual-Bed, Solid Adsorption (for low temp.)	230 or 208 V, Single or 3-Phase, Δ or Y	90 lb Water/24 hrs @ 40 & 75% R. H.	400 cfm	90,000 cu ft
1500	Dual-Bed, Solid Adsorption	230, 208, or 440V 3-Phase, Δ or Y	880 lb Water/24 hrs @ 90 & 75% R. H.	1500 cfm	700,000 cu ft

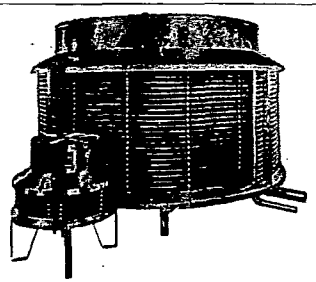
A DRYOMATIC DRY CONDITIONER FOR EVERY DEHUMIDIFYING NEED

f5 AIR CONDITIONING CORP.

Manufacturers of Air Cooled Fandaire Air Conditioners & Condensers

1815 South Maybelle • Tulsa, Oklahoma • CHerry 2-9266

Representatives and Distributors in Principal Cities



Fandaire condensers above are the 3 and 80 ton units

FANDAIRE AIR COOLED CONDENSERS

Available in 2 through 80 ton capacity

Conservatively rated, these condensers can be used on any application . . . either for new installation or as a replacement for outmoded equipment. While designed for outside installation, they work equally well inside. Their unique circular coil design delivers far greater operating efficiency than rectangular units. As standard equipment they incorporate many features normally sold as "extras."

FANDAIRE AIR COOLED AIR CONDITIONERS

Available in 2 through 10 ton capacity

Fandaire air cooled air conditioners are very good for commercial, residential and industrial installation. The unique circular coil construction delivers far greater operating efficiency than rectangular units. They can easily be combined with existing heating systems. Fandaire advanced design incorporates, without extra charge, many features normally sold as "extras." Available with a full five year warranty.

FANDAIRE FEATURES

on both the air conditioning and condenser units

GREATER FIN SURFACE EFFICIENCY

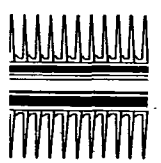
The fins are helically formed around a heavy-wall copper tube with a spacing of eight fins per inch to permit the free passage of lint and dust carrying air. This self-cleaning action helps maintain a higher operating efficiency. The shoulder of each fin is firmly swaged to the tube and to the adjoining fin. No foreign matter or corrosion can form between the tube and the shoulder to cause loss of heat transfer even after years of exposure to the elements. In addition, the shoulder provides a full 100 percent contact with the tube, guaranteeing the maximum amount of heat will be transferred to the fin. The fins are pyramidal in shape for sturdiness and to permit the best possible dissipation of the heat.

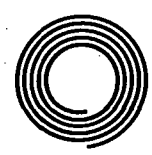
TRUE COUNTER FLOW

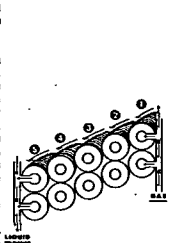
True counter flow is an action in which the hot refrigerant is gradually cooled by flowing against (or counter to) the cooling medium. In other words, the refrigerant flows hot to cool, or counter to the cooling medium which flows cool to hot. In Fandaire units, the hot gas enters the condenser tubes at the inlet header *inside* the condenser and makes five complete circles before leaving the tube at the outlet header. Since the cool air enters the condenser around the periphery of the coil and is pulled through the bank of tubes, the coolest gas is always in closest contact with the tubes carrying the coolest gas. This provides the best possible counterflow and assures a greater mean temperature at all times, giving a higher rate of heat transfer, lower condensing temperature and a greater degree of sub-cooling.

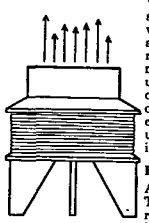
POSITIVE CONDENSATE DRAINAGE

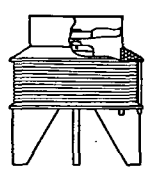
Each row or layer of tubes slopes downward at a 30 deg angle. This means the tubes at the outlet header are lower than the tubes at the inlet header. This slope assures a better drainage of all condensate as fast as it forms on the tube wall. No pockets of compressor oil can form to act as an insulator between the hot gas and the air cooled tube walls.











STRAIGHT THROUGH FLOW

Each circuit consists of one continuous length of a single tube which completely eliminates ALL pressure drop caused by return bends. Laboratory tests have shown that up to 60 percent of the pressure drop normally found in rectangular units is due to the short radius return bends necessary in their construction. Fandaire's long-sweeping, one-circuit construction approaches the almost non-existent pressure drop of a single pass unit while retaining all the advantages of a five-pass unit.

VERTICAL AIR DISCHARGE

Vertical air discharge is part of the standard construction. No air scoop or wind deflector is necessary. This vertical air discharge eliminates any tendency to re-circulate hot air and is a "must" in multiple installations. With Fandaire units there are no dead spots to be found on the face of the coil since the air is drawn around the circumference of the coil. Wind from any direction can be used, it is not necessary to face the units in the direction of the prevailing wind.

HOUSING

All Fandaire models are weatherproof. They are housed in attractive, corrosion resistant housings of a spun construction. Models up to 7½ ton are made of aluminum alloy—all other models are galvanized after construction for a durable, maintenance free finish. The air conditioners are supplied with a two-piece, wrap-around skirt. Many other features, normally sold as "extras" on other units, are incorporated in the standard Fandaire design. These include legs, mounting brackets, fan guards.

MOTORS AND FANS

Either direct drive or belt drive weather proof or totally enclosed, ball bearing, permanently lubricated motors are used on all Fandaire models. Fans are axial flow type and are constructed from plated, heavy-duty corrosion-resistant steel or aluminum.

Allied Chemical — GENERAL CHEMICAL DIVISION

40 Rector Street, New York 6, N. Y.

Manufacturer of **genetron®** *Super-Dry Refrigerants*

"Genetron" 12—White Label—CCl_2F_2—Dichlorodifluoromethane; and "Genetron" 22—Green Label—$CHClF_2$—Monochlorodifluoromethane. Used in virtually all types of refrigeration and air-conditioning equipment, large and small, household and industrial, direct and indirect expansion systems.

Typical units in which "Genetron" 12 and 22 are used: refrigerators, freezers, frozen food lockers, window air conditioners, home or office console units, large custom-built units for commerical comfort or industrial process, large store units, mobile units for transportation equipment, large home units for addition to present hot air heating systems.

"Genetron" 11—Orange Label—CCl_3F—Trichloromonofluoromethane. For industrial and commercial refrigeration and air-conditioning systems using single or multi-stage centrifugal compressors. Can also be used for either direct or indirect expansion-type systems.

"Genetron" 113—Purple Label—$C_2Cl_3F_3$—Trichlorotrifluoroethane. Used in 50-ton and larger centrifugal compressors, primarily for large comfort cooling systems, brine cooling systems and other commercial and industrial air conditioning and refrigeration systems.

"Genetron" 114a—Blue Label—$C_2Cl_2F_4$—Dichlorotetrafluoroethane. Used in centrifugal and rotary compressors for commercial, industrial and household refrigeration.

SELECTED PHYSICAL DATA

Selected Physical Data (Performance based on 5°F evaporator temperature and 86°F condenser temperature)	genetron 11	genetron 12	genetron 22	genetron 113	genetron 114a
Chemical Formula	CCl_3F	CCl_2F_2	$CHClF_2$	$C_2Cl_3F_3$	$C_2Cl_2F_4$
Molecular Weight	137.4	120.9	86.5	187.4	170.9
Boiling Pt. (°F) at 1 Atm. Pressure	74.7	−21.6	−41.4	117.6	37.6
Evaporator Pressure at 5°F (psig)	24.0*	11.5	23.3	27.9*	15.6*
Condensing Pressure at 86°F (psig)	3.6	93.3	150.8	12.9*	22.7
Freezing Point (°F) at 1 Atm. Pressure	−168	−252	−256	−31	−76
Critical Temperature (°F)	388	234	205	417	294
Critical Pressure (psi absolute)	635	597	716	495	478
Compressor Discharge Temperature (°F)	113	101	131	88	88
Compression Ratio (86°F/5°F)	6.24	4.08	4.06	8.02	5.33
Specific Volume of Saturated Vapor at 5°F (cu ft/lb)	13.97	1.46	1.25	27.04	4.04
Latent Heat of Vaporization at 5°F (Btu/lb)	84.0	68.2	93.6	70.6	60.2
Net Refrig. Effect of Liquid—86°F/5°F (Btu/lb)	67.5	50.0	69.3	53.7	43.0
Specific Heat of Liquid at 86°F (Btu/lb°F)	0.21	0.24	0.24	0.22	0.23
Specific Heat of Vapor at Constant Pressure of 1 Atm. & 86°F (Btu/lb°F)	0.13	0.15	0.15	0.15	0.16
Specific Heat Ratio at 86°F & 1 Atm. (k = Cp/Cv)	1.14	1.14	1.18	1.09	1.01
Coefficient of Performance	5.09	4.70	4.66	4.23	4.60
Horsepower/Ton Refrigeration	0.927	1.002	1.011	0.960	1.025
Refrigerant Circulated/Ton Refrig. (lbs/min.)	2.96	4.00	2.89	3.73	4.65
Liquid Circulated/Ton Refrig. (cu in/min.)	56.0	85.6	68.0	66.5	88.7
Compressor Displacement/Ton Refrig. (cfm)	36.33	5.83	3.60	100.76	18.78
Toxicity (Underwriters' Laboratories Group No.)	5A	6	5A	4-5	6
Flammability & Explosivity	None	None	None	None	None

* Inches of mercury vacuum.

- Guaranteed exceptionally low moisture content
- Non-corrosive to standard equipment materials
- Non-toxic, non-flammable, stable, safe
- Critical and freezing points well outside range of operating uses
- Solvent action on oil helps prevent solidification or congealing of lubricant
- Freely interchangeable and may be mixed in any proportions with comparable fluorinated hydrocarbons meeting the same strict refrigerant specifications
- Aid in lubrication of equipment; generally miscible with oil
- Available everywhere from refrigeration wholesalers throughout the country

QUICK FACTS on "Genetron" Super-Dry Refrigerants:

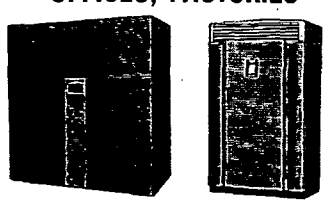
GENERAL ELECTRIC

AIR CONDITIONING DEPT.

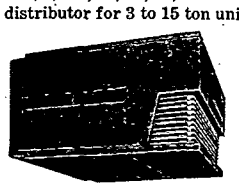
Tyler, Texas

Atlanta 3, Ga.—410 Red Rock Building.....JAckson 3-7447
Birmingham 9, Ala.—2846 South 18th St.....TRemont 9-3466
Chicago 54, Ill.—
The Merchandise Mart, Room 1117.....DElaware 7-2137
Cleveland 31, Ohio—Shaker Heights,
6900 Granger Rd.....LAfayette 4-4406
Dallas, Texas—Suite 2001, Trade Mart,
2100 Stemmons Freeway.....RIverside 8-6733
Detroit 21, Mich.—16247 Wyoming Ave.....DIamond 1-1459
Houston, Texas—2425 Broad St.....MIdway 4-5401
Jacksonville 5, Fla.—5266 Highway Ave.....EVergreen 7-3571
Kansas City 15, Kansas
3260 North 7th St., Trafficway.....MAyfair 1-3700
Linden, N. J.—1611 Elizabeth Ave. West.....WAbash 5-3600
Los Angeles 21, Calif.—
2958 East 46th St.....MAdison 7-4602
Louisville, Ky.—Appliance Park,
Room 106, Bldg. 5.....Ext. 2665, GLendale 4-7511
Memphis, Tenn.—500 South Front St.....JAckson 7-5531
New Orleans 19, La.—
4221 Bienville St.....GAlvez 5-5771
New York 22, N. Y.—
570 Lexington Ave., Rooms 301-2-3.....PLaza 1-1311
New York 16, N. Y.—150 East 42nd St.....PLaza 1-1311
International General Electric Company
Philadelphia 20, Pa.—
5660 Rising Sun Ave.....PIlgrim 2-2880
Salt Lake City, Utah—200 South Main St.....EMpire 4-1891
Burlingame (San Francisco), Calif.—
1649 Adrian Rd.....OXford 7-3321
Seattle 4, Wash.—705 Second Ave.....MAin 4-8300
Washington 5, D. C.—
777 14th St., N. W., Room 917.....EXecutive 3-3600

FOR STORES, OFFICES, FACTORIES

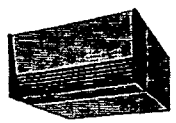


FLOOR MODELS. Self-contained—water-cooled. Capacities: 3, 5, 7½, 10, 15, 20, 25 and 30 tons. In-space air distributor for 3 to 15 ton units.



CEILING-MOUNTED. Self-contained—may also be shelf-mounted. Air-cooled in 3 and 5 ton capacities—water-cooled in 3, 5 and 7½ tons.

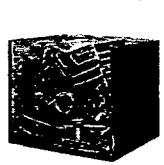
CEILING-MOUNTED. Split systems. Condensing unit air-cooled, weatherproofed for remote installation, indoors or out. Small, compact, lightweight air conditioner includes blower, evaporator filter. Capacities: 2½, 3, 4, 5, 7½ and 10 tons.

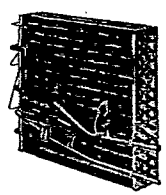


Zone-by-Zone installation permits individual control and servicing of each area. Accessory steam and hot water coils available for winter heating.

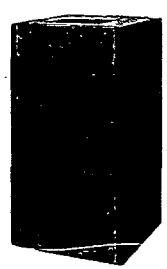
WHOLE-HOUSE RESIDENTIAL AIR CONDITIONERS

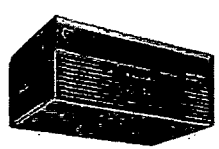
COOLING COILS to convert warm-air furnaces *into year 'round whole-house air conditioning.* Type "A" for low headroom, upflow or downflow—type "F" (flat) for upflow, downflow, horizontal. 5 sizes, 24,000 BTUH to 60,000 BTUH.





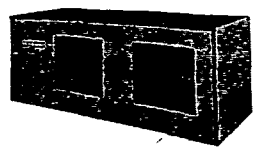
AIR HANDLING UNITS. Horizontal (shown with optional air diffuser) and vertical. For residential *and light commercial application when teamed with* remotely installed G-E air-cooled condensing unit. 2½ through 5 tons.



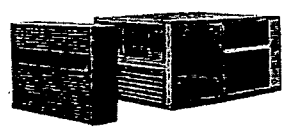


GENERAL ELECTRIC

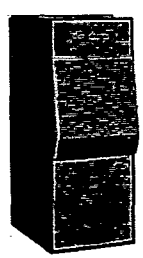
REMOTE CONDENSING UNITS. Factory-sealed, air-cooled, weatherproofed. For outdoor installation. For use with G-E cooling coils or air handling units. 6 models, 2 through 6 tons.



PACKAGED COOLING UNIT. *Compact, air-cooled,* self-contained—for homes with or without ducts. Decorative diffuser-filter frame for commercial in-space installation. Install in attic, crawl space, cellar, hallway—on roof, slab—through transom, wall. 2, 2½, and 3 ton sizes.



HOME HEATING UNITS



OIL FURNACE. High-pressure gun burner and other standard parts for easy servicing, replacement. Quick-heating "Vertifin" heat exchanger—dependable safety features. Smallest model takes only 28" x 22" floor space. All models fire-tested, assembled, wired at factory. *BTUH output, upflow,* 84,000 to 168,000—downflow-horizontal, 84,000 to 112,000

IMPERIAL GAS FURNACE

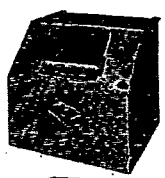
Cast-iron "Pinpoint" heat exchanger provides quick, economical heat. Simplified wiring, quiet performance, rugged construction. Smallest unit takes only 21" x 30⅛" floor space. BTUH input (upflow), 90,000 to 210,000.

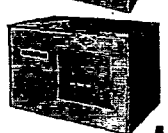


STEEL GAS FURNACE. Smallest unit takes only 15" x 28" floor space. Minimum clearances approved by AGA:—*sides, rear and combustible flooring,* zero—front and flue, 6"—above plenum, 1". Ideal for closet installation. Simplified wiring, quiet blower, advanced cast-iron burner design, "Thermal-Trap" heat exchanger. Two types—one for normal heating and air conditioning, one for above-average cooling loads. *BTUH input (upflow and downflow),* 75,000 to 150,000—(horizontal), 70,000 to 140,000.



ALL FURNACES *"grow" into year 'round* whole-house air conditioning by addition of G-E cooling coils. All handsomely styled in two-tone grav.





COOLS WITHOUT WATER, heats without fuel, all-automatic. Capacities (self-contained), 2½ and 3 tons—(split systems), 2½, 3 and 4 tons. Weathertron not available in all states.

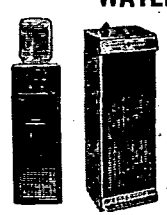
AIR-WALL* REGISTER SYSTEM

*Reg. Trademark of General Electric Company

G-E's perimeter system for distribution of conditioned air blankets walls with warm air in winter, cool in summer. No drafts.



WATER COOLERS



PRESSURE AND BOTTLE TYPES. Hot and cold combinations with and without refrigerated compartment. Cold water models with and without refrigerated compartments. Widest selection all safeguarded by G-E's protection plan.

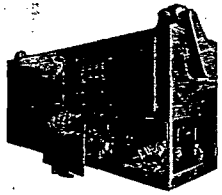
ALL PRODUCTS PROTECTED BY A GENERAL ELECTRIC WARRANTY
General Electric Company,
Air Conditioning Department, Tyler, Texas.

Governair Corporation

4840 North Sewell, Oklahoma City 4, Oklahoma

GOVERNAIR

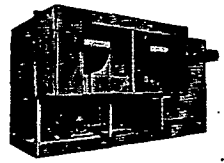
"Satisfabricated" Air Conditioners

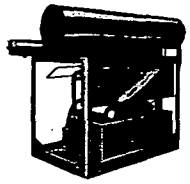


"SATISFABRICATED" for numerous choices of components and models.

SELF-CONTAINED AIR CONDITIONERS—completely packed air conditioners with evaporative condenser. Ready for connection to duct system—3 through 100 tons—multiple circuit refrigeration systems included—copper fitted condensers available.

SELF-CONTAINED MULTI-ZONE AIR CONDITIONERS—with evaporative condenser. Zoned conditioner section—hot and cold deck, with mixing dampers for each zone. Sizes 7½ through 80 tons—multiple circuit refrigeration systems included.





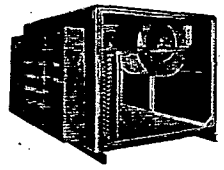
CABINET MODELS AVAILABLE with built-in or remote evaporative condenser.

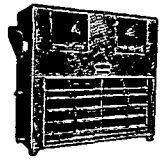
SELF-CONTAINED CHILLER UNITS

Standard equipment includes heat exchanger and built-in capacity reduction at no additional cost. 68 combinations available from 10 basic models. Capacity 10 through 120 tons. Factory tested before shipment.

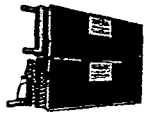
AIR UNIT CONDITIONERS

Available in horizontal GHC unit (shown), or GVC vertical unit. From 3 through 150 tons. In 2, 4, 5, 6, 8 or 10 rows of coil depth. Variable face velocities from 400 to 600 fpm.





FAN AND COIL CABINETS — cooling, heating, humidifying, dehumidifying, ventilating, 1 through 100 tons, 500 to 40,000 cfm, vertical or horizontal types. Governair, multi-zone fan and coil units are available in sizes ranging from 5 to 80 ton capacity, suitable for individual zoning.



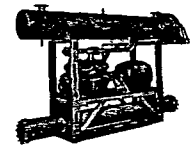
BLAST COILS—direct expansion, water steam—hydraulically expanded tubes for close fin spacing—mechanically expanded tubes for wide fin spacing—copper or aluminum fins—complete range of sizes.

The copyrighted term "Satisfabricated" is defined as follows: A method of purchasing air conditioning equipment which allows the engineer to determine the completed unit, fulfilling specified requirements and dimensions by selecting the specific components to be used.

Kennard Division of American Air Filter Co., Inc.

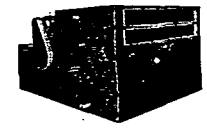
1270 North Price Road • St. Louis 14, Missouri

KENNARD



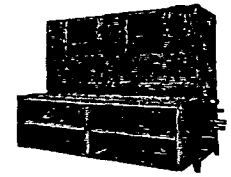
PACKAGED LIQUID CHILLERS

20 to 150 tons. Water and electrical connections only required for installation. Unloaded starting and multi-step capacity control. Automatic chilled water temperature control.



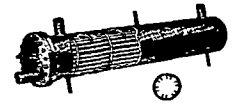
DUAL DUCT CENTRAL STATION AIR CONDITIONING UNITS

Normal and High Pressure Types, 7 sizes. Cooling Coil Face Areas up to 60 sq ft.



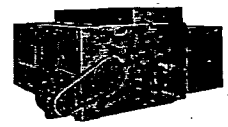
CENTRAL STATION AIR CONDITIONING UNITS

16 Sizes, 300 to 36,000 cfm. Cooling and heating coils up to 60 sq ft face area. Horizontal or Vertical. Single and double fan units. Flexibly adapted, by combining optional elements, to provide all or part of the air conditioning cycle, comfort or industrial. Choice of blower discharge positions simplifies ductwork, eliminates turns.



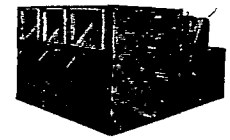
LIQUID CHILLERS

Direct Expansion, Shell and Tube Type. Internally-finned tubes. Capacities 2 to 200 tons. Kennard design eliminates refrigerant head gasket, prime source of refrigerant leakage. Controlled distribution of refrigerant to tube circuits. Rapid, positive response to control. Positive oil return.



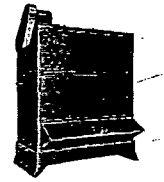
HEATING AND VENTILATING UNITS

16 Sizes, 300 to 48,000 cfm. Horizontal or Vertical, floor or wall mounting or ceiling suspension. Hot water or steam coils. Choice of blower discharge positions.



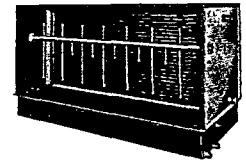
MULTI-ZONE CENTRAL STATION AIR CONDITIONING UNITS

9 Sizes. Cooling Coil face areas up to 60 sq ft. Choice of Direct Expansion or Chilled Water Coils for Cooling, and Steam or Hot Water Coils for heating. Standard units furnished with Zone Control Dampers and multiple individual zone outlets.



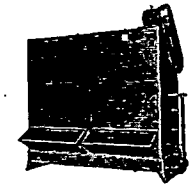
COOLING TOWERS

Outdoor or Indoor, 15 to 75 tons. Completely corrosion-resistant. Hot-dipped galvanized after fabrication. Stainless steel or bronze fastenings used throughout. Simple disassembly of blower, spray deck and sump pan sections facilitates field handling, hoisting and rigging.



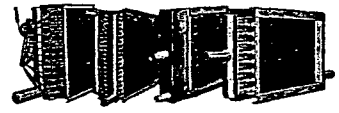
SPRAYED COIL DEHUMIDIFIERS

56 Sizes, Coil Face Areas up to 81 sq ft. Cooling, dehumidifying, evaporative cooling, and washed air, all in one compact unit. Direct expansion or water coils for cooling; steam or water coils for heating.



EVAPORATIVE CONDENSERS

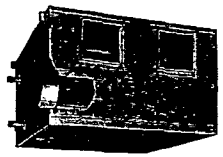
18 Sizes, 16 through 125 tons. Outdoor or Indoor. Fan wheels and housings, and sump pan, hot-dipped galvanized. Non-corroding, cleanable, swirl-type spray nozzles. Liquid sub-cooling coils available.



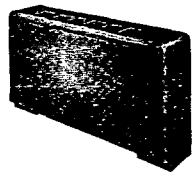
FINNED COILS

Direct Expansion Coils, Water Coils, Steam Distributing Tube Coils, Standard Steam Coils. Sizes up to 48 in. wide, 156-in. finned length.

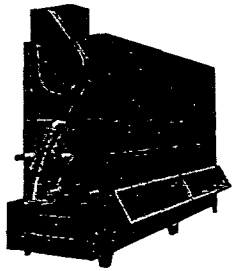
Marlo COIL CO. • 7100 S. GRAND BLVD. • ST. LOUIS 11, MO.



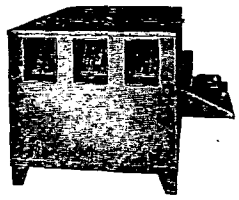
CENTRAL STATION AIR CONDITIONING UNITS cooling — heating — dehumidifying — humidifying. 38 sizes—1 to 175 tons —400 to 52,000 cfm. Ceiling, floor types. *All accessories.* *Bulletin 30*



REMOTE ROOM AIR CONDITIONING UNITS—Permit individual climate selection. No duct-work, only simple piping needed. Floor and ceiling type units. *Bulletin 33*



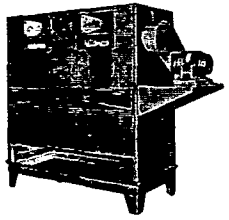
MULTI-ZONE UNIT—Complete winter and summer functions. Individual zone control—1,300 to 37,000 cfm. *Bulletin 31*



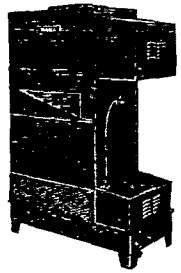
COOLING TOWERS AND EVAPORATIVE CONDENSERS—Industrial and commercial towers—20-100 tons—induced draft, wetted deck surface, water spray—indoor-outdoor. "Thrifti-Tower" adaptation for residential, light commercial use—2-16 tons.

Evaporative condensers, 3 to 250 tons —all refrigerants—all prime surface coils—no fins—quiet—indoor-outdoor. Also air cooled condensers.

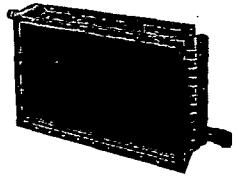
Bulletins 50 ,55, 60, 61



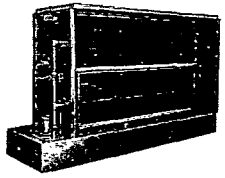
INDUSTRIAL COOLERS —15 unit sizes—1,000 to 26,500 cfm —water, electric and continuous brine spray defrosting. Ceiling, floor types. *Bulletin 40*



BLAST COILS—Air conditioning —industrial refrigeration—heating. Any material—all refrigerants—every application. Pressure-expanded staggered tubes, continuous plate-type fins. *Bulletins 10, 11*



SPRAY COIL DEHUMIDIFIER—washing, cleaning, humidifying, dehumidifying, heating and cooling. Air volumes from 600 to 76,000 cfm. *Bulletin 37*



UNIT COOLERS—*water and* electric defrosting—manual or automatic control. *Bulletins 21, 22*

NIAGARA BLOWER COMPANY

Service and Development in Air Engineering for Over 40 Years

Factory and General Offices: Buffalo, N. Y.—Executive Offices, New York, N. Y.

AIR ENGINEERING



Bulletin describes Niagara equipment for heat removal by the air method, cooling liquids and gases, condensing vapors; heat and moisture control by complete function air conditioning; vapor equilibrium by direct liquid contact.

BULLETIN No. 135

COOLING AND FREEZING



Bulletin describes Niagara "No-Frost" Method; production of low air temperatures for industrial uses, such as testing or processing at extreme sub-zero temperatures or for control of air moisture in atmospheres below the freezing point of water.

BULLETIN No. 95

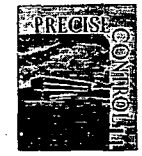
AERO HEAT EXCHANGERS



Bulletins describe evaporative cooling apparatus that keeps control of temperature within two degrees by modulating the flow of air. Saves 95 % of cooling water. Provides heat when needed.

BULLETIN No. 120
BULLETIN No. 124
BULLETIN No. 152

AIR CONDITIONING



Bulletins describe methods of obtaining precise control of temperature and moisture to help you make your product in any climate at any season. Illustrated with diagrams and photographs.

BULLETIN No. 122
BULLETIN No. 112

"NO-FROST" METHOD



Bulletin describes Niagara "No-Frost" Method of refrigeration for food freezing, frozen food warehousing, fresh food precooling and storage, giving completely automatic operation with no shut-down for defrosting.

BULLETIN No. 105

AERO AFTER COOLERS



Describes equipment for cooling compressed air or gases, holding temperatures below ambient atmospheric temperature to prevent condensation of moisture. Includes application to air liquefaction systems.

BULLETIN No. 130

AIR CONDITIONERS



Bulletin describes Niagara coil surface air conditioners for use with any refrigerant, offering any selection of air conditioning functions. Models available to match any load requirement.

BULLETIN No. 153

FAN COOLERS



Bulletins describe Niagara Fan Coolers, dry coil and brine spray types for use with brine, ammonia or freon refrigerants.

BULLETIN No. 110
BULLETIN No. 158

AERO VAPOR CONDENSERS



Providing high vacuum, independent of a large supply of cooling water. Solves the problems of water supply and disposal. Fully automatic. Saves power. Saves steam.

BULLETIN No. 139R

DRYING AND DEHUMIDIFYING



Bulletins illustrate air conditioning cabinet and drying installations including siccative coatings, paper, plastics, provisions, confectionery, pharmaceuticals, metallurgical uses.

BULLETIN No. 88
BULLETIN No. 112

AEROPASS CONDENSERS



Bulletins describe evaporative refrigerant condensers providing protection against scaling of tubes, controlling of compressor head pressures, saving condenser water.

BULLETIN No. 111
BULLETIN No. 123
BULLETIN No. 131

LIQUID COOLERS



Cooling for chilled water or solutions, with elimination of freeze-up damage, with rapid chilling of large fluid volume and close temperature control.

BULLETIN No. 104

HIGH PRESSURE STEAM HEATING



Bulletin describes method completely utilizing both the sensible and latent heat of high pressure steam. Trustworthy, long-lived equipment for heating or process.

BULLETIN No. 109

REFRIGERANT CONDENSERS



Bulletin describes evaporative condensers for ammonia and freon refrigerants offering high condensing capacity at a low cost per ton in power cost and upkeep as well as low installation cost.

BULLETIN No. 137

MOTOR BLOWERS



Bulletin describes blowers in two series up to 3½" static pressure. Available in one, two and three fan assemblies. Applicable to ventilating, drying, air conditioning.

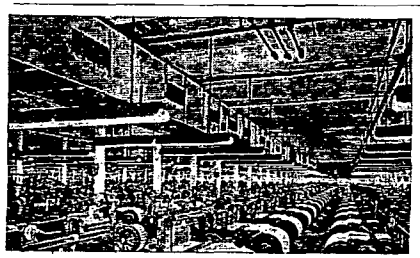
BULLETIN No. 89

For complete information, copies of Bulletins, Engineering Data, address NIAGARA BLOWER COMPANY, 405 Lexington Ave., New York 17, N. Y. Niagara District Engineers are located in principal cities.

PARKS-CRAMER COMPANY

Fitchburg, Mass. **Charlotte, N. C.** **Atlanta, Ga.**

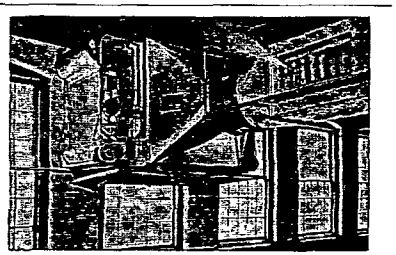
Industrial Humidifying and Air Conditioning Since 1904



AIR WASHER CERTIFIED CLIMATE

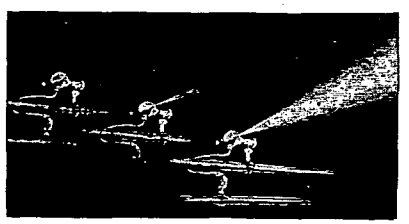
A complete system for conditioning the factory atmosphere, with positive air circulation and controlled ventilation. One or more air washer units. High humidifying and evaporative cooling capacity. Heating, filtering and refrigerated cooling where needed. Ducts with adjustable outlets distribute conditioned air uniformly. Slight air pressure also improves uniformity. Centralized maintenance. Used with or without booster humidification. Under gradual acting automatic control. Multi-speed control of fan improves economy.

Central Station Air Washer Units. Nozzles and Self-Cleaning Tank Screens for Central Station Air Washers.



AUTOMATIC AIRCHANGER

Improved system of forced air change and distribution used with direct humidification. Insures fixed uniform humidity and and maximum evaporative cooling. Amount of air change, and operation of humidifiers under gradual automatic control. For complete new installations or for supplementing existing direct humidifying equipment. Includes heating, refrigeration, filtering, where needed. With or without ducts.



DIRECT HUMIDIFICATION

Atomizer humidifier for all uses. Evaporative capacities up to 16 pounds of water per hour. Turbomatic—gravity type, on-and-off, fixed output, several sizes. GraduVac—full range modulating control, applicable to any gravity type atomizer system. Gradumatic—water under pressure, full range modulation. All types—self-cleaning, used alone or with Airchangers or as boosters for air washer systems.



LABORATORY AIR CONDITIONING

An essential of good testing is adequate and dependable air conditioning. Accuracy in testing is maintained only when samples and room atmosphere are free from fluctuations in both temperature and humidity. A vertical or horizontal unit is located within or adjacent to the laboratory. Includes equipment for heating, humidifying, dehumidifying, refrigeration, circulation, filtering. Adjustable recorder-controller. Heat by steam or electricity.



CERTIFIED CLIMATE REGULATION

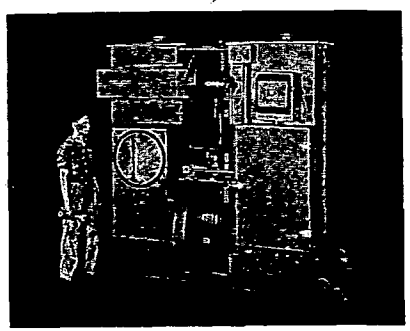
For humidity and temperature. Gradual acting, or on-and-off. Psychrostat—wet and dry bulb. Hygrostat—absorption type. Both types sensitive, rugged, reliable. Autographic recording included where desired.



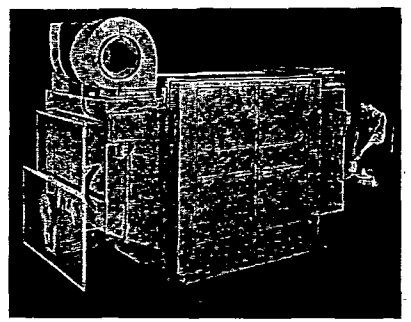
PITTSBURGH LECTRODRYER DIVISION

McGRAW-EDISON COMPANY

P. O. BOX 1766 **PITTSBURGH 30, PA.**



This specially designed LECTRODRYER protects equipment in storage by maintaining a relative humidity of 35 percent or lower



Air-conditioning (CH-type) LECTRODRYERS are used for automatic lowering of relative humidities in many types of industrial applications

INDEPENDENTLY CONTROLLED DEHUMIDIFICATION FOR INDUSTRIAL APPLICATIONS AND AIR CONDITIONING

Air-conditioning (CH-type) Lectrodryers provide an efficient, economical means of dehumidification in industrial areas in which the material in process or in storage requires a relatively dry atmosphere. Lectrodryers are used for this purpose in pharmaceutical houses, food processing plants, metal storage areas, and many types of packaging plants. The uninterrupted production, higher quality and safer storage made possible by the introduction of dry air in such applications often save the user many times the cost of the Lectrodryer. In addition, operating costs are negligible.

The dehumidification process is continuous and completely automatic. It is completely independent of any air-conditioning apparatus that might serve the same general area. If cooling is desired along with dehumidification—either for comfort air conditioning or in conjunction with an industrial application—an after-cooler is mounted directly on the Lectrodryer.

Wide Range of Sizes and Capacities

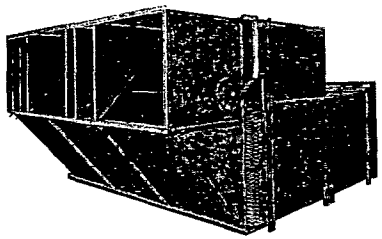
Relative humidities of 35 percent or lower are obtained with standard CH-type Lectrodryers. Capacities range from 350 cfm to as high as 15,000 cfm. Moisture is removed from the air by means of a solid adsorbent—activated alumina—with automatic reactivation being accomplished by gas, steam or electricity.

Lectrodryers require almost no maintenance. The only moving parts are the motor and blowers, and the valve drive mechanism. Units are shipped completely assembled, ready for easy installation. All are manufactured to rigid standards that have made Lectrodryer one of the world's foremost manufacturers of adsorbent dryers.

Write for full details.

RECOLD CORPORATION

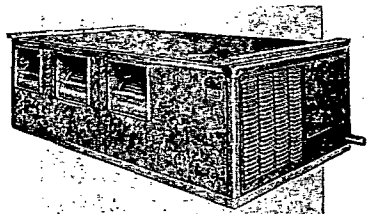
7250 E. Slauson Ave., Los Angeles 22, Calif.



MULTIZONE AIR CONDITIONING UNITS

Capacities from 840 through 36,000 CFM. Intermediate coils provide 27 sizes in 9 basic cabinets. Cabinets may be horizontal or vertical. Zones may discharge horizontally and/or vertically. One unit simultaneously heats and cools up to 28 zones. Units are available with backward curved wheels for static pressures up to 6 in. Units are also available less zone dampers for double duct high or low velocity systems.

Catalog 55C7c



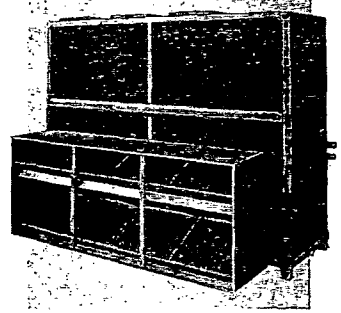
Horizontal Air Conditioning Unit

AIR CONDITIONING UNITS

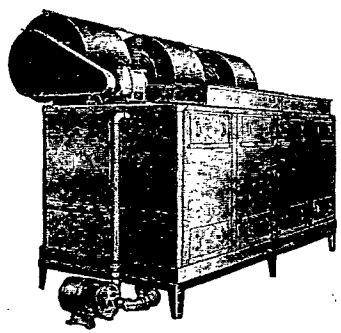
Horizontal and Vertical Air Conditioning Units. Capacities from 735 through 36,000 CFM. Intermediate coils provide 30 sizes in 10 basic cabinets. Cabinets may be horizontal or vertical. Units are available with backward curved wheels for static pressures up to 6 in.

Both Horizontal and Vertical units are available with the following accessories: *Insulation is ½ in. or 1 in. fiberglass* attached to panels with special mastic adhesive. Flat filter section for cleanable filters, removable from side, top or bottom on Horizontal units, or sides on Vertical units. Angular filter section for disposable filters, removed from sides. Face dampers for coils. By-pass dampers. By-pass section. Spray humidifier. Mixing box and dampers.

Catalog 54C7c



Vertical Air Conditioning Unit



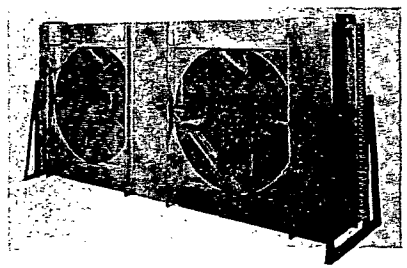
DRI-FAN EVAPORATIVE CONDENSERS & COOLING TOWERS

Capacities from 3 through 233 tons. Fan scroll assembly handles only dry air. Patented features include: clog-proof "bleed funnel," free-sliding "access doors," and "corner construction" that provides complete accessibility. Dri-Fan Evaporative Condensers are available with multiple circuiting and may also be used as jacket water coolers. Complete water cooler selection data appears in the catalogs listed below.

Dri-Fan Evaporative Condensers Catalog 30C8c
Dri-Fan Evaporative Cooling Tower Catalog 32C7a

RECOLD CORPORATION

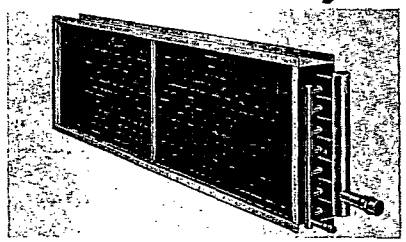
7250 E. Slauson Ave., Los Angeles 22, Calif.



DRICON AIR COOLED CONDENSERS

Capacities from 5 through 40 tons in seven cabinet sizes. Large, slow speed, belt-driven fans. Adjustable angle iron stand. Balanced circuiting. Multiple circuiting is also available. Complete water cooler selection data appears in the Dricon Air Cooled Condenser Catalog.

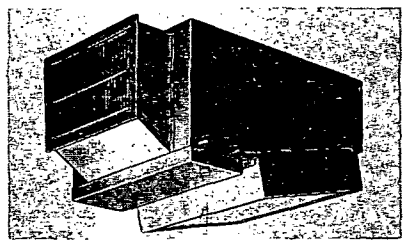
Catalog 34C8a



HEATING AND COOLING COILS

Each coil is especially circuited to insure a completely active coil for each individual application. Types of coils include: steam, water, direct expansion, non-freeze steam, heavy duty, low capacity, cleanable tube water coils and copper tube aluminum fin or copper tube copper fin coils.

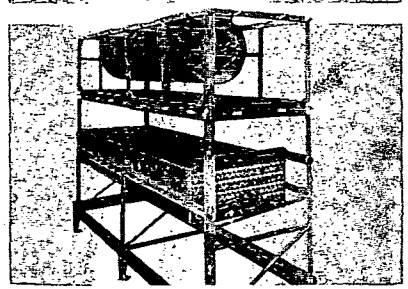
Steam Catalog 51C7a
Direct Expansion Catalog 52C8a
Water Catalog 53C8a



RE UNITS

½, ¾, 1 ton capacities, silent, compact, individual room units. *Units are available circuited for water or direct expansion* refrigerant. Standard equipment on Recold RE units includes: vertical bar adjustable grill; grill extension; filter box and filter; two speed switch and wall plate. Coils and drain pan are reversible for either right or left hand connections.

Catalog 56C8a



PATENTED "CORNER CONSTRUCTION" FOR COMPLETE ACCESSIBILITY

Recold's exclusive "corner construction" is used on all evaporative condensers, cooling towers, air conditioners and product coolers.

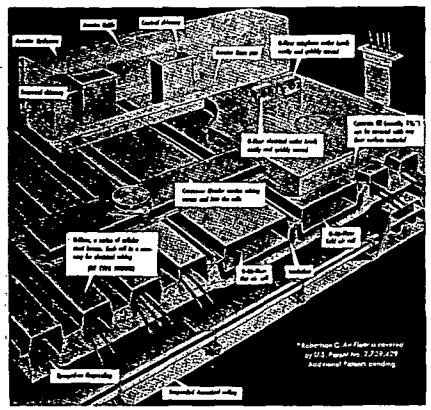
ELECTRICALLY WELDED ANGLE-IRON FRAME

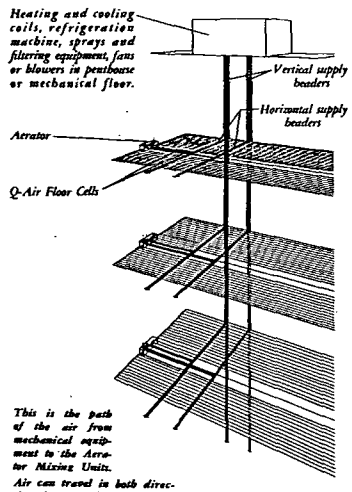
All concentration of weights, such as motor mounts, lifting channels, bearing, legs and coils are connected directly to the frame.

H. H. ROBERTSON COMPANY

2400 Farmers Bank Building , Pittsburgh 22, Pennsylvania

ROBERTSON Q-AIR FLOOR SYSTEM





This is the path of the air from mechanical equipment to the Aerator Mixing Units.
Air can travel in both directions in Air Cells. One direction is shown here for simplicity.

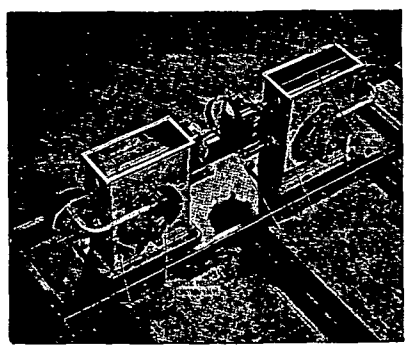
Robertson Q-Air Floor is cellular steel sub-floor construction that incorporates within its thickness structural, load-bearing cells that are available as electrical raceways, plus larger structural cells for use in connection with high velocity, dual duct air conditioning. (See cutaway illustration at left.) These special, load-bearing cells can be adapted by the air-conditioning contractor to transport air from horizontal supply ducts to mixing units for discharge into the room.

In Robertson Q-Air Floor, use of the floor cells for air passages reduces the space between ceiling and floor above. Overall building height can be reduced (without sacrificing ceiling height) by running the secondary supply headers through the web of the horizontal beams instead of below them. Great savings in power consumption, too, are a result because of the reduced cubage of the building. Except for peak summer cooling periods, Robertson's SEASONAL CHANGEOVER feature keeps the power requirements at about 65 percent of top consumption.

As with conventional dual duct systems, Q-Air Floor requires a central plant and a return air system.

Except for the change in width of occasional cells, Robertson Q-Air Floor is similar to standard Q-Floor—a product having the experience of 27 years and more than 15,000 installations behind it. Cell depth is the same and the 12-in. unit width of the large cell fits into the 6-in. Q-Floor module. Employment of cells to carry air is simply another use added to the widely accepted Q-Floor principle of using cellular steel units for load-bearing and electrical distribution.

The unique Robertson Aerator—shown below—is an ingenious piece of apparatus. In addition to its normal function as a mixing box, in which air from the hot and cold sides of the system is mixed in response to the setting of a thermostat, it also makes possible the conversion of the hot side for supplemental cooling use during the hot months of the year. This "seasonal changeover" device is the heart of Robertson Q-Air Floor and no other system has it.*



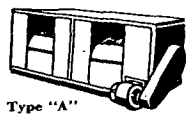
M. BLAZER & SON

PASSAIC NEW JERSEY

"WEATHER" HOT or COLD Think of Us

Manufacturers of Air Conditioning, Refrigeration, Heating and Ventilating Equipment

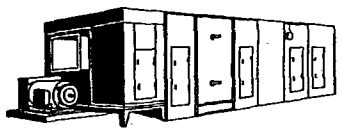
Air Handling Units



Type "A"

Custom fabricated air handling units for heating, ventilation and air conditioning. Available as illustrated in types "A" or "B".

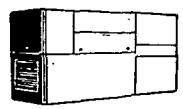
Type "A". Made in horizontal or vertical units in 39 cabinet sizes, in 1, 2 or 3 fan assemblies, handling from 400 to 50,460 cfm at pressures up to 3 in. WG.



Type "B"

Type "B". High pressure type. Up to 9 in. WG sectionalized, factory assembled air handling units, in dry coil or sprayed coil type with double insulated casings, access plenums, filter plenums, mixing and return air plenums, etc. In capacities to 28,160 cfm at 9 in. static.

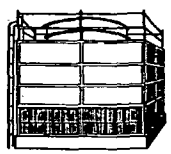
All units are available with steam or water heating coils, freon or water cooling coils and various filter media and arrangements. Humidifier, damper and recirculating spray assemblies, various insulations, motors and durable double casing construction. Made in Multizone and Hot and Cold Deck blowthrough types. Also specially designed to fit space limitations.



Self Contained Unit

Self Contained Air Conditioning Units

Complete with refrigeration compressors, built-in evaporative condensers or water cooled condensers for use with remote towers or city water. Also available with receivers only for use with remote air cooled condensers. All units are completely piped, wired and shop-run before shipment. Available with any or all modifications listed under other equipment shown here. Can be ordered also in roof mounted units for outdoor duty with air cooled condensers and special insulation. In sizes from 5 to 100 tons.



Cooling Towers

Cooling Towers

All metal inside and outside and completely fireproof. In sizes from 10 tons up to unlimited thousands. Very good for cooling water with air through any range to a 7 deg approach to wet bulb. Available in transite paneled casings, galvanized steel or stainless steel. Entire casing is paneled for easy access into interior. Designed to require minimum maintenance. Requires the least floor area of any large tower with water loadings up to 7 gpm per square foot floor space. Evaporative condensers and coolers in similar construction are available for any type of cooling or condensing application.



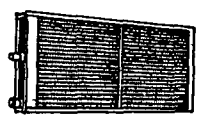
Mechanical Dehumidifiers

Mechanical Dehumidifiers

Utilize Freon compressors. In sizes from 5 to 60 tons, air handling from 640 to 11,850 cfm and moisture removal to 1,600,000 grains of moisture per hour at dewpoints to 36 deg. Greater capacities down to −25 deg dewpoint by spraying ethylene glycol over cooling coils to prevent frost. Double evaporator type with automatic defrosting also available. Completely wired, piped and shop-run.

Heating and Cooling Coils

Helically finned type in copper, aluminum, carbon or stainless steel for all kinds of installations. Heating coils may be obtained for steam in standard and non-freeze types and water in regular or high temperature, high pressure type. Cooling coils for Freon, ammonia or brine in direct expansion, flooded, and standard or cleanable types for water. Available with from 1 to 36 tubes high, 1 to 30 rows deep, 6 in. to 180 in. finned length. Made in steel or non-ferrous casings. All coils tested to 300 lb pressure under water.



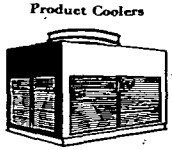
Coils



Product Coolers

Product Cooling and Freezing Units

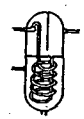
Blower capacities from 2,000 to 30,000 cfm. Btu ratings to 225,000 at 10 deg TD. Available for above and below freezing duty. Patented defroster or glycol spray permits use to −40 F.



Air Cooled Condensers

Air Cooled Condensers

Large heavy duty commerical and industrial air cooled condensers in sizes up to 300 ton. Unique penthouse type design gains high capacity from limited space. Available with specially circuited coils and discharge dampers for head pressure control.

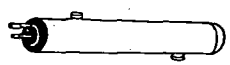


Slug Eliminator

Patented Refrigeration Specialties

Balance Loaders for obtaining compressor capacity control in infinite steps from 0 to 100 percent.

Slug Eliminators for protecting compressors from breakage due to liquid slugging. In sizes to 100 ton.



Balance Loader

WALTON LABORATORIES, INC.

Irvington 11, New Jersey

Walton

One of the world manufacturers of the largest diversified line of humidifiers

ALL MODELS TROUBLE-FREE WITH THE WALTON CENTRIFUGAL *ATOMIZER*



For over 25 years this well balanced, whisper quiet atomizer has given positive humidification without pressure devices, nozzles, or other complicated auxiliary equipment.



PORTABLE ROOM HUMIDIFIERS

are available in both table and automatically controlled console models with attractive wood-grained finishes. There are no heating coils or filters; no wet packs to become odorous or clogged. No fans that create annoying air movements in confined areas.

SPACE TYPE HUMIDIFIERS

for commercial and industrial installations requiring controlled moisture conditions for manufacturing operations or storage of products. Need only small water supply line and electric connection through a control circuit. No costly ducts, compressors or drains are required.





DUCT TYPE HUMIDIFIERS

for commercial and residential installations. The right way to add needed moisture to any warm air-circulating system. Use water supply line and electric connection. Drain unnecessary. Produce vapor instantly without adding heat.



WALTON'S CONSULTANTS' FILE

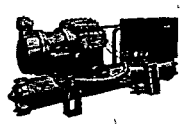
offers you literature on all types of humidifiers with their many commercial, industrial and residential applications. Tell us your needs and covering technical information will be forwarded to you.



WORTHINGTON CORPORATION

Air Conditioning and Refrigeration Division

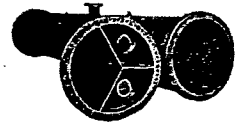
Section 12-14, Ampere Station, East Orange, N. J.



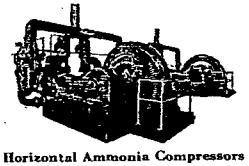
Hermetic Condensing Units



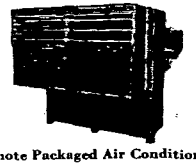
Packaged Air Conditioners



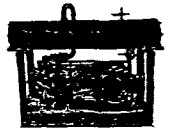
Condensers and Brine Coolers



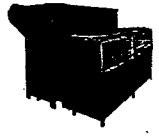
Horizontal Ammonia Compressors



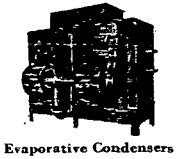
Remote Packaged Air Conditioners



Refrigerant 12 and 22 Condensing Units



Central Station Air Handling Units



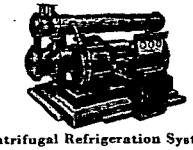
Evaporative Condensers



Packaged Liquid Chillers



Vertical Ammonia Compressors



Centrifugal Refrigeration Systems



Hermetic Centrifugal Refrigeration Systems

In air conditioning . . . in refrigeration

Worthington line means *complete* line

Whatever the application . . . industry . . . business . . . Worthington offers you the trusted experience of over 70 years of leadership in the air conditioning and refrigeration field. In new equipment, design, or the unmatched experience of Worthington engineering and application staff, you are sure of precise recommendations tailored to your exact system needs.

Further, a nation-wide organization of distributors stands ready to provide you with total sales and engineering service. Worthington's experienced staffs will plan for single unit or central system installations, too. Architects, engineers and contractors are cordially invited to consult with us on *any* air conditioning or refrigeration problem. Offices or representatives in principal cities. Or, if you prefer, write to Worthington Corporation, Air Conditioning and Refrigeration Division, Section 12-14, Ampere Station, East Orange, N. J. In Canada, Worthington (Canada) Ltd., Brantford, Ontario.

WORTHINGTON

YORK CORPORATION

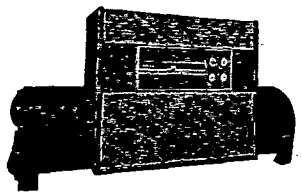
York, Pennsylvania

Factory Branches and Distributor Engineering and Sales Offices throughout the World
Subsidiary of Borg-Warner Corporation

Air Conditioning and Refrigeration for maintaining proper temperature and/or humidity for industrial processes or comfort requirements. Installations of unit and central systems in a wide range of capacities and types for every design need.

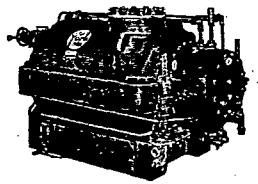
WATER COOLING SYSTEMS

YORK HERMETIC TURBOPAK



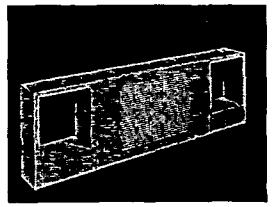
A packaged single stage hermetic centrifugal water cooling system for every air conditioning and process cooling purpose. Capacities range from 67 tons to 550 tons. Completely factory assembled and insulated with all controls factory mounted.

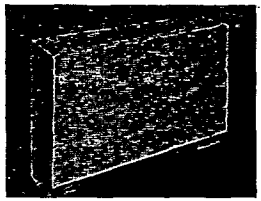
MULTI-STAGE TURBOS



Multi-stage centrifugal compressors for large comfort or industrial air-conditioning and refrigeration applications. Available over a wide range of capacities to 5,000 tons in a single unit. Suitable for motor or steam turbine drive. Pre-rotation vanes offer peak performance at peak or partial loads.

AIR CONDITIONING SYSTEMS





YORK INDUCTION SYSTEMS

Utilizes York Hermetic Turbopak or Packaged Water Chiller as a cooling source. Air, brought into building at one intake by fan, is filtered, dehumidified and distributed in high pressure ducts throughout the building to individual units. Units in individual rooms also provide heat in winter by hooking into existing hot water system. Normal and high capacity units both available.

YORK FAN COIL SYSTEM

Cools in summer, heats in winter, uses Turbo or V/W compressor for cooling, building's hot water system for heating. Economical for buildings where space is at a premium. No large fan room required; units individually operated. Floor, wall and ceiling models available.

HEAT PUMPS

Developed by York, compound compression air source heat pumps without supplementary heaters are economical and practical even in cold climates. Providing all-weather heating and cooling as needed for every air conditioning application, the heat pump eliminates the need for boilers, fuel storage and stacks.

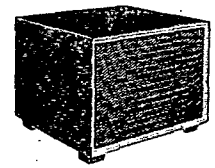
ADDISON PRODUCTS COMPANY

ADDISON, MICHIGAN

Contract Manufacturers

Central Air Conditioning—Self-Contained or Split Systems • Room Coolers • Dehumidifiers • Also Milk and Beverage Coolers—Other Refrigeration Systems to Your Specifications

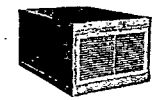
CENTRAL AIR CONDITIONERS—AIR COOLED SPLIT SYSTEMS



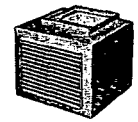
3 hp Condensing Unit

AIR COOLED CONDENSING UNIT—Sizes from 24,000 Btu/hour to 80,000 Btu/hour, single or 3 phase. Horizontal air flow. Completely weather-proofed for outdoor installation. *Underwriters'* approved. Equipped with oversized condensers for efficient, uninterrupted performance at high ambient temperatures. Compact design uses minimum space. Quiet operation—low velocity air flow through large openings. Completely accessible. Attractive appearance. Furnished for capillary feed or expansion valve applications. Optional refrigerant connections—including soldered, flared or improved quick connect couplings for use with refrigerant charged lines.

EVAPORATORS TO MATCH CONDENSING UNITS



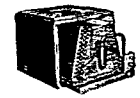
"EB" Evaporator Blower Unit



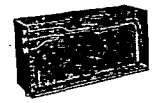
"UB" Universal Blower Unit

"EB" EVAPORATOR BLOWER—Quiet blower and smart appearance allow installation anywhere. Unit has adjustable four-way louvers. Return grille equipped with air filter. Fully accessible through removable panels.

"UB" UNIVERSAL BLOWER—Use with existing warm air system or separate ductwork. Blower assembly easily relocated to alternate positions—choice of vertical upward, horizontal or vertical downward discharge air connections. Fully accessible.



"A" Coils, Vertical Air Flow

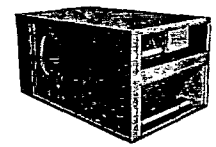


"H" Coils, Horizontal Air Flow

"A" COILS—Designed for vertical air flow plenum installation. Uses the ductwork and blower of the heating system. Minimum resistance to air flow. Fully insulated. Coils slide into housing through removable front panel.

"H" COILS—Designed for horizontal air flow using ductwork and blower of the heating system. Install in plenum chamber or supply ductwork. Minimum air flow resistance. Insulated condensate drain pan.

SELF-CONTAINED, CENTRAL AIR CONDITIONERS

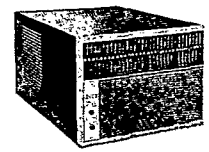


4 hp Model

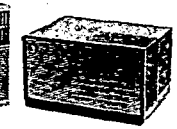
Available in 2 through 4 hp sizes. Designed and *Underwriters'* approved for outdoor installation. Equipped as factory assembled to meet *FHA* requirements. Compact designs allow installation through wall or in crawl space, attic or other limited access locations. Use with heating ductwork or as separate system. Capillary tube refrigerant feed system. Extra large centrifugal blower delivers volumes of low velocity air over maximum surface coils for smooth, silent operation.

Water Cooled systems also available.

WINDOW AIR CONDITIONERS (Approved by the Underwriters' Laboratories)



2 hp Unit



½ hp Portable

Models from ½ to 2 hp. Compact design—quiet operation. Automatic temperature control. Positive exhaust and ventilation. Designed for high temperature operation. Capillary tube refrigerant feed system. Easy access to all components. Variety of grilles and trim.

SPECIAL SYSTEMS FOR MOBILE HOMES, DEHUMIDIFIERS, BEVERAGE AND MILK COOLERS.

CHRYSLER AIRTEMP

AIR CONDITIONING FOR EVERY NEED

Dayton 1, Ohio **Engineered by Chrysler**



ROOM AIR CONDITIONER

Easily installed in homes, offices or buildings. *Chassis slides out if service* is required. Thermostatically controlled, cools, filters, circulates, ventilates, exhausts and dehumidifies air. Capacities in 1, 1½, 2 and 2½ hp. Models designed for 7½ and 12 amp, 115 and 230 volt operation, also for reverse cycle heating.

CASEMENT WINDOW CONDITIONER

Quickly installed from inside. No cutting or removing of glass; no outside projection. Window opens, closes, locks as before. Thermostatically controlled, cools, filters, circulates, ventilates, exhausts and dehumidifies air. Two-speed fan for super-quiet night cooling and exhaust features on Royal models. Capacities in ½, ¾, and 1-hp.



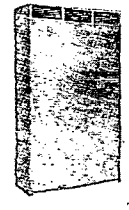


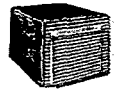
ROOM AIR CONDITIONERS

For *year-round cooling and heating* by connecting to central chilled and hot water systems. Designed for multi-room buildings, offices, and hotels. Three types—floor, wall, and ceiling. Built-in controls permit operation to suit tenant. Two quiet centrifugal fans provide air circulation.

PACKAGED AIR CONDITIONERS

Completely self-contained, assembled, factory-tested, easily installed. Waterless and water cooled in 2 to 30-hp capacities. These Airtemp air conditioners cool, dehumidify, filter, and circulate air. Free-air discharge or duct distribution. Heating coil for year-round air conditioning optional. Bonderized steel cabinet occupies minimum floor space. Famous Airtemp sealed compressor—quiet, all moving parts balanced and bathed in oil for long life. Flexible mounting reduces vibration. 5-year warranty on entire refrigerant circuit.



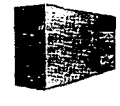


LOW-COST HOME AIR CONDITIONING

Compact waterless package installs easily in attic, dormer, or basement, either separately or connected to existing heating ducts, 2 and 3-hp capacities. Uses air and electricity only, no water piping needed.

CENTRAL DUCT PACKAGED AIR CONDITIONERS

Self-contained Central Air Conditioners in 25, 40, 60, 75, and 100-ton capacities. Complete with Airtemp radial compressor and its exclusive features. Available with *evaporative condenser*, or for use with cooling tower.



CHRYSLER AIRTEMP

Air Conditioning Systems and Equipment

Dayton 1, Ohio **Engineered by Chrysler**

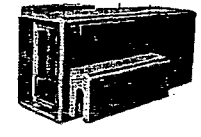


GAS AND OIL-FIRED AUTOMATIC FURNACES

Wide range to fit any application; Lo-Hi-Boy, Lo-Boy, and Counterflow automatic furnaces. Install in closet, attic or basement. Btu output from 65,000 to 250,000. Easy to add Airtemp cooling for year-round air conditioning.

HORIZONTAL GAS AND OIL FURNACES

For space-saving installation; in attic, under floor, or hung from ceiling Btu outputs from 64,000 to 168,000. Dual flue outlets. Draft diverter mounts on either side. Completely-automatic controls fully enclosed.



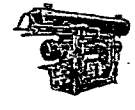


WATERLESS CONDENSING UNIT

Ideal for home or commercial use. Operates on air and electricity only—no water, no water piping, no cooling tower. Does not require winter shutdown. Can be installed outside. Cabinet is weatherproof. In 2, 3, 5 and 7½ hp sizes.

PACKAGED WATER CHILLERS

Capacity from 25 to 250-hp, using Freon 22; and 3 to 75-hp, using Freon 12. Each package is factory-assembled, piped, pressure-tested, and furnished with refrigerant holding charge.





RADIAL COMPRESSORS AND CONDENSING UNITS

10 to 125-hp capacities—heavy duty, for use with Freon. Specially designed for air conditioning or refrigeration. Compressors have direct drive and have forced-fed lubrication. Automatic capacity-reduction. Light weight, economical to operate.

CENTRIFUGAL WATER CHILLER

A complete refrigerating system with two-stage centrifugal compressor, evaporator, condenser, inter-connecting piping, purger, safety and operating controls. Wide application for summer air conditioning or process cooling. Capacities from 150 to 700 tons. Six compressor sizes, plus 64 evaporator and condenser combinations, meet any requirement.



AMERICAN GAS ASSOCIATION

ARKLA-SERVEL

. . . . . now we're cooling



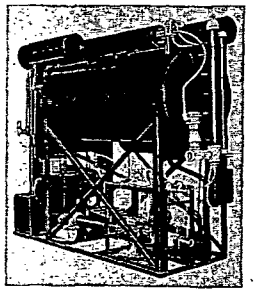


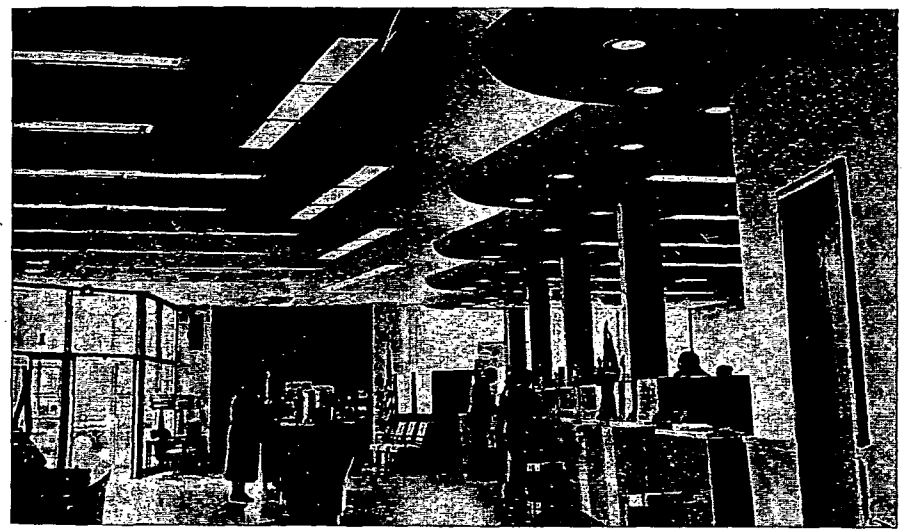
Specify Arkla-Servel Gas Air

With their new Arkla-Servel Gas Absorptive Cooler, the La Grange Federal Savings and Loan Association keeps customers cool in summer with the same compact system that keeps them warm in winter.

Before installing Gas, a complete study was made of available air conditioning systems. The Arkla-Servel unit—the only 25-ton absorptive cooler—was chosen because it is compact, easy to install, and costs are low for installation, operation and maintenance. No specially trained operating or maintenance personnel are required.

with GAS





Conditioning and you specify years of trouble-free comfort

Only Gas gives these important advantages:

- high efficiency at all times—even during the light loads
- temperature control is constant
- modular adjustment of capacity (instant automatic adjustment to match actual cooling requirements)
- dependability of fuel service at all times

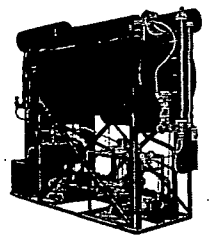
Gas absorptive cooling can put your commercial and industrial clients' heating plant on a year around paying basis. It utilizes low pressure steam to cool water, has no moving parts to wear out, and provides quiet, economical operation. What's more, it's vibration-free.

Take advantage of the consulting services provided by your Gas company. They have trained specialists who have been working with contractors and builders for years. They belong to your associations or affiliations and are familiar with your problems. Check the facts about Gas and you'll see modern Gas air conditioning out-performs all other fuels. *American Gas Association.*

ARKLA AIR CONDITIONING CORPORATION

Evansville 7, Indiana

GENERAL SALES OFFICE AND APPLICATIONS ENGINEERING DEPARTMENT
SHANNON BUILDING, LITTLE ROCK, ARKANSAS



ABSORPTION-TYPE 25 TON WATER CHILLER—SERIES 3000

Also available in multiples of 50, 75, and 100 tons.

THE ARKLA-SERVEL SERIES 3000 Water Chiller is a factory-sealed absorption-type refrigeration unit with no moving parts, providing an *economical* source of chilled water for wide range of comfort air conditioning and industrial process applications. It is readily adaptable to multiple installations, with basic component of 25 tons. Water is the refrigerant, lithium bromide, the absorbent, and steam, the source of energy. Steam for operation is normally obtained from a *direct-connected* gas boiler, but can come from practically any source. The unit utilizes any steam pressure from 6 to 15 psi. It operates under a vacuum, eliminating pressure hazard; the chilled water circuit, however, is under pump pressure and not part of the vacuum system. A refrigeration "package," the unit requires no *field assembly*, and is easily installed on any floor of a building. Vibration dampening is unnecessary. Automatic modulation of steam input is provided by factory-installed steam controls.

SERVICE CONNECTION SIZES

Steam Inlet	2" IPS
Steam Condensate Drain Connections:	
Generator Drain	1⅛" OD Copper
Diverter Valve	¾" IPS
Steam Inlet Drip Trap	½" IPS
Steam Vent at Diverter Valve	1½" IPS
Chilled Water (Male Connections)	2" IPS
Condensing Water (Female Connections)	2" IPS

RATED REFRIGERATION PERFORMANCE*

Tons Refrigeration	25.0
Total Capacity, Btu per hour	300,000
Steam Input:	
Btu per Hour, Max.	450,000
Pounds per Hour, Max.	464
Clean dry steam at any pressure above 6 lb is suitable, properly controlled.	
Chilled Water:	
Inlet Temperature, °F	50.0
Leaving Temperature, °F	40.0
Water Quantity, gpm	60.0
Pressure Drop, psi	7.6
Condensing Water:	
GPM at 75° Inlet	63.0
Pressure Drop, psi	13.9

* Refrigeration Capacity shown is based upon standard method of rating, specified by *American Society of Refrigerating Engineers;* Inlet condensing water at 75°F and chilled water entering at 50°F and leaving at 40°F.

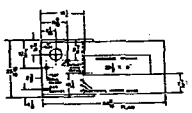
PHYSICAL DATA

Net Weight, lb	8,800
Shipping Weight, lb	9,225

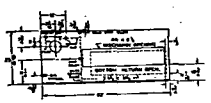
ARKLA-SERVEL SUN VALLEY*
GAS Air Conditioners

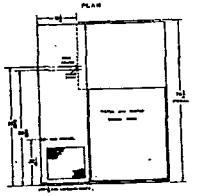
THE ARKLA-SERVEL SUN VALLEY* All-Year Air Conditioner is a gas-fired, water-cooled absorption unit which automatically cools, dehumidifies, cleans and circulates the air at a controlled temperature in summer, and cleans and circulates the air at a predetermined temperature in winter. Principal parts of this single-package year-round air conditioner are an absorption combination refrigeration and heating unit, a filter section and a centrifugal blower . . . In cooling cycle, a gas flame in the generator boils a water-lithium bromide solution, producing a water vapor which goes to the condenser. The condensed vapor flows into the single coil, over which air is blown. Heat is removed from the air and the vapor, through absorption, is returned to liquid and flows by gravity to the generator for re-use. In heating, the water vapor by-passes the condenser and goes direct to the coil still hot. Then heat is taken from the vapor and the water by gravity returns to the generator . . . Circulation of the water, acting as refrigerant, and the *lithium bromide*, as absorbent, is achieved by differences in temperature and height of liquid columns, without moving parts. The unit operates under a high vacuum . . . available in two basic models, the Model 500, 3½-ton and Model 750, 5-ton.

* Trademark



Model 500

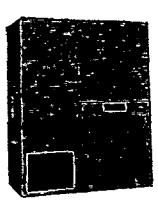




Model 750

Chilled Water Temperature °F.	75° Condensing Water (F) 450,000 Btu Steam Input (464#)			85° Condensing Water (F. °) 400,000 Btu Steam Input (412#)		
	Cooling Capacity Tons	Chilled Water		Cooling Capacity Tons	Chilled Water	
		G.P.M. at 10° T. M	Pressure Drop P.S.I.		G.P.M. at 10° T. M	Pressure Drop P.S.I.
35°	30.8	61.5	10.6	24.1	48.2	7.3
40°	29.6	59.2	10.0	24.1	48.2	7.3
45°	28.1	56.2	9.7	23.2	46.5	6.6
50°	27.0	54.0	9.4	22.3	44.6	6.1
55°	25.6	51.3	9.2	21.3	42.6	5.8
60°	24.1	48.2	8.8	19.8	39.5	5.4
65°	22.0	44.0	7.6	18.3	36.6	5.0

REFRIGERATION CAPACITY			
Gas Input, Full Capacity, Btuh	80,000	120,000	120,000
Delivered Capacity, Btuh (Rated Cooling Capacity Based on Inlet Water 75°F., Inlet Air 80 deg db and 67 deg wb., 1400 CFM)	42,000	60,000	60,000
HEATING CAPACITY			
Gas input, Full Capacity, Btuh	120,000	120,000	180,000
Bonnet Capacity, Btuh	96,000	96,000	144,000
WEIGHT:			
Net (Approx.)	953 lbs	1,153 lbs	1,250 lbs
Shipping (Approx.)	1,077 lbs	1,303 lbs	1,400 lbs



MODEL 500



MODEL 750

Model 500
¼ hp, *115 V, 60 cy*, 1 ph Motors shipped as standard.
¼ hp, 230 V, 60 cy, 1 ph Motors available at extra cost.
¼ hp, 115/230 V, 60 cy, 1 ph Motors available at extra cost.

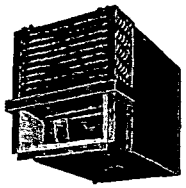
Model 750
¼ hp, 115/230 V, cy, 1 ph Motors shipped as standard.
¾ hp, 115/230 V, 60 cy, 1 ph Motors available at extra cost.

For additional information, write Arkla Air Conditioning Corporation, Shannon Building, Little Rock, Arkansas

KRITZER PRODUCTS

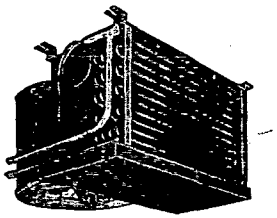
A Division of
PEERLESS OF AMERICA, INC.

MAIN FACTORY AND OFFICES
5800 N. Pulaski Road, Chicago, 46, Ill., U.S.A.



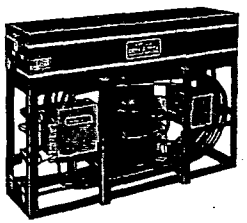
CBX COMBINEX® WEATHER BEAM

The Combinex Heating and Air Conditioning System (Patent Applied For) utilizes hot and chilled water and provides high-level discharge and low-level return for heating and high-level return and low-level discharge for air conditioning. Mounted in ceiling and discharges into plenum chamber or conditioned space. Has control lever for changing direction of air flow. Capacity: ¾ ton to 5 ton.



CXW CONCEAL-X (CEILING MOUNTED FAN COIL UNIT)

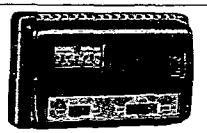
Adaptable for new or old construction. Uses hot and chilled water providing compact, high capacity forced air heating and cooling installations. Capacity: ¾ ton to 5 ton.



WCW CHILL-A-TRON® LIQUID CHILLERS

Completely packaged with water cooled condensers. Furnished with Kritzer patented Dimpled Tube exchanger surface. Easily installed. Complete hermetic unit with capacity range 2 ton to 22½ tons. Two to 5 ton units furnished for single or 3 phase, 220 v current; units 7½ to 22½ tons furnished for 3 phase, 220 v current. Larger models to 350 tons also available.

WRITE FOR LITERATURE containing complete specifications.



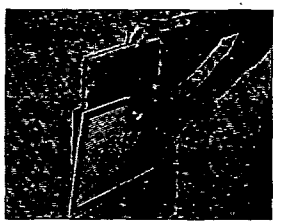
Kritzer Low Voltage Control

The Kritzer Unit control is completely factory wired and contains thermostat with thermometer, 3-speed fan control, heating and cooling cross-over switch and thermostat adjustment for heating and cooling.



Kritzer Live Air Grill

Beautifully designed. Has broad flat hammered glass vanes which provide for full spread of discharge air. Three sizes: 14 in. x 6 in., 22 in. x 6 in. and 30 in. x 6 in.



Kritzer Electrostatic Filter and Grill

Assembly furnished in sizes to suit systems. Grill is hinged and spring loaded and permits quick access to filter for removal and cleaning.

DEHUMID-I-TRON

A *direct expansion refrigeration type* dehumidifier with reheat. Capacities from ½ hp to 5 hp. Available in portable models or for permanent installations.

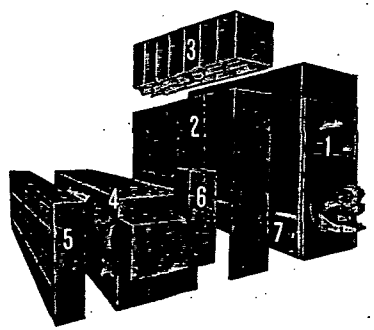
Direct expansion, water and condenser coils; cooling towers; and air-cooled condensers available in many models and sizes.

AMERICAN FOUNDRY and FURNACE COMPANY

Bloomington, Illinois

THE COMPLETE LINE IN WARM AIR HEATING—
ALL CAST IRON CONSTRUCTION FURNACES

The New *June-Aire*® Optional Unit Construction. . . . The system that does not rely on excess outside air for temperature control . . thus is inherently more economical



"June-Aire" assemblies offer heating capacities from 222,000 to 3,420,000 btu per hour output and standard air flow from 2600 to 45,300 cfm.

SYSTEM DESCRIPTION

A completely pre-engineered unit combining heating—ventilating—humidifying, providing for convenient addition of summer cooling and dehumidifying. Made up of the following components:

1. "June-Aire" Furnace Unit designed to accommodate all burner models.
2. By-pass unit with clearances for cooling coil.
3. Mixing damper unit.
4. Blower Unit.
5. Filter unit—Permanent or throw-away filters.
6. Control panel, pre-wired.
7. Humidifier.

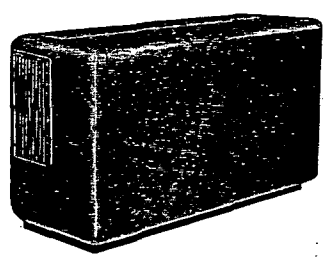
***June-Aire*®**

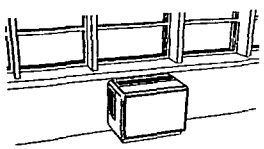
AIR DISTRIBUTION CONTROL CENTER

assures accurate, automatic room comfort

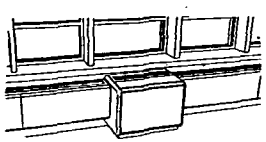
Designed for dual duct systems providing heating, ventilating and air conditioning. Unit contains both balancing and mixing dampers—suitable for any type of temperature control system.

- DRAFTLESS AIR DISTRIBUTION.
- QUIET OPERATION—No noisy fans or motors.
- FINGERTIP AIR VOLUME ADJUSTMENT.
- READY ACCESS TO DAMPERS AND OPERATOR.
- MINIMUM MAINTENANCE—Blowers, motors, filters, fresh air intake and humidifier all at central location.

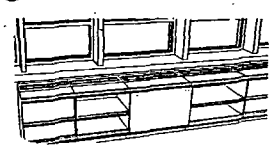




Individual application of Air Distribution Control Center with side grills.



Air Distribution Control Center used with wing ducts.



Air Distribution Control Center used with air handling wall bookcases.

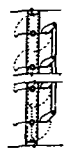
FOR COMPLETE INFORMATION CONSULT SWEETS' ARCHITECTURAL FILE

AMERICAN FOUNDRY AND FURNACE COMPANY
AIR HANDLING PRODUCTS

CONSTRUCTION DETAILS
Basic Designs

DAMPERS

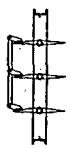
VOLUME CONTROL



Parallel Blade F12

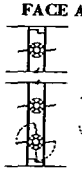


Opposed Blade F18

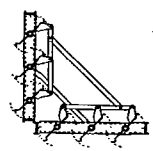


Airfoil Blade F20

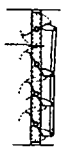
FACE AND BY PASS MIXING



ROTO-LINK



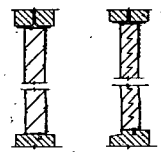
Right Angle Ducts F16



Vertical Ducts F19V

LOUVERS

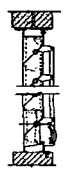
STATIONARY



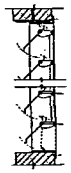
Weatherproof F8A

Stormproof F9A

ADJUSTABLE



Weatherproof WLD



Automatic FLHV

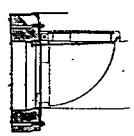
EXTRUDED

Comparable designs are available using extruded aluminum sections.

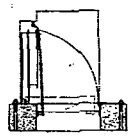
A choice of frame and blade designs permits adapting extruded louvers to most air handling requirements.

FIRE DAMPERS

CLASS A

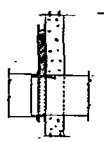


Off Center Pivot



Trap Door

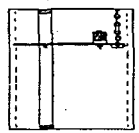
CLASS B



Guillotine A or B



Multiblade FS



Swing Type ST

ROTO-LINK



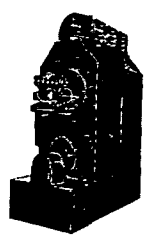
FOR ANY DAMPER REQUIREMENT ALL LINKAGE INSIDE DUCT OUTSIDE AIR STREAM RECESSED MECHANISM PREVENTS DAMAGE OPERATING TORQUE LESS THAN 1 INCH POUND PER SQUARE FOOT OF DAMPER AREA

SPECIAL PROCESS

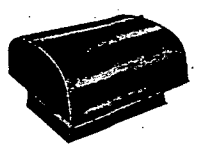
DAMPERS WHICH NECESSITATE A SINGLE PURPOSE DESIGN APPLICATION ENGINEERED TO MEET YOUR SPECIFIC NEED FOR

- INDUSTRIAL PROCESS CONTROL
- TUNNEL VENTILATION
- HIGH PRESSURE FAN OUTLET
- DUST COLLECTING SYSTEMS
- DRAFT CONTROL

DESIGNED TO WITHSTAND UP TO 80 in. W. G. AIR PRESSURE.



DYNE-AIRE Space Heaters



AFFCO Roof Hoods

SPECIFICATIONS
Standard

DAMPER SPECIFICATIONS

Frames—2 x ½ x ⅛ in. or 3 x 1 in. x 14 ga Steel Channel. Maximum dimensions single section 48 in. wide x 96 in. high.

Blades—#16 gauge steel dieformed. Width—Maximum 12 in. Minimum 4 in. Length—Maximum 48 in. Minimum 4 in.

Shafts—½ in. diameter steel rust-proofed.

Bearings—Oilite bronze Self-lubricating.

Finish—Commercial steel painted aluminum or Galvanized steel not painted.

Built to any dimension. Also available any material and any finish.

LOUVER SPECIFICATIONS

Constructed of commercial galvanized steel

Frames

Stationary—4 in. x ½ in. x 20 ga channel.

Adjustable—4 in. x 1 in. x 14 ga box channel or 4 in. x 1½ in. x 14 ga angle.

Blades

Stationary—#20 ga. Dieformed with weatherlip both edges.

Adjustable—#16 ga. Dieformed with weatherlip both edges.

Built to any dimension. Also available any material and any finish.

FIRE DAMPER SPECIFICATIONS

All meet requirements of *NBFU* as suggested in Pamphlet 90A.

Class A dampers are approved type for ducts passing through fire walls or fire partitions. Class B are acceptable for installation in ducts at other than fire-wall openings.

Protecting links are *UL* approved. Built to fit opening size specified.

FOR COMPLETE INFORMATION CONSULT SWEETS' INDUSTRIAL CONSTRUCTION FILE

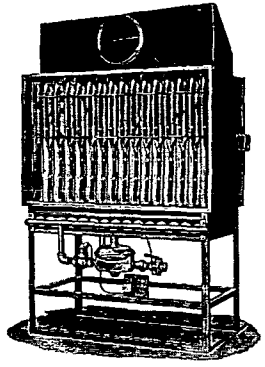
Hayes Furnace Mfg. & Supply Co.

3233 South LaCienega Blvd., Los Angeles 16, Calif.

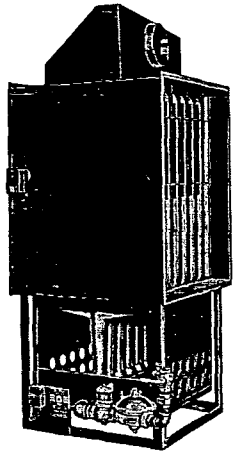
(Telephone Texas 0-3734)
Contact Nearest Representative

M. Blazer & Son, Passaic, N. J.
R. E. Holcombe, Atlanta, Ga.
Sinclair Heating Supplies, Ltd., Edmonton, Alberta, Canada
Boyd Engineering Co., Albuquerque, New Mex., and El Paso, Texas
R. J. Clark Equip. Co., Clearwater, Fla.
Dealers Supply Co., Atlanta, Ga.
Reube O. Emery, Nashville, Tenn.
F & D Distributors, Salt Lake City, Utah
Frederick Co., Spokane, Wash.
General Engineering Co., Omaha, Nebr.
J. C. Lewis Co., Little Rock, Arkansas
Luedecke Engineering Co., Austin, Texas
McCombs Supply Co., Denver, Colo.
McDonald Mfg. Co., Dubuque, Iowa
Mechanical Equipment Co., Boise, Idaho
Noland & Co., Newport News, Va.
Nunn Electric Co., Lubbock, Texas
O'Connor-Oklahoma Co., Oklahoma City, Tulsa, Okla., and Wichita, Kansas
Smith Steam Specialty Co., Kansas City, Mo.
Central Service Co., Inc., San Jose, Calif.
Temp Control Corp., Portland, Oregon
Valesco Inc., Visalia, Calif.

DUCT FURNACES



Heat exchanger constructed of Type 321 Stabilized Stainless Steel. Non-corrosive property of heat exchanger permits installation downstream of cooling coils, where codes permit, or washer. Heat exchanger will not corrode when subjected to continuous ventilation during the summer season. Designed for continuous blower operation for offices, theaters, schools, churches, and factories. Air throughput in either direction. Draft Hood and Vent Manifold reversible. 15 sizes, 45,000 to 675,000 Btu per hr input in 45,000 Btu increments. *A.G.A.* certified for all gases.



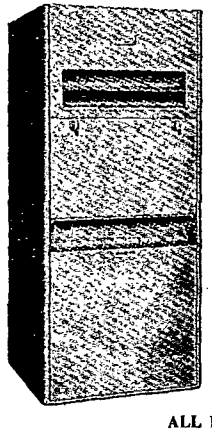
MODEL SED-E

Non-corrosive type 321 stainless steel heat exchanger. Permits installation downstream of cooling equipment, where codes permit, for continuous blower operation. Air through-put in either direction. 45,000 to 675,000 Btu input.

MODEL SED-S

Same as SED-E except controls are at the side, instead of at the end where accessibility requires. SED-S is available in sizes from 45,000 to 675,000 Btu input in increments of 45,000 Btu.

Both SED-E and SED-S can be used for industrial applications for drying.



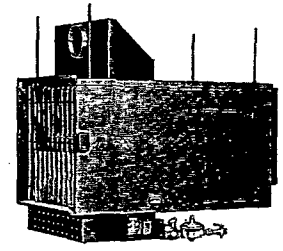
FORCED AIR FURNACE MODEL SEC

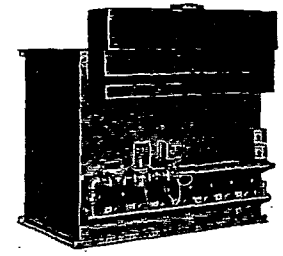
Heat exchanger constructed of identical die formed sections of type 321 stabilized stainless steel. Sections are seam and arc welded. One piece cast iron burners with drilled ports. Oversize double inlet blower. Long hour motor. Equipped with glass filters. Can furnish permanent filters at extra cost. Finished in baked silver gray. 6 sizes 70,000 to 245,000 Btu per hr input in 35,000 Btu increments. For natural, manufactured and L.P. gases. Can be used for 0 clearance.

ALL HAYES FURNACES ARE *A.G.A.* APPROVED

DUCT FURNACE MODEL SED-VF

Tested and approved as a DUCT FURNACE—Constructed of type 321 stabilized stainless steel. Equipped with air bypass. Handles large cfm per Btu input. Vent in front of furnace. Made in 6 sizes from 70,000 to 280,000 Btu per hr input in increments of 35,000 Btu. For natural, manufactured and L.P. gases.





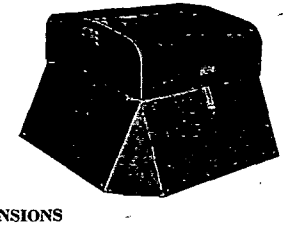
FORCED AIR SUSPENDED FURNACE, MODEL SES

Equipped to install in a suspended duct system. Inlet and outlet designed for sheet metal duct connection.

UNIT HEATER MODEL SEU

Used where room air is to be recirculated. Diffuser outlet with adjustable vertical and horizontal vanes. Both types of Suspended units are designed for factories, commercial and other large Btu requirements — equipped with brackets for suspension, save floor space. Double inlet forward curve blower. Continuous duty variable pitch drives; cast iron, raised port, precision machined burners. Made in sizes 80,000 to 480,000 Btu per hr input in 40,000 Btu increments. For natural, manufactured and L.P. Gas.



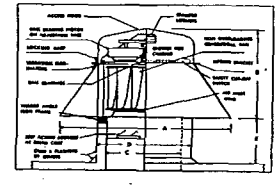


EJECTAIR Power Roof Ventilator by HAYES

Centrifugal backwardly curved blower type ... the blower wheel cannot be overloaded regardless of length or curves in the duct or variation in design. It is belt driven to provide flexibility. The ventilator is mounted on a built up curb on the roof. Special features include low profile with streamlined design and a corrosion resistant housing. The motor and fan bearings are grease packed ball bearings and are out of the air stream. A manually operated safety cutoff switch is provided.

DIMENSIONS

Unit Size	A	B	C	D	E	Approx. Ship. Wt.
1000	24¾	15½	11½	19½	16	130
1200	28	17½	12	21	16	137
1300	29	24¼	13½	21	16	180
1600	35½	25½	16½	24½	16	275
1800	39½	27½	20	28	16	300
2000	43	30	20	28	16	325
2200	46½	30½	22½	30½	16	350
2400	49½	32½	25	33	16	500
2700	55¼	34¾	28	36	16	590
3000	60¾	36½	31	39	16	630
3300	66	40	33½	41	16	700
3600	75	42½	40½	48½	16	795
4000	85	45⅝	40	52	16	875
4400	90½	49	44½	56	16	1120
4900	98¾	54	50	62	16	1500
5400	108	57½	55	67	16	1900



LARCO, INC.

LINTERN PRODUCTS

Route 20 East, Painesville, Ohio

Crane Cab Conditioners • Pulpit Conditioners • Crane Cabs

For steel, copper, and aluminum plants, forge shops, paper mills, glass plants, foundries, cement plants, incinerators ... wherever on-the-spot cooling is required ... for temperature ranges up to 230°F.



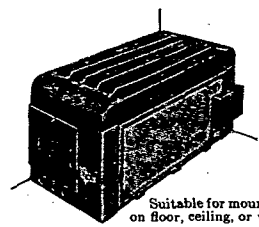
Air Cooled Condenser... the model supplied for highest temperatures

CRANE AIRE-RECTIFIERS

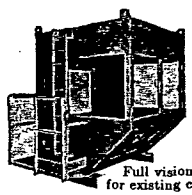
This Lintern air conditioning unit is of the split system type allowing the condenser to be located high above the usual heat source on footwalk or trolley where the cab is so attached, as on soaking pit or stripper cranes.
The condenser is connected to the cooling coil in the cab by two refrigerant lines, of high pressure hydraulic hose especially designed for "Freon" refrigerants, fitted with high quality "Hoze-Lok" fittings, eliminating soldered copper lines.

CAB UNITS

This new Cab Unit is unique in that it can be easily mounted not only on ceiling or floor of crane cabs and pulpits, but also vertically against wall at any desired point between ceiling and floor.
Compactly designed, fabricated of 11 gage steel, the unit includes an all copper evaporator or cooling coil, electric heat, dust and dirt filters, and activated carbon filter for elimination of odors and waste gases. A feature is the condensate fiber glass drip pan, light, easily handled, non-corrosive, and with large drain connection.



Suitable for mounting on floor, ceiling, or wall



Full vision cabs for existing cranes

CRANE CABS

These full vision Lintern crane cabs are of our standard construction design requiring a minimum of engineering, permitting quick shipments, keeping costs down. They have a minimum of through-metal to reduce the cooling load. Provision is made for wiring controls and switches, as required. (Also furnished with new switches and controls, if desired.) Well designed, properly insulated, they centralize responsibility for proper, cooling performance.

PULPIT AIRE-RECTIFIERS

The Lintern Split System Aire-Rectifier allows for a minimum of space consuming equipment within the pulpit, provides thoroughly cooled, dehumidified even air distribution throughout the pulpit without costly, space-using air ducts.
Condensers are located conveniently adjacent to the pulpit, connected to the cooling coils within the pulpit by high pressure hydraulic hose, especially designed for "Freon" refrigerants, and attached with "Hoze-Lok" fittings. A majority of the parts are interchangeable with the Crane Cab Conditioners, minimizing the number of spare parts required in any mill.



Cooling Unit mounted on ceiling

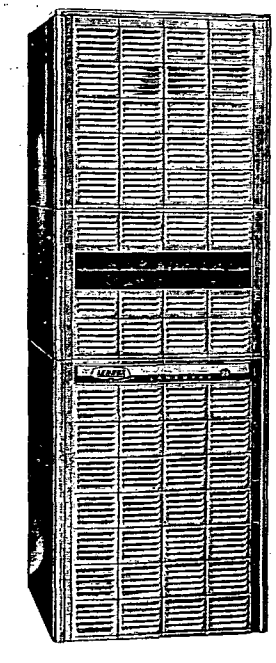
LENNOX INDUSTRIES INC.

Marshalltown, Iowa

Founded 1895. Plants located in Marshalltown and Des Moines, Ia.; Syracuse, N. Y.; Columbus, O.; Decatur, Ga.; Fort Worth; Los Angeles; Salt Lake City. In Canada: Toronto, Montreal, Calgary, Vancouver, Winnipeg

LANDMARK* SERIES, HEATING AND AIR CONDITIONING, SEPARATE COMPONENTS OF MODULAR DESIGN TO ASSEMBLE INTO FULLY INTEGRATED UNITS

One cabinet contains the air filter and blower; another contains the heating section, whether it is gas or oil, ranging from 68,000 Btuh with gas, and from 91,000 Btuh to 378,000 Btuh of input with oil. Another cabinet contains the cooling coil, or the combination heating and cooling coil in the case of the Heat Pump. Heat pump and electric strip heating can also be used with Landmark units. (Capacities can be increased by banking units.) Cooling and air handling capacities range from two tons through ten tons of refrigeration. Because of different capacities and different fuels for the heating section of each unit, the system designer can achieve perfect balance between heating and cooling in a building, whether it is residential or commercial, in the North or in the South.
For complete information on Landmark heating and air conditioning, or specialized assistance, phone, write, or wire Lennox Industries Inc., Marshalltown, Iowa.



LENNOX LANDMARK CFM PERFORMANCE TABLE

STATIC PRESSURE EXTERNAL TO LANDMARK UNIT

Landmark Combinations		.05"		.10"		.20"		.35"		.50"		.65"	
	CFM	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
Heating Only (Up-Flo or Down-Flo)	600	340	.02	350	.031	440	.045	475	.055	505	.07	575	.095
	700	390	.03	395	.045	460	.06	490	.07	525	.08	580	.10
	800	375	.04	410	.06	480	.075	520	.09	550	.10	610	.125
	900	365	.055	450	.075	510	.10	550	.12	575	.13	630	.15
	1000	400	.08	475	.10	540	.125	570	.14	600	.155	655	.185
	1200	480	.13	550	.16	605	.19	630	.21	655	.23	700	.255
	1400	560	.20	620	.24	670	.28	690	.29	715	.31	760	.35
	1600	645	.31	650	.32	680	.35	760	.415	780	.44	825	.48
	800	340	.025	350	.04	440	.065	475	.08	520	.09	—	—
	1000	300	.05	390	.06	470	.095	500	.105	530	.12	610	.155
	1200	370	.08	430	.10	570	.13	530	.14	570	.16	625	.19
	1400	425	.12	485	.145	550	.175	575	.19	605	.215	640	.25
	1600	480	.16	530	.21	590	.24	625	.268	650	.28	700	.325
	1800	540	.24	590	.275	645	.32	670	.35	690	.37	740	.42
	2000	600	.33	650	.38	695	.43	720	.455	740	.475	785	.525
	2200	670	.45	705	.50	745	.54	765	.57	790	.595	830	.66
Heating, Cooling and Heat Pump Up-Flo or Down-Flo	600	290	.03	380	.04	445	.06	510	.07	560	.08	618	.10
	700	320	.04	410	.06	500	.08	530	.09	565	.10	630	.12
	800	370	.05	450	.07	525	.10	555	.11	590	.12	650	.14
	900	415	.07	490	.10	555	.13	590	.14	620	.15	680	.18
	1000	466	.10	530	.13	595	.15	625	.17	650	.18	710	.22
	1200	535	.17	615	.20	673	.24	700	.26	725	.28	775	.31
	1400	650	.27	700	.31	750	.35	775	.37	800	.39	850	.44
	1600	740	.39	785	.44	830	.49	860	.52	880	.54	924	.59
	800	295	.02	400	.05	480	.073	520	.088	560	.105	625	.135
	1000	365	.06	450	.083	525	.11	560	.125	595	.14	655	.175
	1200	440	.10	510	.125	575	.14	610	.18	640	.20	695	.237
	1400	515	.157	575	.19	536	.23	663	.25	690	.27	740	.33
	1600	587	.225	640	.27	700	.32	720	.337	745	.36	790	.42
	1800	660	.33	710	.375	760	.42	780	.46	800	.47	840	.515
	2000	730	.45	775	.50	820	.555	840	.58	860	.60	900	.655
	2200	805	.60	845	.65	880	.70	900	.73	920	.765	950	.81

All CFM capacities listed here are extremely quiet, greater delivery can be achieved at a slight increase in sound level

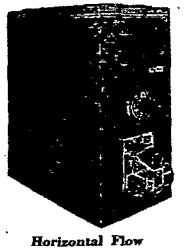
LENNOX INDUSTRIES INC. has more than 140 products to fill any heating and/or air conditioning need.

* Trademark, Patents Pending

The Majestic Co., Inc.,

Huntington, Ind.

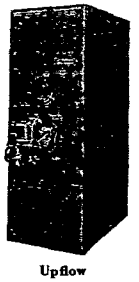
Majestic 250-400 Series Furnaces—Winter Air Conditioners in Two Sizes and Twelve Models for Oil or Gas, in Up, Down, or Horizontal-Flow Design



Horizontal Flow



Downflow



Upflow

Model No.	Btu Input*	Blower CFM @ ½" Total Stat. Pres.	Btu at Plenum	Casing Size W-D-H	W. A. & C. A. Plenum Size	Smoke Collar	Blower Motor	Approx. Ship. Wt.
U-250-G	250,000	3,000	200,000	32 × 49½ × 91	24 × 39½	8"	½ HP	820
U-250-P	286,000	3,000	220,000	32 × 49½ × 91	24 × 39½	8"	½ HP	855
U-400-G	400,000	4,000	320,000	32 × 55½ × 91	24 × 45½	9"	¾ HP	975
U-400-P	429,000	4,000	335,000	32 × 55½ × 91	24 × 45½	9"	¾ HP	1,100
D-250-G	250,000	3,000	200,000	32 × 49½ × 91	24 × 39½	8"	½ HP	820
D-250-P	286,000	3,000	220,000	32 × 49½ × 91	24 × 39½	8"	½ HP	855
D-400-G	400,000	4,000	320,000	32 × 55½ × 91	24 × 45½	9"	¾ HP	975
D-400-P	429,000	4,000	335,000	32 × 55½ × 91	24 × 45½	9"	¾ HP	1,100
H-250-G	250,000	3,000	200,000	32 × 49½ × 67	24 × 39½	8"	½ HP	715
H-250-P	286,000	3,000	220,000	32 × 49½ × 67	24 × 39½	8"	½ HP	750
H-400-G	400,000	4,000	320,000	32 × 55½ × 67	24 × 45½	9"	¾ HP	900
H-400-P	429,000	4,000	335,000	32 × 55½ × 67	24 × 45½	9"	¾ HP	1,000

*Oil Btu input based on 143,000 Btu per gallon. G—gas. P—oil.
Gas units also available, same specifications, for LP gas.

Here are furnaces for large homes, office buildings, churches, schools, lodge buildings, factories—furnaces *not* just made bigger but expressly designed to deliver high bonnet capacity with the desired space and fuel economy.

From top to bottom, these superbly engineered units are built for just one function . . . to extract every usable bit of heat from each unit of fuel and to deliver that heat quietly and efficiently through any type of modern forced-air system.

In the upflow models, tandem blowers are installed beneath the heat exchanger, while in the downflow models tandem blowers are mounted in a compartment above the furnace, delivering volumes of air into a *sub-floor* plenum for modern perimeter distribution. A separate blower compartment is provided with horizontal-flow models so that the blower unit may be mounted on either side or rear of the furnace.

Power gas burners with inputs of 250,000 and 400,000 Btu are furnished on the gas models. Oil burners are rated at 286,000 and 429,000 Btu input. Both oil and gas units utilize a pre-cast ceramic combustion chamber to shape and contain the flame for full efficiency. The heat exchanger is seam-welded of heavy steel plates and is surrounded by a four-tube secondary of unique design.

Majestic also offers a full line of smaller furnaces and a complete series of system-matching summer air conditioners in a wide range of sizes and types.

For full details and specifications, write The Majestic Co., Inc., Heating and Air Conditioning Division, Huntington, Indiana.

Pennsylvania Furnace & Iron Company

Warren, Pennsylvania
316 North Pine Street



Series 10A Pennsylvania Forced Air Gas Furnace

COMBUSTION CHAMBER: Extra large providing increased heating surface. Heavy gage metal insures long life.

HEAT EXCHANGER: Die stamped corrugated sections furnish many flue ways from the combustion chamber and provide extra radiating surfaces. Corrugations create staggered flow of air across the flue ways. All seams in combustion chamber and heat exchanger are electric welded and gas tight.

BLOWER: Multiblade centrifugal type with double inlet and single bottom discharge equipped with special self-aligning bearings large oil reservoirs, rubber mounted for quiet operation, low speed motor, either split phase of capacitator type depending on size. Overload protector, adjustable pulley, and V-belt drive.

FILTERS: Spun glass. Large area for *efficient filtering* over long periods.

BURNERS AND CONTROLS: Cast iron burners with raised ports accurately drilled for each size furnace and for the kind of gas to be burned. Venturi injection tubes prevent flash back. Operation is noiseless. Controls governed by room thermostat. Entirely automatic.

CASING: Heavy and rigid. Doors provide easy access to blower and heating chamber. Controls enclosed. Finish hammertone steel blue enamel baked on. All *joints tight. Furnace* easily erected.

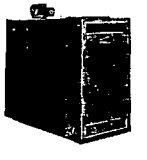
This furnace was designed for those who want a high grade and efficient unit in the better class of homes. Made in seven sizes. Input 75,000 to 225,000 Btu. *A.G.A.* approved.

SERIES 11 ULTRA FURNACE

This furnace is designed for gas or oil where standby equipment is specified. It is made of 12 gage in three sizes: 75,000, 100,000, and 125,000 Btu. Can be furnished with gas burners and controls or with gun type oil burner. Efficient on either fuel. *Has* the same blower and filter equipment as the 10A. (Furnace not shown).

SERIES 55 PENNSYLVANIA GRAVITY FURNACE

This line of gravity furnaces is made in three sizes with inputs of 75,000 to 125,000 Btu. The combustion chamber and radiators are cylindrical. The rectangular casing has corrugated inner lining providing efficient insulation. The burner is single port type with venturi outlet and chrome flame spreader. Controls are automatic with special safety feature. Casing is finished in steel blue baked enamel.



PENNSYLVANIA GAS CONVERSION BURNER

(Not Shown)

This is a single port burner with automatic controls suitable for installation in coal or oil fired furnaces. Three sizes are available; 75,000 to 350,000 Btu input.

PENNSYLVANIA MULTITUBE GAS CONVERSION BURNER

(Not Shown)

This burner is particularly designed for use in high pressure and tubular boilers in commercial and industrial installations. They are regularly installed through fire or ashpit doors in single or double units. Fully automatic controls are provided. Capacities range from 720,000 to 4,680,000 Btu.

REFRIGERATION APPLIANCES, Inc.

KRACK Refrigeration Appliances, Inc.

909 West Lake Street, Chicago 7, Illinois

Manufacturers of Freon, Ammonia & Flooded Ammonia Heat Transfer Equipment Since 1931

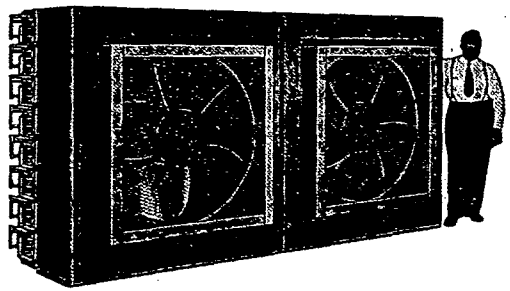
AIR COOLED CONDENSERS

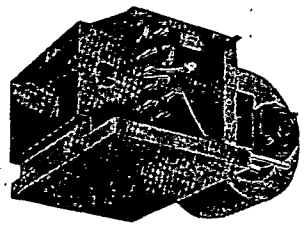
Quiet, remote type

Multiple banking in the field permits unlimited capacity ranges. Counter-flow refrigerant circuiting. Multi-circulating at no extra cost.

Automatic Winter Control: Unique design eliminates low operating pressure and expansion valve problems due to falling temperatures. One valve controls pressure in condenser, one controls pressure in receiver and a check valve stops liquid escape from receiver to condenser under all conditions.

Rated Capacities: Single units 7.5 to 50 tons (belt drive, patented semi-automatic adjustment), 1 to 7.5 tons (direct drive). Certified nominal capacities based on 110F condensing temperature, 90F ambient.



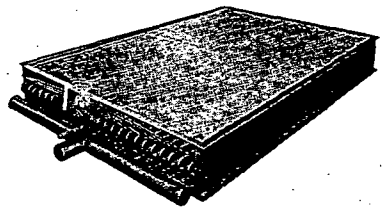


TRANSOM AIR CONDITIONERS

Quiet-remote type for small jobs

All aluminum construction. Insulated double drain pan. 3-speed control and thermal overload protection on motor.

Rated Capacities: 9000 and 12,000 Btu/hr.



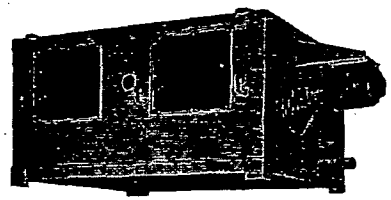
STANDARD AIR CONDITIONING COILS

For new or old installations

Patented construction features. Positive fin spacing.

Rated Capacities: 2 to 50 tons, 4 or 6 rows deep.

Special sizes: ⅜, ½, ⅝ or ¾ in. OD tubes with 3 to 12 fins/in. available. All copper and steel hot-dip galvanized after fabrication coils available.



COOLING OR HEATING UNITS

Quiet-remote type

Choice of coil arrangements for direct expansion, water, steam or combinations.

Rated Capacities: 2 to 50 tons. Use with or without ducts for ceiling or floor. Insulated cabinets. Rust-resistant blowers.

FURNACE AIR CONDITIONING COILS

Low cost-bonnet type

Converts old or new forced-air systems to summer cooling.

Rated Capacities: 24,000 to 90,000 Btu/hr.

EVAPORATIVE CONDENSERS AND COOLING TOWERS

Freon or ammonia for indoors and outdoors.

Rated Capacities: 5 to 100 tons.

REQUEST NEW DETAILED BULLETINS

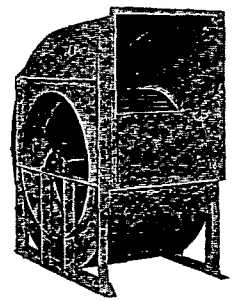
UTILITY FAN CORPORATION

A Division of Utility Appliance Corp.

911 East 59th Street, Los Angeles 1, Calif.

Manufacturers of heavy and standard duty blowers for heating, air conditioning and ventilating installations. Producers of blowers and blower part for original equipment manufacturers. Built, tested and rated in accordance with ASHAE and *AMCA* codes.

- Sizes 12 in. to 73 in.
- **Write for catalog data.**

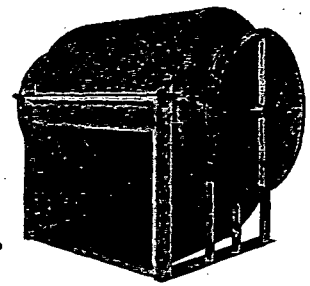


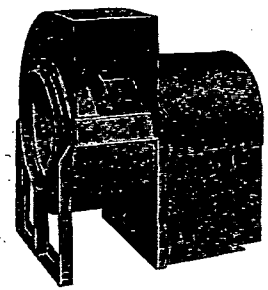
Heavy Duty Type "F" Design 29 Blowers are of the forward curved blade type for extraordinary efficiency and quiet operation. Low tip speeds give lower noise levels. These blowers are available in sizes from 12 to 73 in., single and double width, Class I and Class II construction and all standard drive arrangements and discharge positions.

• Heavy Duty Type "F" Blower, SISW

Heavy Duty Type "B" Design 29 Blowers are of the backward inclined blade type for high efficiency at higher static pressures and non-overloading horsepower characteristics. These blowers are available in sizes from 12 to 73 in., single and double width, Classes I, II, III and IV construction and all standard drive arrangements and discharge positions.

Heavy Duty Type "B" Blower with Variable Inlet Vanes, DIDW •





Utility Sets for a wide variety of ventilation and exhaust applications are available in sizes from 10 to 37 in. with Type "F" and Type "B" designs, single inlet arrangement Number 9 with rotatable housings for any desired discharge position. These Utility sets are completely self-contained units, ready to run with a weather-proof drive enclosure available for roof installation.

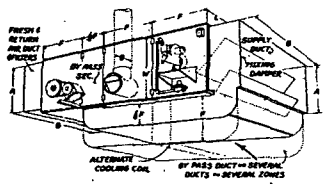
• Utility Set Type "B" with Weather-Proof Drive Enclosure

Campbell "WINTER-CHASER"

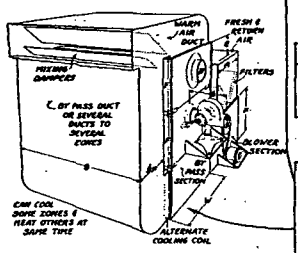
CAMPBELL HEATING CO.

3121 Dean Ave., Des Moines 17, Iowa

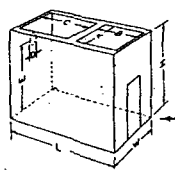
Telephone AMherst 6-5169



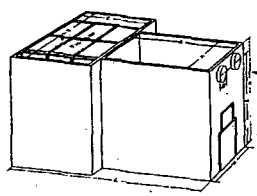
HORIZONTAL UNIT HEATER



HI-BOY UNIT HEATER



LO BOY HEATER



OUTSTANDING PERFORMANCE

You can be sure of outstanding performance at low cost, when you specify Campbell Warm Air Heaters. Fast *heat delivery, positive control of temperature,* humidity and ventilation—PLUS—low first cost, low installation cost, and low operating cost, all add up to customer satisfaction.

ENGINEERING SERVICE

Our Engineering Department will be glad to help the consulting engineer or architect solve any heating, ventilating or air conditioning problems or help with layouts and specifications for any large building. Our Engineering Department consists of trained graduate engineers backed by 73 years of practical experience.

HEATER						BLOWER‡					
Unit No. *	Output Capacity BTU/Hr. †	Heating Surface Sq. Ft.	Steel Thickness Primary	Steel Thickness Secondary	BTU per Sq. Ft. Heating Surface	Grate Area Sq. Ft.	Normal CFM 140° Discha. Temp.	Wheel Diam. x Width Inches	Assumed External Static In. Water	RPM	Motor Size (HP)
HI BOY & HORIZONTAL UNIT HEATER — OIL & GAS											
U225	225,000	84	14		2700		2750	Twin 13x12	½	725	½
U275	275,000	84	14		3300		3350	Twin 13x12		704	¾
U325	325,000	121	12		2700		4000	Twin 14x14		631	¾
U400	400,000	121	12		3300		4900	Twin 14x14		635	1
U500	500,000	182	11		2750		6100	Twin 16x16		570	1
U600	600,000	182	11		3300		7300	Twin 16x16		600	1½
U700	700,000	244	10		2850		8500	Twin 18x18		505	2
U800	800,000	244	10		3300		9800	Twin 18x18		515	2
U950	950,000	334	10		2850		11600	Twin 22x22		448	3
U1100	1,100,000	334	10		3300		13500	Twin 22x22		462	3
U1300	1,300,000	456	10		2850		16000	Twin 25x25		375	3
U1500	1,500,000	456	10		3300		18500	Twin 25x25		390	5
LO BOY OIL & GAS FIRED FOR LARGE											
H200	235,000	100			2350		2600	Twin 13x10	½	700	½
H250	275,000	125			2700		3300	Twin 13x10		706	¾
H300	340,000	136			2500		4000	18x18		506	¾
H350	400,000	152			2600		4600	18x18		516	1
H400	450,000	167			2700		5300	18x18		536	1½
H500	500,000	200			2750		6600	22x22		463	1½
H600	650,000	240			2700		8000	22x22		486	2
H700	750,000	280			2700		9200	Twin 18x18		516	2
8000 SERIES OIL, GAS, STOKER OR HAND FIRED											
8032	400,000	133	#7	#7	3000	3.2	5300	22½"	½"	677	1.0
8048	500,000	167				4.8	6600	22½"		760	1½
8060	600,000	200				6.0	8000	27"		547	1½
8074	700,000	233				7.4	9200	27"		586	1½
8075	840,000	280	#3	#7		7.5	11000	30"		521	2
8100	960,000	320				10.0	12500	33"		445	2
8125	1,080,000	360				12.5	14100	36½"		384	2
8150	1,320,000	440				15.0	17300	40½"		351	3
8175	1,440,000	480				17.5	18900	40½"		381	3
8200	1,800,000	600				20.0	23600	44½"		334	3
8250	2,250,000	750				25.0	29500	49"		307	5
8300	2,700,000	900				30.0	35400	54½"		274	5
8350	3,150,000	1050				35.0	41300	54½"		300	7½

* Type A units include Special Oil or Gas Heater, Casing, Blower, Motor, Drive, Filters, Humidifier, and Blower Control but less Burner, Controls and Built-in Combustion Chamber.
Type B units include same as Type "A plus Oil or Gas Burner and Built-in Combustion Chamber and Standard Controls.

† Based on 80% thermal efficiency for oil and gas and 70% for coal. Add 10% to 35% over calculated heat loss for pick-up load and 10% to 15% for duct radiation losses. The higher figure is preferable for best operating economy and life of the equipment.

Campbell "WINTER-CHASER"

CAMPBELL HEATING CO.

3121 Dean Ave., Des Moines 17, Iowa

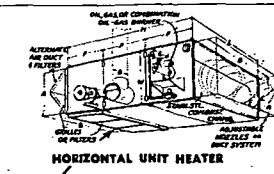
Telephone AMherst 6-5169

75 F HEATING GUARANTEE

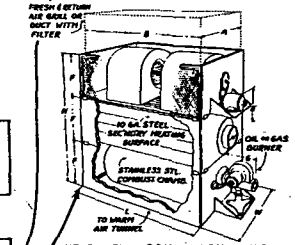
If the heating plant design is approved by our Engineering Department or designed by a Registered Engineer and the heater and blower are furnished by us and are in accordance with our ratings, we will guarantee the heating of all rooms to which warm air is delivered to 75F in the coldest and windiest weather.

10 YEAR DURABILITY GUARANTEE

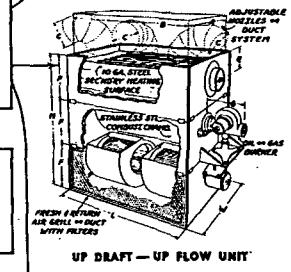
All Campbell Heaters installed as described above are guaranteed for 10 years against repairs from any cause, parts to be furnished free f.o.b. the factory. Motor, humidifier, automatic burner and controls and other parts not manufactured by us carry their manufacturers' guarantee of one year against defects.



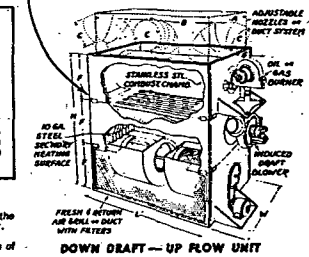
HORIZONTAL UNIT HEATER



UP DRAFT — DOWN FLOW UNIT



UP DRAFT — UP FLOW UNIT



DOWN DRAFT — UP FLOW UNIT

Filters All 2" Thick Sq. Ft.	MAX. FUEL RATE Oil GPH	1000 BTU Gas CFH	LP Gas GPH	Coal Lbs. Per Hr.	DIMENSIONS Largest Piece if KD ¶ Dimension Inches	Wt. lbs.	A in.	B in.	C in.	§ D in.	F in.	H in.	L in.	W in.	Approx. Shipping Weight Type A *	Type B
CHURCHES, SCHOOLS, COMMERCIAL & INDUSTRIAL BLDGS.																
12.5	2.0	280	2.8		22x45x48	500	19	44	12	9	22	66	48	23	1300	1350
12.5	2.5	350	3.5		22x45x48	500	19	44	14	9	22	66	48	23	1350	1400
14.5	2.9	400	4.0		24x29x54	700	25	50	14	10	24	72	54	29	1700	1750
14.5	3.6	500	5.0		24x29x54	700	25	50	16	10	24	72	54	29	1750	1800
21.0	4.5	625	6.3		28x34x60	1000	30	56	18	12	28	84	60	34	2200	2250
21.0	5.4	750	7.5		28x34x60	1000	30	56	18	12	28	84	60	34	2250	2300
34.0	6.2	875	8.8		33x40x78	1400	36	74	20	14	33	99	78	40	3250	3350
34.0	7.2	1000	10.0		33x40x78	1400	36	74	22	14	33	99	78	40	3300	3400
52.5	8.5	1200	12.0		36x48x108	1900	44	92	24	16	36	108	96	48	4400	4500
52.5	9.8	1375	13.8		36x48x108	1900	44	92	26	18	36	108	96	48	4500	4600
62.5	11.6	1625	16.3		42x54x114	2600	50	110	28	18	42	126	114	54	5800	6000
62.5	13.4	1875	18.8		42x54x114	2600	50	110	30	18	42	126	114	54	6000	6200
RESIDENCES, SMALL CHURCHES, SCHOOLS, ETC.																
9.0	2.1	293	2.9		18x41x30	450	37	25½	26½	8		52	75	41	1700	2100
11.0	2.5	344	3.4		29x41x40	550	37	25½	36¾	8		57	85	41	1900	2500
16.5	3.0	425	4.2		29x50x48	600	36	34	38	10		61	87	50	2100	2800
16.5		500	5.0		29x50x48	680	36	34	38	10		61	87	50	2400	3200
16.5	4.0	560	5.6		29x50x48	750	36	34	38	10		61	87	50	2700	3500
22.0	4.5	625	6.2		34x60x48	900	40	36	44	10		74	91	60	3200	4100
25.0	5.8	810	8.1		34x60x54	1100	40	36½	48½	12		76	97	60	3600	4600
33.0	6.7	940	9.4		39x80x54	1250	60	40	50	12		72	104	80	4000	5000
FOR LARGE SCHOOLS, CHURCHES, ETC.																
28.0	3.6	500	5.	45	29½x48x64	650	47	44¾	46	2-8	62	66	94½	51	2200	2800
33.0	4.5	625	6.3	56	29½x48x64	650	47	53	46	2-8	62	66	105	51	2400	3200
40.0	5.4	750	7.5	67	29½x48x64	650	47	61½	56	2-9	80	66	126	51	2750	3600
40.0	6.3	875	8.8	78	29½x48x64	650	47	70	56	2-9	80	66	134	51	3000	4200
50.0	7.5	1050	10.5	100	28x72x76	1100	72	76	56	2-9	80	80	140	76	4500	6000
55.0	8.8	1250	12.0	114	28x72x76	1100	72	89	66	2-10	84	80	157	76	5500	7200
66.0	9.7	1350	13.5	129	28x72x76	1100	72	101	71	2-12	92	80	180	76	6500	8500
82.0	11.8	1650	16.5	157	28x72x76	1100	72	114	76	2-12	104	80	198	76	7500	9700
82.0	12.9	1800	18.0	171	28x72x76	1100	72	126	76	2-14	104	80	210	76	9000	11500
125.0	16.1	2250	22.5	214	35x90x95	1700	90	133	86	2-14	108	100	227	94	10000	12700
133.0	20.1	2810	28.0	268	35x90x95	1800	90	152	92	2-16	116	100	252	94	12000	15000
155.0	24.1	3380	33.8	320	40x100x103	2100	100	152	108	2-16	28	118	268	104	15000	18000
194.0	28.1	3940	39.4	375	40x110x113	2400	110	152	108	2-18	28	118	268	114	18000	21000

‡ For other air volumes and static pressures consult our Engineering Department.

¶ Dimensions given are for the heater KD to be assembled and welded in the building. Ordinarily the heater will be put in the building in one piece before the building construction has progressed too far.

§ An induced draft blower can be furnished if a stack or chimney is not available. The gas passages of the heaters are adequate so an inducted draft blower is not ordinarily needed.

E. K. CAMPBELL COMPANY

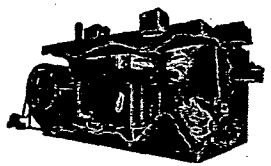
Manchester at 18th · Kansas City 26, Mo.

Sales and Service Offices in Principal Cities

E. K. Campbell Co. Thermidaire

Thermidaire

- Heavy Duty Warm Air Furnaces Capacities to 3,000,000 btu
- Builders of Custom Heating Equipment Since 1910

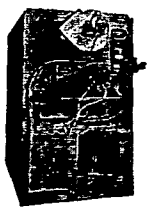


"Thermidaire" Design 10

"THERMIDAIRE" DESIGN 10 HEAVY DUTY FURNACES—CUSTOM HEAVY DUTY EQUIPMENT FOR ENGINEERED APPLICATIONS.

Typical Problems for which Design 10 Furnaces Offer an Excellent Solution:

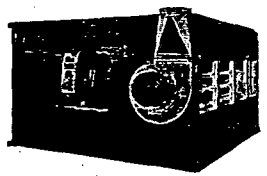
- **Need for constant ventilation**—To maintain freshness in any structure, particularly where high occupancy loads occur, modulating face and by-pass zone arrangements are used to obtain constant air circulation with "Thermidaire" Design 10 Heavy Duty Furnaces.
- **Need for air conditioning**—Since most modern structures have either present cooling requirements, or will have at some future date, the large air capacities needed and the basic distribution system required are readily available in the structure using "Thermidaire" Design 10 Heavy Duty Furnaces.
- **Need for very large capacity equipment**—Where large loads exist, the high output capacity available from single units simplifies the solution. "Thermidaire" Design 10 Heavy Duty Furnaces available in standard sizes with output to 8,000,000 Btuh and 145,000 cfm, larger on special request.
- **Need to meet special design requirements**—For balancing Btuh vs. cfm component sizing, separate sizing of furnace and blower elements permits "Thermidaire" Design 10 Heavy Duty Furnaces to meet special requirements for very high or low air volumes, or temperature rises, and to handle unusual jobs, process needs, or exceptional buildings.
- **Need for flexible or special equipment arrangements**—Where building dimensions limit outlet or inlet duct arrangements, location of firing front or position of blower relative to air conditioning or furnace components "Thermidaire" Design 10 Heavy Duty Furnaces are excellent due to the extreme flexibility possible in arranging equipment.
- **Need for full convertibility to any fuel**—Where dual fuel equipment is required, or in event of future changes in fuel requirements, (including gas, L. P., oil light or heavy, and solid fuels), "Thermidaire" Design 10 Heavy Duty Furnaces offer complete protection since any of these fuels can be efficiently burned.
- **Need for extra quality equipment**—"Thermidaire" Design 10 Heavy Duty Furnaces are the highest quality equipment available, designed for continuous duty and long life on tough jobs and large loads.
  Only Heavy Duty blower equipment is used.
  Only highest grade burner and control equipment used.
  Extra Heavy welded steel construction throughout.
  Baked enamel exterior finish.
  Fiberglas insulated casing.
  Low Internal resistance to flue gases.
  Operating efficiencies up to 84 percent with resultant fuel economy.
  Counter-Flow heat transfer.
  Positive pressure or Drive through principle.
- **Need for special services**—
  Our half-century of application experience in the specialized field of heavy duty warm air heating is available to assist in arriving at the proper solution to your problem.
  We recognize that satisfactory solution of the Users problem is the end objective and we back our equipment with competent field service to accomplish this.



"Thermidaire" Design 40

"THERMIDAIRE" DESIGN 40 HEAVY DUTY FURNACES

Standard heavy duty equipment for general heating applications—Available from 250,000 to 2,000,000 Btuh output in vertical, horizontal, and inverted models—available for gas, oil or combinations. Provides convenient compactness, high efficiency, simple installation. Blower, burner, ID fan and controls, mounted, wired and fire tested. Normally shipped assembled.



"Thermidaire" Design 41

"THERMIDAIRE" DESIGN 41 HEAVY DUTY FURNACES

Standard heavy duty equipment for engineered applications—designed for use with face and by-pass, constant ventilation, or air conditioning—requires separate blower—has built in by-pass section—available in same models and fuels as Design 40, in sizes from 250,000 to 3,000,000 Btuh output. Provides convenient compactness, high efficiency, simple installation. Burner, controls and ID fan, mounted, wired and fire tested. Normally shipped assembled.

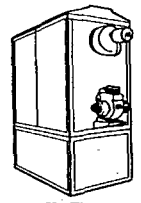
OTHER DESIGNS AND SPECIAL EQUIPMENT AVAILABLE ON REQUEST

CHICAGO STEEL FURNACE CO.

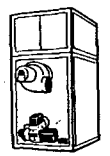
Home Office & Plant

9326 S. Anthony Ave., Chicago 17, Illinois

Engineering Sales Offices from Coast to Coast **REgent 1-0280**

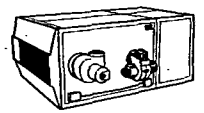


Up Flow

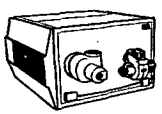


Counter Flow

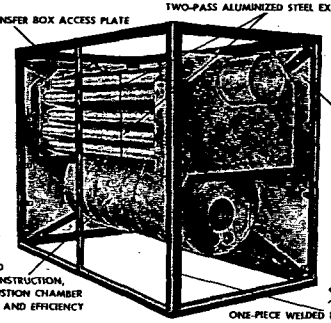
Up Flow and Counter Flow Horizontal Furnaces and Duct Heaters
120,000 to 2,500,000 Btu Output
Single and Dual Fuel; Oil and/or Gas



Horizontal Furnace



Duct Heater



Stainless Steel Unit

Chicago Steel Furnace Co. announces the production of a new line of steel furnaces made of stronger 442 Stainless in capacities to 2½ million Btu's in all popular designs. Chicago's 442 Stainless line supplements the refractory lined series still available.

All models available with either duct flange connections or (optional) with 4-way adjustable diffuser heads and screened intakes.

Cuts and features here shown pertain to our stainless steel series of furnaces only. Firing equipment, all fuels, *UL* and *A.G.A.* inspected and approved.

STAINLESS STEEL SERIES*

SPECIFICATIONS							UPFLOW					COUNTERFLOW					HORIZONTAL					DUCT HEATER				
Output Btu/Hr	CFM @ ½" E. S. P.	Blower Wheel	Blower HP	Inducer Outlet	Inducer HP	Diff. Head	Type Size	Cabinet Dimens. H	L	W	Approx. Ship. Wt.	Type Size	Cabinet Dimens. H	L	W	Approx. Ship. Wt.	Type Size	Cabinet Dimens. H	L	W	Approx. Ship. Wt.	Type Size	Cabinet Dimens. H	L	W	Approx. Ship. Wt.
120,000	1,500	9- 9"	½	8¼ × 5⅝	¼	4	UT- 120	55	32	24	800	CT- 120	51	32	24	780	HT- 120	24	55	32	800	DT- 120	24	32	32	680
160,000	2,000	9- 9"	¾	8¼ × 5⅝	¼	4	UT- 160	55	32	24	800	CT- 160	51	32	24	780	HT- 160	24	55	32	800	DT- 160	24	32	32	680
200,000	2,500	9-10"	¾	8¼ × 5⅝	¼	4	UT- 200	70	40	30	970	CT- 200	65	40	30	950	HT- 200	30	70	40	970	DT- 200	30	50	40	850
250,000	3,200	9-10"	1	8¼ × 5⅝	¼	4	UT- 250	70	40	30	970	CT- 250	65	40	30	950	HT- 250	30	70	40	970	DT- 250	30	50	40	850
300,000	3,800	9-12"	1	8¼ × 5⅝	¼	4	UT- 300	78	53	34	1200	CT- 300	71	53	34	1170	HT- 300	34	78	53	1200	DT- 300	34	54	53	1060
400,000	5,000	9-12"	1½	8¼ × 5⅝	¼	4	UT- 400	78	53	34	1200	CT- 400	71	53	34	1170	HT- 400	34	78	53	1200	DT- 400	34	54	53	1060
500,000	6,200	9-15"	2	8¼ × 5⅝	¼	4	UT- 500	83	62	38	1540	CT- 500	78	62	38	1520	HT- 500	36	83	62	1540	DT- 500	38	59	62	1380
600,000	7,500	9-15"	3	8¼ × 5⅝	¼	4	UT- 600	83	62	38	1540	CT- 600	78	62	38	1520	HT- 600	36	83	62	1540	DT- 600	38	59	62	1380
700,000	8,800	9-18"	3	10¾ × 8¼	¼	6	UT- 700	87	78	40	1980	CT- 700	82	78	40	1940	HT- 700	40	87	78	1980	DT- 700	40	61	78	1750
800,000	10,000	9-18"	3	10¾ × 8¼	¼	6	UT- 800	87	78	40	1980	CT- 800	82	78	40	1940	HT- 800	40	87	78	1980	DT- 800	40	61	78	1750
900,000	11,300	9-18"	3	11¾ × 7¼	½	6	UT- 900	98	80	45	2320	CT- 900	92	80	45	2270	HT- 900	45	98	80	2220	DT- 900	45	66	80	2040
1,000,000	12,500	9-18"	5	11¾ × 7¼	½	6	UT-1000	98	80	45	2320	CT-1000	92	80	45	2270	HT-1000	45	98	80	2220	DT-1000	45	66	80	2040
1,250,000	15,800	2-20"	5	11¾ × 7¼	½	6	UT-1250	104	92	46	2880	CT-1250	98	92	46	2830	HT-1250	46	104	92	2880	DT-1250	46	74	92	2580
1,500,000	18,600	2-23"	7.5	11¾ × 7¼	½	6	UT-1500	108	100	54	3590	CT-1500	102	100	54	3510	HT-1500	54	108	100	3590	DT-1500	54	78	100	3280
1,750,000	22,000	2-20"	7.5	13¾ × 9¾	¾	8	UT-1750	111	120	50	4400	CT-1750	106	120	50	4300	HT-1750	50	111	120	4400	DT-1750	50	81	120	4060
2,000,000	25,000	3-23"	7½	13¾ × 9¾	¾	8	UT-2000	115	129	52	4450	CT-2000	109	129	52	4350	HT-2000	53	115	129	4450	DT-2000	53	81	129	3980
2,250,000	28,000	3-23"	10	13¾ × 9¾	¾	8	UT-2250	132	140	62	5850	CT-2250	126	140	62	5740	HT-2250	62	132	140	5850	DT-2250	62	97	140	5350
2,500,000	29,000	3-23"	15	13¾ × 9¾	¾	8	UT-2500	132	140	62	5850	CT-2500	128	140	62	5740	HT-2500	62	132	140	5850	DT-2500	62	97	140	5350

* Specifications and dimensions subject to change without notice. Contact factory for certified dimensions. Some changes currently being considered.

REFRACTORY LINED SERIES**

SPECIFICATIONS							UPFLOW					COUNTERFLOW					HORIZONTAL					DUCT HEATER				
Output Btu/Hr	CFM @ ½" E. S. P.	Blower Wheel	Blower HP	Inducer* Outlet	Inducer* HP	Diff. Head	Type Size	Cabinet Dimens. H	L	W	Approx. Ship. Wt.	Type Size	Cabinet Dimens. H	L	W	Approx. Ship. Wt.	Type Size	Cabinet Dimens. H	L	W	Approx. Ship. Wt.	Type Size	Cabinet Dimens. H	L	W	Approx. Ship. Wt.
120,000	1,500	9- 9"	¼	8¼ × 5⅝	¼	4	UR- 120	76	34	32	780	CR- 120	76	34	32	780	HR- 120	36	84	33	796	DR- 120	38	57	33	680
160,000	2,000	9- 9"	¾	8¼ × 5⅝	¼	4	UR- 160	76	34	32	780	CR- 160	78	34	32	780	HR- 160	38	84	33	800	DR- 160	38	57	33	700
200,000	2,500	9-10"	¾	8¼ × 5⅝	¼	4	UR- 200	82	43	31	1000	CR- 200	82	43	31	1000	HR- 200	38	87	43	1100	DR- 200	38	59	43	1000
250,000	3,200	9-10"	1	8¼ × 5⅝	¼	4	UR- 250	82	43	31	1000	CR- 250	82	43	31	1000	HR- 250	38	87	43	1100	DR- 250	38	59	43	1000
300,000	3,800	9-12"	1	8¼ × 5⅝	¼	4	UR- 300	93	53	44	1500	CR- 300	93	53	44	1500	HR- 300	45	108	53	1700	DR- 300	48	68	53	1400
400,000	5,000	9-12"	1½	8¼ × 5⅝	¼	4	UR- 400	93	53	44	1500	CR- 400	93	53	44	1500	HR- 400	45	108	53	1700	DR- 400	48	68	53	1400
500,000	6,200	9-15"	2	8¼ × 5⅝	¼	4	UR- 500	84	91	42	2800	CR- 500	84	91	42	2800	HR- 500	45	126	62	2800	DR- 500	45	80	62	2200
600,000	7,500	9-15"	3	8¼ × 5⅝	¼	4	UR- 600	84	91	42	2800	CR- 600	84	91	42	2800	HR- 600	45	126	62	2800	DR- 600	45	80	62	2200
700,000	8,800	9-18"	2	10¾ × 8¼	¼	6	UR- 700	93	100	50	4000	CR- 700	93	100	50	4000	HR- 700	52	150	75	4000	DR- 700	52	110	75	3400
800,000	10,000	9-18"	3	10¾ × 8¼	¼	6	UR- 800	93	100	50	4000	CR- 800	93	100	50	4000	HR- 800	52	150	75	4000	DR- 800	52	110	75	3400
900,000	11,300	9-18"	3	11¾ × 7¼	½	6	UR- 900	110	110	54	5200	CR- 900	110	110	54	5200	HR- 900	54	158	82	5100	DR- 900	54	114	82	4300
1,000,000	12,500	9-18"	5	11¾ × 7¼	½	6	UR-1000	110	110	54	5200	CR-1000	110	110	54	5200	HR-1000	54	158	82	5100	DR-1000	54	114	82	4300
1,250,000	15,800	2-20"	5	11¾ × 7¼	½	6	UR-1250	112	118	56	5800	CR-1250	112	118	56	5800	HR-1250	58	162	96	5800	DR-1250	56	118	96	4800
1,500,000	18,600	2-23"	7.5	11¾ × 7¼	½	6	UR-1500	115	128	62	6700	CR-1500	115	128	62	6700	HR-1500	62	180	96	6300	DR-1500	62	128	96	5400
1,750,000	22,000	2-20"	7.5	13¾ × 9¾	¾	8	UR-1750	118	132	64	7500	CR-1750	118	132	64	7500	HR-1750	62	180	124	7500	DR-1750	62	128	124	6200
2,000,000	25,000	3-23"	7½	13¾ × 9¾	¾	8	UR-2000	125	144	66	8500	CR-2000	125	144	66	8500	HR-2000	62	180	124	8500	DR-2000	62	128	124	7020

Over 2,000,000: Contact factory on needs, custom manufactured.
* Draft Inducer is Optional on Refractory-Lined Series.
** Specifications and dimensions subject to change without notice. Contact factory for certified dimensions. Some changes currently being considered.

DRAVO CORPORATION

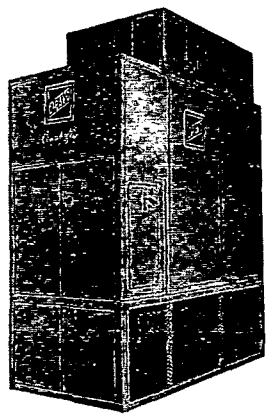
Neville Island

Pittsburgh 30, Penna.

Atlanta, Baltimore, Boston, Chicago, Cincinnati, Cleveland, Detroit, Dallas, Indianapolis, New York, Philadelphia, Pittsburgh, San Francisco, St. Louis
Sales Representatives in Principal Cities

DRAVO *Counterflo* COMMERCIAL SPACE HEATERS

350,000 to 2,000,000 Btu/hr output



Dravo Commercial Space Heaters, in addition to comfort heating, are easily adapted to heat curing and process drying, tempering or replacement air, as duct furnaces, and for summertime ventilation.

●**PYROJET BURNER**—Designed and built by Dravo exclusively for the Commercial heater, the induced draft Pyrojet burns gas, light oil, heavy No. 5 oil or combination gas/light oil. Combination firing has manual or automatic switchover control.

●**STAINLESS STEEL COMBUSTION CHAMBER**—Designed for maximum flame travel and efficient wiping action of air stream for efficient heat transfer. Ruggedly built for long, trouble-free service.

●**SAFETY TESTED**—Approved by *A.G.A.* and listed by *Underwriters Laboratories, Inc.*, Dravo Safety Control Circuit is accepted by *Factory Mutual Engineering Division* and *Factory Insurance Association*.

●**10-YEAR GUARANTEE**—All standard model **Commercial** Heaters carry a fully bonded 10-year guarantee on all parts in direct contact with flame or flue gases.

DRAVO *Counterflo* INDUSTRIAL SPACE HEATERS

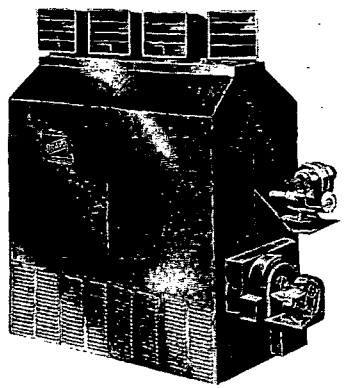
400,000 to 2,000,000 Btu/hr output

●**ECONOMICAL**—Package type burner for gas, light oil, No. 4 oil or automatic switchover from gas to light oil. Easy to service, this is a rugged mill-type heater which will provide long-time, low-cost, efficient heating in most types of installations.

●**STAINLESS STEEL COMBUSTION CHAMBER**—Full floating stainless steel combustion chamber, designed for four-pass flame travel to assure maximum heat transfer. Economizer tubes whirlcool the hot gases for added efficiency.

●**APPROVALS**—Burner and control components approved by *Underwriters Laboratories, Inc.*, and by *A.G.A.* where applicable.

●**10-YEAR GUARANTEE**—All standard model **Industrial** Heaters carry a fully bonded 10-year guarantee on all parts in direct contact with flame or flue gases.



The Dravo Industrial Space Heater is a rugged mill-type economy heater. Easy serviceability and long life make this the most economical buy for many installations. Is readily adaptable to process uses, for tempering of replacement air and many other applications.

DRAVO *Paraflo* SPACE HEATERS

200,000 to 250,000 Btu/hr. output

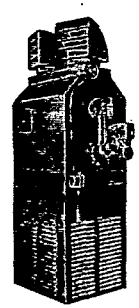
●**POWER AND FUEL**—Heater operates on 120 volt 60 cycle single-phase current. 440/220 volt three-phase operation can be supplied if desired. Burns light oil or gas.

●**FAST TRANSFER OF HEAT**—Stainless steel combustion chamber allows faster heat transfer and guarantees longer heater life and sustained efficiency. Equipped with *UL* approved gun-type oil burner or *A.G.A.* approved gas burner.

●**FLOATING COMBUSTION CHAMBER**—Supporting straps hold stainless steel chamber in place for inverted or horizontal mounting.

●**HEAT ANY DIRECTION**—Warms areas up to 3600 sq ft. Directional nozzle can be rotated 360° heating any desired location, high velocity discharge assures adequate distribution up to 90 ft from the unit without the use of ductwork.

DRAVO Gas Fired Suspended Unit Heaters available for 40,000 to 184,000 Btu/hr applications.

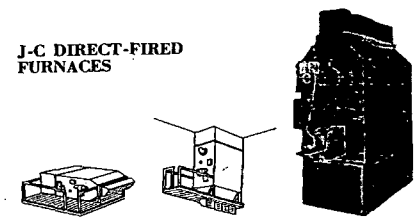


DRAVO *Paraflo*

JACKSON & CHURCH

Division of York-Shipley, Inc., York, Pa.

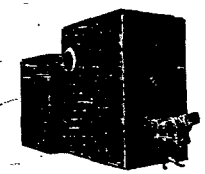
J-C DIRECT-FIRED FURNACES



These new *units incorporate* a true counter-flow design. Heating surfaces are arranged for highly efficient heat transfer, allowing minimum pressure drop through the heater. Flexible application as they can be installed upright, laid on their side, or inverted. *Fully automatic controls*, installed and wired for complete programming sequences. For No. 2 oil, natural, mixed, *manufactured, L. P. gases or combination.*

	Output 1000 Btu	CFM	H In.	W In.	L In.
JC-40	400	4450	89	38	63
JC-50	500	5450	89	38	63
JC-75	750	8170	110	43	78
JC-100	1000	10900	110	43	78
JC-125	1250	13600	120	56	97
JC-150	1500	16320	120	56	97
JC-175	1750	19050	131	80	105
JC-200	2000	21800	131	80	105

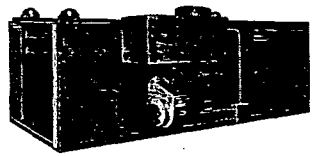
J-C POWERATED TUBULAR SERIES



Firing with light oil and/or natural gas in all capacities. Architects and engineers will find in this line the latest engineering advances in furnaces designed for heavy duty requirements.

Model No.*	Output 1000 Btu	CFM Bonnet	Casing H In.	Casing W In.	Casing L In.
25E†	208	2600	62	32	89
33E†	260	3200	62	32	89
50‡	330	4000	66	34	81¾
50†	400	4770	67½	38	87
63‡	500	5900	77	44	73
80‡	640	7650	77	44	73
1000‡	800	8850	77	54	73
1500	1200	14300	103¾	67	93½
2000	1600	19100	103¾	80	93½
2500	2000	23900	110	71	126
3200	2560	30600	110	90	154
4750	3800	45400	110	109	128

* Oil-fired, prefix OL, gas-fired-prefix GA; dual fuel-prefix OG
† Blower integral part of furnace cabinet.
‡ Blower cabinet separate. Dimensions as follows: Model 50—53"H, 53"W, 42"L; Model 63—53"H, 53"W, 42"L; Model 80—59"H, 59"W, 59¾"L; Model 1000—55¾"H, 65"W, 81¾"L.

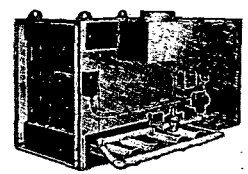


J-C OIL-FIRED SUSPENDED FURNACES

Economical and trouble-free operation of Jackson & Church *Suspended Oil-Fired Furnaces* is assured. Compactly designed with features that architects, engineers, and builders prefer. Units have stainless steel combustion chamber with over-size air passages for increased heating surfaces. Cabinets are heavy-gage, *rust-resistant steel.*

Model No.	Output 1000 Btu	CFM Bonnet	H In.	W In.	L In.
OL-125-S10	100	1250	23	26½	56
OL-180-S15	123 to 153	1890	23	26½	68
OL-250-S22	200 to 224	2530	29	35	80
OL-350-S28	250 to 280	3540	29	35	80
OL-560-S39	325 to 350	4960	35½	35	80
OL-560-S45	400 to 450	5680	35½	35	80
OL-1250-S80	800	10000	52½	55	116¾
OL-1250-S100	1000	12500	53½	55	116¾
OL-220-CF17*	175	2220	73½	27	30

* Counterflow Model.



J-C GAS-FIRED FURNACES

Each size comes in three different styles—as a horizontal furnace (GS) for suspended or floor mounting, as a blower-type unit heater (GU) for suspended or floor mounting, and as a duct furnace (GDF) for quick adaptation to year-round air conditioning. All *A.G.A.* approved.

	Output 1000 Btu	CFM Bonnet	H In.	W In.	L In.
GS-20	160	2000	34¼	26¾	68¾
GU-20	160	2125	34¼	26¾	68¾
GDF-20	160		34¼	26¾	46¾
GS-24	192	2430	34¼	26¾	68¾
GU-24	192	2560	34¼	26¾	68¾
GDF-24	192		34¼	26¾	46¾
GS-28	224	2830	34¼	26¾	68¾
GU-28	224	2990	34¼	26¾	68¾
GDF-28	224		34¼	26¾	46¾
GA-220-CF17†	175	2220	73½	27	30

† Counterflow Model.

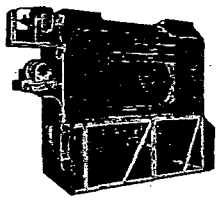
JOHNSON HEATER CORP.

1 Winnisimmet St., Chelsea, Mass.

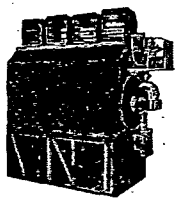
DESIGNERS — **MANUFACTURERS**

JOHNSON MODEL FA FACTORY ASSEMBLED "FYRE TUBE" HEATING UNITS

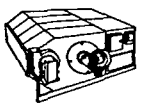
Engineered and designed for all types of commercial and industrial direct-fired heating applications, with or without duct work. Single heater output capacities from 400,000 to 4,000,000 Btu/hr. Fired by light or heavy oil, gas or combination of both. Furnished as standard with *UL* and *FM* accepted burner and control systems, chrome nickel stainless steel combustion chamber, large capacity circulating blowers and separate motor driven induced-draft fan.



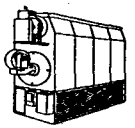
Cutaway view of Johnson Model FA Unit (Type VP). Note the rugged compact construction with accessible service and maintenance features.



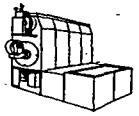
TYPE VN Vertical floor model with adjustable discharge heads.



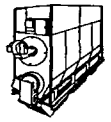
TYPE HP Horizontal suspended type for duct connections.
TYPE HN Horizontal suspended type with discharge nozzles.



TYPE VP Vertical floor mounted units for duct installations.



TYPE VPO Using separate outside blower for combination heating, cooling or ventilating applications. Also furnished for horizontal mounting TYPE HPO.



TYPE IN Inverted type using adjustable discharge nozzles. Furnished without nozzles for duct installations TYPE IP.

HEATER MODEL	BTU/HR. OUTPUT AT PLENUM	NOMINAL CFM AT OUTLET	BLOWER MOTOR HP	PHYSICAL DIMENSIONS Width	Length	Height	Length with I. D. Fan	Height with Nozzles	APPROX. SHIPPING WEIGHT
FA-400	400,000	5,600	1	36	66	76	88	94	2175
FA-500	500,000	6,800	1½	36	66	76	88	94	2250
FA-600	600,000	7,500	2	36	66	76	88	94	2250
FA-650	650,000	8,500	1½	42	80	88	102	106	2425
FA-750	750,000	9,800	2	42	80	88	102	106	2450
FA-850	850,000	11,000	3	42	80	88	102	106	2500
FA-1000	1,000,000	13,000	3	48	91	90	115	110	3150
FA-1150	1,150,000	14,700	5	48	91	90	115	110	3200
FA-1250	1,250,000	16,500	5	54	106	106	133	129	4300
FA-1500	1,500,000	21,500	7½	54	106	106	133	129	4700
FA-1800	1,800,000	23,700	7½	60	126	106	151	125	5770
FA-2000	2,000,000	28,800	10	60	126	106	151	125	5800
FA-2500	2,500,000	33,500	10	66	141	120	171	149	7050
FA-3000	3,000,000	39,000	15	66	141	120	171	149	7250
FA-3500	3,500,000	46,000	15	72	157	133	190	166	9250
FA-4000	4,000,000	53,500	20	72	157	133	190	166	9450

The Johnson Heater Corp. Reserves The Right To Change Any Dimensions Contained Herein Without Notice.

JOHNSON HEATER CORP.

1 Winnisimmet St., Chelsea, Mass.

DESIGNERS — **MANUFACTURERS**

MODEL JA HEAVY DUTY COMMERCIAL AND INDUSTRIAL "AIRE TUBE" HEATERS

15 Sizes—Output Capacities from 400,000 to 5,000,000 Btu/hr. Direct-Fired by heavy or light oil, gas or a combination of both fuels

"AIR ROTATION" SPACE HEATING

One, free standing, Model JA "AIRE TUBE" Heater can uniformly heat in excess of 1,500,000 cu ft of open building space without duct work, noticeable drafts or the need to spot, throw or direct the heated air.

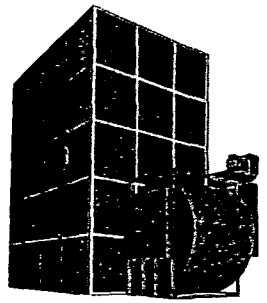
This is accomplished with our "AIR ROTATION" system of space heating in combination with our special heater design. Large volumes of the cooler and denser, low-level air from the building are rotated through the heater. This air is then heated to a low temperature and discharged at a low velocity from the heater outlets back into the area to be heated.

DUCT DISTRIBUTION SYSTEM

JA "AIRE TUBE" Heaters with low internal resistance are excellent for use with all types of duct distribution systems.

HEAVY OIL FIRING

JA Heaters are especially designed for firing with the low cost #5 or #6 fuel oils in the larger sizes.



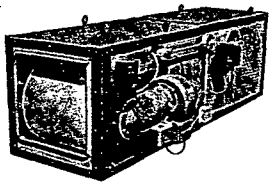
JA Heaters have approximately 90 percent direct heating surface with multiple banks of practically self-cleaning, air-heating tubes located over the fire.

MODEL JAS

Horizontal Suspended "AIRE TUBE" Heating Units. 17 Sizes Output Capacities from 112,000 to 1,000,000 Btu/hr.

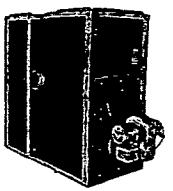
- Factory assembled
- Multiple air tube design
- Practically self cleaning
- Low draft requirements
- Chrome nickel steel combustion chamber

Available as duct heaters without blower section in range of sizes listed above.



Direct-fired by light oil, gas or a combination of both fuels

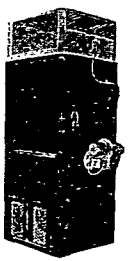
SMALL MODEL JA



(EXTENDED CASING TYPE SHOWN)
Nine sizes output capacities from 112,000-350,000 Btu/hr. Fired by light oil, gas or a combination of both.

MODEL HB

For heating small commercial-industrial and domestic buildings
Designed for use as a space heater or with duct distribution systems



(SPACE HEATER SHOWN)
Nine sizes output capacities from 112,000-350,000 Btu/hr. Fired by light oil, gas or a combination of both.

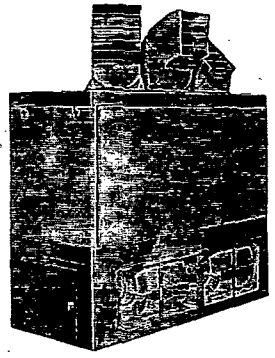
LENNOX Industries Inc.

Manufacturers of Warm Air Heating Equipment

1701 EAST EUCLID, DES MOINES 4, IOWA

Representatives In All Principal Cities

FOR | SPACE HEATING, DRYING, MAKE-UP AIR AND OVEN APPLICATIONS



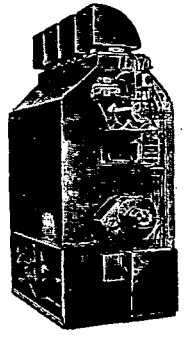
"OG" SERIES HEAVY DUTY HEATERS

Specifications—Gas, oil or combination burners with manual or automatic fuel selection. 300,000 to 500,000 Btuh output. Control system is packaged, factory-tested.

Applications—Can be used as a central furnace with ducts or as a unit heater. Installations may be horizontal, vertical, suspended or floor mounted. Listed by *U. L.* and complies with factory mutual insurance requirements.

Blower Section—Huge air volume delivered quietly from dynamically balanced blowers. Up to 2 in. external static pressure. Blower section can be used for air conditioning, will handle up to 30 tons of cooling. A wide choice of blower motors and drives satisfies all air volume requirements.

Construction—Compact, rugged, with 16 gage stainless steel heat exchanger; secondary heat surface of aluminized steel to resist corrosion. Exceptional expansion joint allows unit to "breathe." Burner enclosed in sound proof compartment. Heater uses adjustable discharge nozzles or ducts. (*Write for Bulletin OG-581-M10.*)



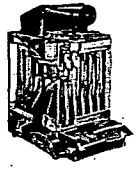
"AT" SERIES HEAVY DUTY HEATERS

Specifications—Gas, oil or combination burners with manual or automatic fuel selection. 750,000 to 2,000,000 Btuh output. One master controller eliminates non-standard, hard-to-replace system components.

Design Features—Special 4-pass counter flow design without the use of internal baffles gets the most heat from the flame. Boiler tubes, equipped with high-velocity retarders, and double casing, with metal radiation shield, contribute to operating efficiency of 80 percent or better, while holding casing heat loss to minimum.

Blower Section—Quiet, large-capacity blowers may be removed for remote applications if desired.

Applications—"AT" heaters offer wide variety of uses and applications in department stores, warehouses, manufacturing plants, process industries, truck terminals, apartment houses, office buildings. Units can be floor mounted in upright, inverted, or horizontal positions; suspended vertically or horizontally from ceiling (service platform optional). (*Write for Bulletin LDMI-G-5-58.*)

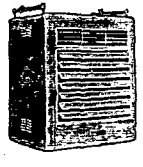


GAS DUCT HEATERS

Wide range of uses: Hotels, motels, stores, schools, service stations, factories. Btuh input ratings per unit from 120,000 to 420,000. Depth is never more than 49 in. including diverter.

Dual limit controls provide complete protection for air travel in either direction. Can be fired with all types of gaseous fuels.

Provides positive zone control with precise thermostatic action. Can be used alone or added to an existing system. (*Write for Bulletin GSD-581-M3.*)



GAS UNIT HEATERS

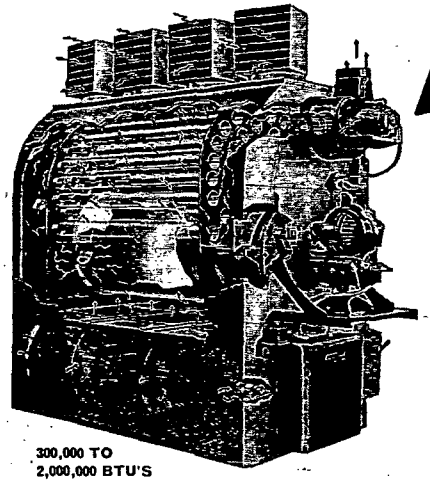
Btuh input ratings per unit from 80,000 to 250,000. Dual vertical and horizontal louvers diffuse warm air evenly over a wide area. Quick, easy installation made possible by easily balanced hangers.

Motor has overload protection, is cushion mounted and totally enclosed. Attractive cabinet design allows "exposed" installations. Controls include automatic pilot with 100 percent safety shut-off.

Heat exchanger: Aluminized steel, with 18 gage tubes and 16 gage headers. (*Write for Bulletin LDMI-G-6-58.*)

MAMMOTH FURNACE COMPANY

6425 CAMBRIDGE STREET • MINNEAPOLIS 26, MINNESOTA



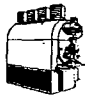
300,000 TO 2,000,000 BTU'S

Commercial-Aire

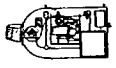
DIRECT-FIRED SPACE HEATERS

Commercial-Aire Direct-Fired Space Heaters have a high fuel-to-heat conversion efficiency. They are ruggedly built for quiet, vibrationless operation. All controls, wiring, and moving parts are accessible for service. Among the many desirable features are the central control cabinet, retractable burner tray, MH-Electronic flame control, pre-ignition draft purge timing and draft sensing, and vortex cooled motor shaft to blower wheel. A variety of controls, power supplies and burners by reputable manufacturers may be specified and integrated. Write for Bulletin C-357.

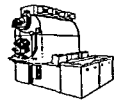
SAFETY TESTED and approved by A.G.A. Laboratories and Listed by Underwriters' Laboratories, Inc. Commercial-Aire safety control circuit is accepted by Factory Mutual Engineering Division and Factory Insurance Ass'n. Control circuit wiring meets J.I.C. Requirements.



MODEL "F" Floor mount with discharge heads for industrial plants, foundries, etc.



MODEL "H" Horizontally suspended unit for use where floor space is unavailable.



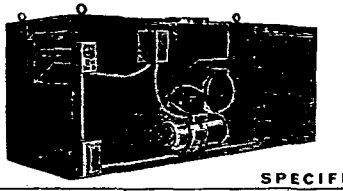
MODEL "FD" For duct distribution. For churches, schools, auditoriums, etc.



MODEL "U" Suspended inverted unit for make-up air applications in plants, foundries, etc.

SPECIFICATIONS

Model	Btu Output	Sq. ft. Htg. Surface E.D.T.	Cfm @ ½" EXT. S.P. 85° RISE	Blower DIDW	Motor hp	Approx. Fan rpm	Max. Firing Rate Oil gph	Max. Firing Rate Gas cfh	Length	Height	Width	Plenum Opening	Mech. Exhaust Discharge	Approx. Shipping Weight
300	300,000	135.3	4,000	14"-twin	¾	575	3.0	375	60"	80"	32"	18x60	8"dia.	1,590
400	400,000	135.3	5,500	14"-twin	1	640	4.0	500	60"	80"	32"	18x60	8"dia.	1,690
550	550,000	183.6	7,000	14"-triple	1½	750	5.0	688	80"	80"	32"	18x80	8"dia.	1,950
700	700,000	183.6	8,700	14"-triple	2	820	7.0	875	80"	80"	32"	18x80	8"dia.	2,150
800	800,000	276.8	10,000	16"-triple	3	710	7.5	1,000	80"	83"	48"	22x80	10"dia.	2,775
1000	1,000,000	276.8	13,000	16"-triple	5	790	9.5	1,250	80"	83"	48"	22x80	10"dia.	2,850
1250	1,250,000	431.0	16,000	18"-triple	5	755	11.5	1,563	100"	96"	54"	26x100	10"dia.	3,850
1500	1,500,000	431.0	19,000	18"-triple	7½	830	14.0	1,875	100"	96"	54"	26x100	10"dia.	4,010
1750	1,750,000	574.1	23,000	21"-triple	7½	680	16.5	2,188	120"	96"	60"	30x120	12"dia.	4,750
2000	2,000,000	574.1	26,000	21"-triple	10	740	18.5	2,500	120"	96"	60"	30x120	12"dia.	4,910



COMPACTO HORIZONTAL SUSPENDED HEATERS

Require no floor space. Easily suspended from rafters or installed in loft or attic with or without ducts. Available as complete wired package ready for fuel line hook-up or with burner and controls unwired and unassembled. Has rugged 10 and 11 gauge welded plate steel heat exchanger with internal stainless steel combustion chamber for trouble-free performance. Oil or gas fired. Write for Bulletin 57S-1.

SPECIFICATIONS

MODEL	B.T.U. INPUT	B.T.U. OUTPUT	SQ. FT. HEATING SURFACE	C.F.M. @ ¾" S.P. 70° AIR	C.F.M. @ ½" S.P. 85° RISE	MOTOR H.P.	BLOWER SIZE	APPROX. BLOWER R.P.M.	FIRING RATES OIL G.P.H.	FIRING RATES GAS C.F.H.	APPROX. SHIPPING WEIGHT
185	232,000	185,000	55.4	2130	2460	⅓	14"	509	1.75	232	695
225	280,000	225,000	55.4	2580	2980	½	14"	539	2.00	280	725
260	326,000	260,000	78.2	3000	3465	½	16"	525	2.50	326	890
300	375,000	300,000	78.2	3450	3985	¾	16"	565	2.75	375	935
350	450,000	350,000	130.3	4030	4655	1	18"	575	3.25	450	1010
400	500,000	400,000	130.3	4600	5313	1½	18"	600	3.50	500	1040
450	560,000	450,000	130.3	5200	6000	1½	18"	650	4.00	560	1055

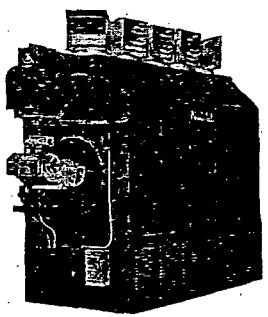
National Heater Company, Inc.

2475 Doswell Avenue

St. Paul 4, Minnesota

Sales representatives in principal cities in U. S. and Canada

NATIONAL Champion WARM AIR SPACE HEATERS



MODEL H—Standard upright (floor set) model. Also MODEL D—without discharge heads for duct installations.

TD Series

- 6 models—66 sizes!
- Gas, oil or gas & oil fired!
- Stainless steel combustion chamber!
- 80 percent operating efficiency!
- Basic models *UL* or *A.G.A.* approved!
- Low initial cost!
- Easy, quick installation!
- Low operating cost!
- Minimum maintenance!
- Floor and suspended models!
- No full time operator required!
- Completely automatic—thermostatically operated!
- Factory flame-tested!
- A complete "packaged" heater requiring only fuel line, power and stack hook-up!

WRITE FOR 1959 CATALOG



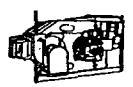
MODEL S—For Churches, Schools, Public Buildings and institutions.



MODEL R—Inverted suspended unit. For make-up air applications.



MODEL P—Selfcontained portable unit.



MODEL C—Horizontally suspended unit. Ideal where floor space is limited.

Heater Number	Btu* Output	Standard† CFM	Blowers DWDI	Fan Motor HP	Supply Fan RPM	Plenum Opening	Approx. Shipping Weight
TD-40	400,000	5,400	15"-Twin	1	560	18 x 60	1400
TD-50	500,000	6,600	15"-Twin	1½	595	18 x 60	1500
TD-55	550,000	7,100	15"-Twin	2	645	18 x 60	1550
TD-65	650,000	7,600	16"-Twin	2	670	18 x 80	1750
TD-75	750,000	8,800	16"-Twin	2	695	18 x 80	1900
TD-80	800,000	9,600	16"-Twin	3	720	18 x 80	1900
TD-85	850,000	10,500	18"-Twin	3	590	18 x 80	2400
TD-100	1,000,000	12,500	18"-Twin	5	680	18 x 80	2600
TD-125	1,250,000	15,300	21"-Twin	5	640	24 x 100	3200
TD-150	1,500,000	19,400	21"-Twin	7½	730	24 x 100	3450
TD-175	1,750,000	22,000	21"-Triple	7½	560	30 x 120	4150
TD-200	2,000,000	26,000	21"-Triple	10	590	30 x 120	4300
TD-250	2,500,000	32,000	24"-Triple	10	600	30 x 160	6300
TD-300	3,000,000	38,000	24"-Triple	15	630	30 x 160	6630
TD-350	3,500,000	46,000	24"-Triple	20	670	30 x 160	6720

* Tested and rated in accordance with the code adopted in 1956 by ASHAE for Heavy Duty Furnaces & Direct Fired Unit Heaters.

† Higher CFM ratings available on application.

Specification data subject to change without notice.

U SERIES HI-BOY and COUNTER-FLO MODELS

NATIONAL Champion WARM AIR SPACE HEATERS

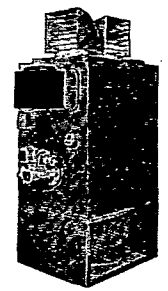
T-F SERIES SUSPENDED UNITS

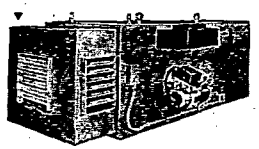
U Series HI-BOY and COUNTERFLO models. ▸

Research-designed for compactness plus efficiency are prime requirements. The **HI-BOY** upright units can be equipped with adjustable discharge heads or arranged for duct connections. **COUNTER-FLO** models for under floor heat distribution systems or inverted ceiling suspension. Five sizes—175,000 to 350,000 Btu output capacity for either gas or oil firing.

•

T-F Series suspended units. Heavy duty suspended units engineered for industrial and commercial heating applications where floor space is at a premium. Rugged, compact, easy to install—with four-way adjustable





discharge head or for convenient integration into a duct distribution system. Duct heaters available in identical Btu capacities. Special stainless steel and aluminized steel can be furnished for high temperature applications and corrosive conditions. Nine sizes—180,000 to 550,000 Btu output capacities—gas or oil fired.

T-F SERIES SUSPENDED UNITS

Model	Output (BTU/Hr)*	Fan CFM Capacity	Avail. SP	Fan Motor HP	Dimension in inches			Shipping Weight
					L	W	H	
TF-18	180,000	2,050	⅜	¼	73	31	28	575
TF-20	200,000	2,300	⅜	½	73	31	28	630
TF-23	230,000	2,800	⅜	½	77	36	32	690
TF-25	250,000	3,000	⅜	½	77	36	32	720
TF-30	300,000	3,700	⅜	¾	77	36	32	760
TF-35	350,000	4,500	⅜	¾	92	48	42	1100
TF-40	400,000	5,200	½	1	92	48	42	1160
TF-45	450,000	5,800	⅝	1¼	99	48	48	1750
TF-55	550,000	6,400	⅝	2	99	48	48	1850

* Tested and rated in accordance with the code adopted in 1955 by ASHAE for Heavy Duty Furnaces and Direct Fired Unit Heaters.

U SERIES—HI-BOY and COUNTER-FLO MODELS

Model	Output BTU/Hr*	Fan CFM Capacity	Fan Motor HP	Approx. Fan RPM	Dimensions in inches			Shipping Weight
					L	W	H	
U-175	175,000	2,000	¼	520	38	35	78	720
U-200	200,000	2,300	⅓	560	38	35	78	740
U-250	250,000	3,000	½	440	42	35	78	900
U-300	300,000	3,700	¾	543	42	35	78	940
U-350	350,000	4,400	1	500	42	35	83	1,060

* Tested and rated in accordance with the code adopted in 1955 by ASHAE for Heavy Duty Furnaces and Direct Fired Unit Heaters.

Specification Data Subject to change without Notice.

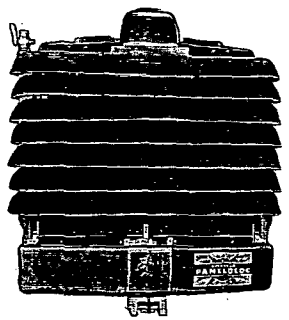
WRITE FOR 1959 CATALOG

THERMOBLOC DIVISION • PRAT-DANIEL CORP.

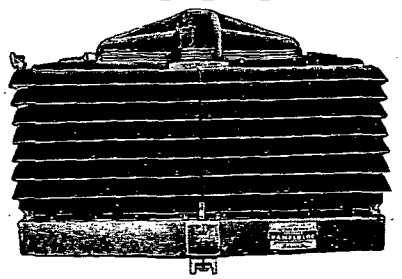
2 Meadow Street, South Norwalk, Conn.

Canadian Affiliate: Canadian Wanson Combustion Equipment Ltd., Montreal

PANELBLOC



CR-63 (62,500 Btu Input)



CR-125 (125,000 Btu Input)

PANELBLOC INFRA-RED RADIANT HEATERS BRING THE SUN INDOORS

For Commercial and Industrial Installations

Through the eons man has benefited from radiant heat. First the sun and now the gas fired Overhead PANELBLOC takes over. With the speed of light healthful infra-red heat waves pleasantly warm objects and people without air disturbance.

A few of PANELBLOC'S advantages are:

- No fan—ideal for dusty areas.
- No motor—quiet as a sun's ray.
- No power consumption—unit generates own operating current.
- No costly electrical power installation charges.
- No possibility of shutdown due to power failure.
- No heat diverted by drafts from open doors or in open pavilions.
- No fuel wasted heating unnecessary spaces in building.

MODERN INFRA-RED METHOD

(U. S. and Foreign Patents Granted)

Developed to fill the gaps in Industrial and Commercial heating, PANELBLOC turns difficult heating problems into routine installations. PANELBLOC spot-heats with no need for costly partitions or false ceilings. Infra-red heat like light goes only where directed by the polished metal louvers. As a result people, equipment and particularly floors are pleasantly warm.

CONSTRUCTION AND OPERATION

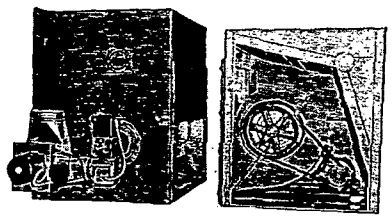
The combustion gases, produced from any commercial gas fuel pass through a unique panel heat exchanger of special steel. To overcome corrosion and high temperatures this heat exchanger is coated inside and out with vitreous enamel. The panel is heated to a carefully controlled temperature so that the infra-red waves (10 times longer than those from a glowing fire) produced have a wave length which gives maximum penetration. Maximum penetration means comfort heating.

Underwriters Laboratories Inc. INSPECTED

TJERNLUND MANUFACTURING COMPANY

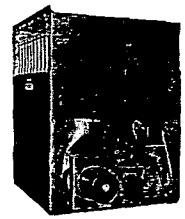
2140 KASOTA AVENUE · ST. PAUL 14, MINNESOTA

Manufacturers of Industrial and Commercial Warm Air Heating Equipment, Oil and Gas Burners, Blowers and Draft Inducers



STYLE "A" LO-BOY

Heater with separate blower section for by-pass application blowers other than standard can be utilized i.e. model 150 blower can be used with model 100 furnace for additional air capacity.



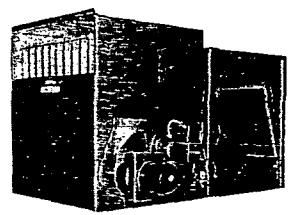
STYLE "A" DUCT HEATER

All heaters available as duct heaters in a variety of outlets and inlets for use in conjunction with air conditioning equipment.



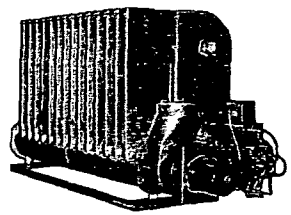
STYLE "E" COUNTER FLOW HEATER

With jet fired power gas burner top or side inlet for return air.



STYLE "A" SUSPENDED OR LO-BOY

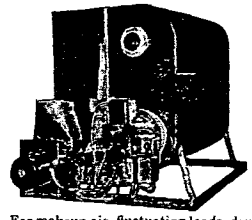
Heater filter inlet top or rear. Discharge top or front.



EVEN-FLOW HEAT EXCHANGER

Multiple flue arrangement. Down flow of hot gasses gives a temperature equalizing effect which distributes the generated heat uniformly over the entire flue area. Direct firing into combustion area with no restrictions to block heat transfer to air stream. Stainless steel wrapper at combustion area.

Hi-Boy, Counter Flow, Suspended, Lo-Boy Heaters in a Complete Range of Sizes—Choice of Firing Equipment.



3 STAGE OIL BURNER

For makeup air, fluctuating loads, drying systems, heating and ventilating systems. Three individual burner heads, each comprising end cone, turbulator ring, nozzle and blast tube make possible efficient air velocities and oil atomizing pressures for each firing stage. Integral draft inducing fan provides positive draft. Single motor for operation of both oil burner and draft inducer. Standard SUNDSTRAND 2 STAGE fuel unit. Delayed action solenoid valves to each burner head provides prepurge timing, control of oil for each firing stage and positive shut-off. Electronic control for fast, safe control. Standard ignition transformer.

ENGINEERING DATA

Model	BTU Output	CFM ½" S.P.	Input Oil GPH	Nat. Gas CFH	Blower H.P.	Phase Volt	RPM	No. Wheels	Wheel Size	Filters Quan.	Filters Size
35	134,000	1600	1:20	160	¼	1/110	680	1	12 × 12	2	16 × 25
50	184,000	2200	1:65	220	½	1/110	560	1	14 × 13	4	16 × 20
70	234,000	2800	2:00	280	½	1/110	600	1	14 × 14	4	20 × 20
100	338,000	4200	2:90	400	¾	1/220	520	1	16 × 16	4	20 × 25
150	500,000	6300	4:50	600	1	3/220	500	2	16 × 14	6	20 × 25
200	670,000	8400	6:00	800	1½	3/220	450	2	16 × 16	6	16 × 25
280	898,000	10000	8:00	1000	2	3/220	480	2	16 × 16	6	20 × 25
340	1,125,000	12000	10:00	1400	2	3/220	490	2	20 × 16	12	20 × 25

WRITE for COMPLETE CATALOG and SPECIFICATIONS

American Air Filter Co. Inc.

Louisville, Kentucky

Plants:
Louisville, Ky.
Chicago, Ill.
Moline, Ill.
Morrison, Ill.
St. Louis, Mo.

HEATING EQUIPMENT



HORIZONTAL SHAFT PROPELLER-FAN TYPE HEATERS

For suspended mounting. Use with steam or hot water. Copper heating element with patented stay tube prolongs unit life. Many models and sizes. *Bulletin 700.*



VERTICAL SHAFT PROPELLER-FAN TYPE UNIT HEATERS

Designed for high ceiling installations. Long life copper heating element for use with either steam or hot water. Units available in range of capacities with high or low velocity discharge. *Bulletin 700.*

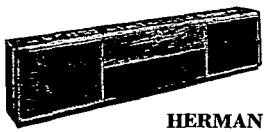
CONSOLE HEATERS

Compact, quiet, economical and attractively styled. Ideal for offices, showrooms, corridors, stores, etc. May be placed on floor, wall or suspended from ceiling. 18 models and sizes. *Bulletin 787.*



CONVECTOR RADIATORS (Not Illustrated)

Range of sizes for steam or hot water. Heating elements made of non-ferrous materials. Cabinets designed for free standing or recessed installation. Also sloping top models. *Bulletin 791.*



FINNED RADIATION AND COVERS (Not Illustrated)

Adaptable to any system as a primary or supplementary heating source. Range of sizes with choice of expanded metal, flat top or slant top covers. *Bulletin 790.*

HERMAN NELSON UNIT VENTILATOR SYSTEMS

Herman Nelson Draft/Stop Systems automatically heat, ventilate, natural cool (with outdoor air) and mechanically cool (with chilled water). Draft/Stop principle traps cold window drafts before they reach classroom occupants. For steam, hot water, electric heating, direct gas-firing, and hot water or chilled water for year 'round air conditioning. *Bulletin 600.*

FANS AND BLOWERS



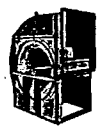
PROPELLER FANS

A complete line of belt or direct drive ventilating fans. Twenty-nine sizes with wheel diameters from 10 to 54 in. Unit capacities ranging from 655 to 36,150 cfm. *Bulletin 800.*



UNIT BLOWERS

Direct or belt drive with forwardly curved or backwardly inclined wheels. Wheel diameters from 4½ in. to 36 in. and capacities from 160 to 18,303 cfm. *Bulletin 826.*



CENTRIFUGAL FANS

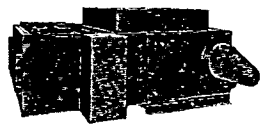
Designed for systems requiring a Class I or II centrifugal fan. Slow speed or non-overloading type. Wheel diameters from 12¼ in. to 73 in., single or double width. Direct or belt drive. Any rotation or discharge. *Bulletin 851.*

American Air Filter Co. Inc.

Louisville, Kentucky

Plants:
Louisville, Ky.
Chicago, Ill.
Moline, Ill.
Morrison, Ill.
St. Louis, Mo.

AIR CONDITIONING UNITS

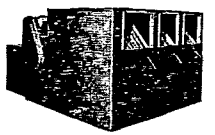


Complete Air Treatment

All basic air conditioning functions of heating, humidifying, cooling, dehumidifying, cleaning, and air movement may be combined in this one unit, providing true "year 'round" air conditioning.

A Size for Every Application

Twelve sizes, with air handling capacities of 675 cfm to 36,000 cfm matched with proper size cooling and heating coils of all styles and capacities make this a universal unit. No application is too small, none too large for this broad line.

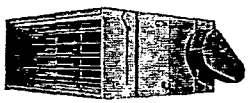


A Model to Fit

Horizontal and Vertical models are available in all sizes to meet the modern need for compact, space-saving equipment. *Bulletin 786.*

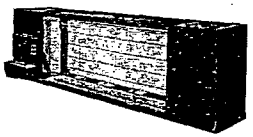
MULTI-ZONE AIR CONDITIONING UNITS

Multi-Zone Air Conditioning Units maintain design conditions, under varying loads, in ALL zones of the system. This single central station Multi-Zone Unit can handle from 2 to 8 zones. Eight sizes, with air handling capacities of 3,440 cfm to 36,000 cfm. Coils can be furnished for direct expansion, chilled water, steam or hot water. One cooling coil and one heating coil in each unit (operating in parallel) supply correctly cooled, tempered or dehumidified air for perfect comfort under year-round conditions. *Bulletin 870.*



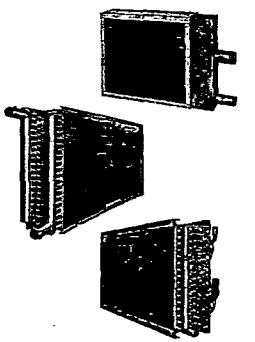
HEATING AND VENTILATING UNITS

Feature sectionalized construction, low outlet velocities, slow fan speeds, wide selection of heating coils. Available with humidifier. Ideal for institutional, industrial and commercial use. *Bulletin 910.*



ROLL-O-VENT AUTOMATIC AIR FILTER

Offered exclusively by AAF, the Roll-O-Vent automatic air filter is available as an optional accessory with Air Conditioning Units, Multi-Zone Units, or Heating and Ventilating Units. Utilizing AAF's famous Roll-O-Matic filtering principle, Roll-O-Vent assures constant air delivery at all times, makes filter changing automatic and gives major savings in maintenance. One roll of media can do the work of a year's supply of disposable filters at about one-half the cost.



HEATING AND COOLING COILS

A. **Standard Steam Coils—Type S** (Not Illustrated) Made with ⅝ in. seamless copper tubing; spaced 8 fins per inch. Recommended for a maximum operating pressure up to 30 psi. Over 30 psi, use heavy duty coils, type HS.

B. **Steam Distributing Coils—Type SD** For use where steam supply is to be modulated and/or exposed to freezing temperatures. *Bulletin 890.*

C. **Standard Water Coils—Type W** Made with ⅝ in. seamless copper tubing, spaced on 1½ in. centers in depth and width. Fins spaced 8 per in. Coils are used for both heating and cooling. Also available with fittings to permit internal cleaning of tubes. *Bulletin 880.*

D. **Direct Expansion Coils—Type DE** Aluminum or copper fins are bonded to ⅝ in. or ¾ in. seamless copper tubing. Fins spaced 8 per in. Refrigerant circuits are of equal length and arranged for counter flow. *Bulletin 880.*

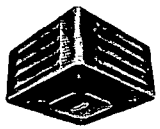
Carrier Corporation

Syracuse 1, New York

UNIT HEATERS

Carrier Unit Heater Agents, jobbers and distributors are located in all principal cities. Listed in your local Classified Telephone Directory under "Heaters, Unit"

Carrier Unit Heaters are the product of more than half a century spent in the design and manufacture of temperature control equipment. For high and low pressure steam, forced hot water and gas-fired systems.



46S Four-Way Unit Heaters feature a variable "clover leaf" air discharge pattern through four separately controlled outlets. In eight sizes ranging from 55,000 to 600,000 Btu/hr.



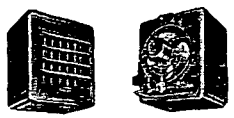
46U Horizontal Unit Heaters heat areas with a conventional horizontal discharge pattern. Feature single row, low air resistance coils; in ten sizes ranging from 13,400 to 200,000 Btu/hr.



46SD Vertical Unit Heaters mount within 2 inches of ceiling to save headroom. Available with standard or low temperature rise coils in ten capacities, from 30,000 to 576,000 Btu/hr output.



46AE Cabinet Unit Heaters include eight cabinet styles; free standing, variable semi-recessed and recessed. Mount on floor, wall or ceiling. Capacities range from 26,200 to 115,000 Btu/hr.



46T Gas-fired Unit Heaters are compact in design and use Aluminized Steel heat exchangers for long unit life. Available in eight sizes ranging in capacity from 50,000 to 230,000 Btu/hr.

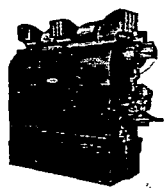


46TD Gas-fired Duct Furnaces include eleven basic sizes from 50,000 to 440,000 Btu/hr capacity. Companion engineered Blower Assemblies provide from 530 to 6500 cfm capacity.

Carrier Heat Diffusers are designed to provide dependable heating and ventilating for large buildings or areas and for industrial process application. Available for steam, hot water or direct-fired installation.



46PQRV Heat Diffusers for steam and hot water systems feature floor mounted or suspended units with "building block" design of components for installation flexibility. Choice of fan-coil capacities to provide from 47,000 to 2,990,000 Btu/hr output.



46AF Heat Diffusers—Direct Fired are self-contained units for oil, gas or combination fuel. With stainless steel combustion chamber and heavy gage steel heat exchanger, they are *A.G.A.* and *UL* approved in ten sizes from 300,000 to 2,000,000 Btu/hr.

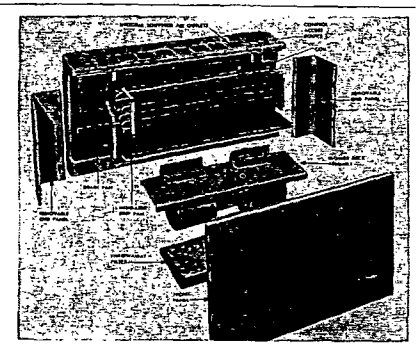
FEDDERS CORPORATION

HEATING DIVISION • LALOR & HANCOCK ST. • TRENTON 7, N. J.

Representatives in Principal Cities



Showing compact installation of floor unit

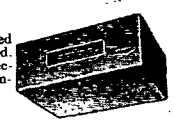


Exploded view showing accessibility of parts

FEDAIR REMOTE TYPE AIR CONDITIONING UNITS

Made in Free Standing, Recessed, and Overhead Models

FEDAIR Units are designed for multiple room installations, in large and small buildings. They simplify planning of new and modernized buildings. No duct work and only 3 lines required—supply, return, and condensate drain. Aluminum finned copper tube coil gives quick response to controls. All models available with motors for standard and low-speed operation. Blowers are sized to deliver rated cfm at specified motor speeds. Outside and recirculated air filtered. Write for complete catalog.



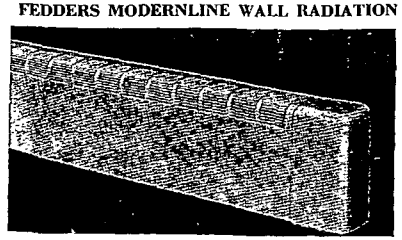
FEDAIR overhead models are used where floor and wall space is limited. Outlet louvers are adjustable for directional control. Available without enclosure for concealed installations.

FEDAIR BASIC CAPACITIES—ALL MODELS

MODEL	AIR FLOW QUANTITY CFM AT 70° F				*COOLING CAPACITY HIGH SPEED BTU/Hr 45° F. ENT. WATER		WATER FLOW		†HEATING CAPACITY HIGH SPEED 180° F. ENT. WATER	
	High	Medium	Low	Max. Fresh	Total	Sensible	GPM	Ft. H2O Pressure Drop	Btu/Hr	Temp. Drop
HC21 LHC21	215	165	135	65	6,710	4,830	1.7	1.9	13,700	22.9
HC31 LHC31	310	250	210	85	10,750	7,800	2.7	5.1	27,500	20.4
HC43 LHC43	430	335	275	110	14,500	10,000	3.6	10.0	37,700	21.6
HC60 HLC60	605	490	410	160	19,100	13,500	4.8	7.1	53,100	23.4

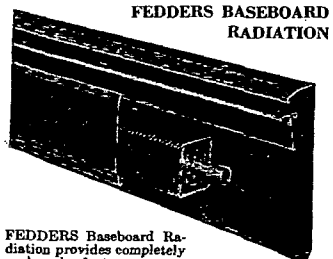
* Cooling Capacity Based on 80°F. DB, 67°F. WB Entering Air Mixture with 8°F. Water Temperature Rise.
† Heating Capacity Based on 60°F. Entering Air Temperature and indicated water flow.

FEDDERS MODERNLINE WALL RADIATION



SMOOTH ROUNDED COVERS, available in 3 heights and 2 depths, combine attractive appearance with functional convected and radiated heat emission. Integral die-formed louvers direct heated air outward into room. Covers finished with baked-on ivory enamel or Quaker gray neutral prime. Can be furnished in five other colors. HEATING ELEMENTS can be used with steam or hot water systems and installed one or two rows high. ML-5 elements have 1¼ in. copper tubes with 4¼ in. square .025 in. aluminum fins in lengths from 2 ft. to 12 ft. in increments of 6 in. ML-5 steel elements available with 1¼ in. and 2 in. IPS and 4¼ in. square fins. Write for catalog.

FEDDERS BASEBOARD RADIATION



MODEL BBH Complete With Deflector Damper

FEDDERS Baseboard Radiation provides completely packaged, factory assembled units for easy handling and time saving installation. Standard lengths are 4-ft, 6-ft, and 8-ft with other lengths available for large projects. Three series—Model BBH, Model S, and Model L provide ratings and prices to fit every job. Write for catalog.

ALSO MANUFACTURERS OF HORIZONTAL, DOWNBLOW, AND CABINET UNIT HEATERS, CONVECTORS, AND CONTINUOUS CONVECTOR RADIATION



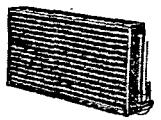
McQuay INC.

1602 Broadway, N. E. Minneapolis 13, Minn.

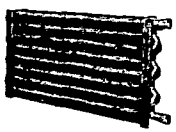




⅜ in. Tube Coil



½ in. Tube Coil



⅝ in. Tube Coil

McQUAY RIPPLE-FIN COILS

The famous McQuay Ripple-Fin coils are made in three tube diameters; ⅜ in., ½ in. and ⅝ in., in a wide variety of styles and sizes for steam, hot water, cold water, direct expansion and refrigerant condensing.

"Custom Engineered" coils are readily obtained, as McQuay's manufacturing methods provide a wide variety of fin spacings, fin heights, finned lengths, rows deep and circuiting, thereby both space and capacity requirements can be met.

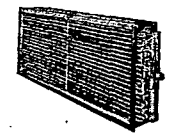
Ripple-Fin coils have all tubes mechanically pressure bonded into wide smooth fin collars providing a permanent bond and a high heat transfer efficiency. McQuay coils are pneumatically tested under warm water at 315 psig.

Ripple-Fin coils have a staggered tube pattern which, combined with the ripple fin, produces a turbulent air flow and reduces air "by-pass".

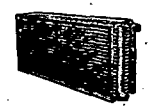
In addition to the ripples across the air flow, the fins of McQuay coils have rippled edges to give strength and a pleasing appearance.

McQuay evaporators and condensers are dried with minus 40°F dew point air to assure moisture-free coils.

McQuay Ripple Fin coils have been approved by *Underwriters' Laboratories* for use as evaporators or condensers with either R-12 or R-22.



Direct Expansion Coil



Water Coil



Steam Coil

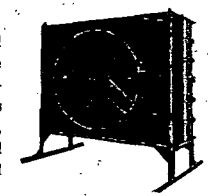
McQuay Means Quality



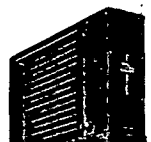
"AL" Aircon

McQUAY AIR COOLED CONDENSERS

Fifteen sizes of belt drive commercial propellers and centrifugal fan models up to 50 tons capacity in a single unit. Split circuiting systems and winter control systems are available for all models. Centrifugal fan models are available with filters, face and by-pass dampers, intake mixing boxes, discharge diverting boxes and humidifiers for combination condenser, ventilator and heating applications. Write for free bulletins.



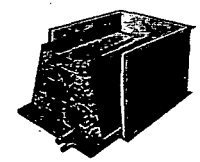
"AB" Aircon



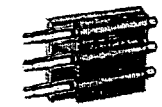
Vertical Air Flow Evaporator

McQUAY RESIDENTIAL EVAPORATORS

Five models of vertical residential evaporators ranging from 26,000 to 90,000 Btu/hr constructed for either up flow or down flow of air. The horizontal residential evaporators range from 25,300 to 92,100 Btu/hr in five models, and are designed for air flow in either direction. Both the vertical and horizontal circuits are for use with R-12 or R-22.



Horizontal Air Flow Evaporator



McQuay INC.

1602 Broadway, N. E. Minneapolis 13, Minn.





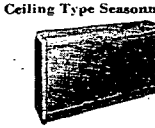
Horizontal Unit Heater



"RH" Air Conditioner



Ceiling Type Seasonmaker



Floor Type Seasonmaker



Vertical Heating and Ventilating Unit



Horizontal Multi-Zone Air Conditioner



Horizontal Type Seasonmaster

McQUAY UNIT HEATERS

Horizontal, down flow and cabinet unit heaters are available in an extremely wide range of sizes and capacities for either steam or hot water. Adjustable louvers are standard equipment on horizontal unit heaters. Down flow unit heaters are furnished with adjustable radial diffusers. There are three types of cabinet unit heaters—floor, wall and ceiling—all for exposed, semi recessed or fully recessed mounting.

SEASONMAKER

Air Conditioning Units

Individual room and large capacity air conditioners, four types—floor, hideaway, basic and ceiling. Cooling and heating medium supplied from central plant. Cools, dehumidifies and filters in summer. Heats and filters in winter. Five sizes of all models—200, 300, 400, 500 and 600 cfm. Hideaway and ceiling types also available with 800, 1200 and 2,000 cfm. The Seasonmaker's compactness, only 8½ in. deep, attractiveness, quiet operation and ease of installation make it ideal for hotels, apartment buildings, motels and other multi-room buildings.

McQuay Means Quality

HEATING AND VENTILATING UNITS

Horizontal, vertical, wall and inverted types available. Covers the entire range to 48,000 cfm and 3,320,000 Btu. Filters, face and by-pass dampers, mixing boxes, humidifiers, discharge plenums and nozzles are available. Special units for school applications.

MULTI-ZONE

Heating and Cooling Units

One unit simultaneously heats and cools up to 22 separate zones. Eleven sizes in horizontal or vertical models with capacities from 1350 to 38,000 cfm.

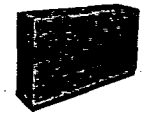
SEASONMASTER

Central Station Air Conditioning

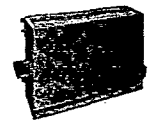
Horizontal and vertical types. Cool, dehumidify, filter and circulate air in summer. Heat, humidify, filter and circulate air in winter. Direct expansion and water cooling coils. Steam and water heating coils. Cooling capacities to 200 tons, from 640 cfm to 38,000 cfm in both suspended and floor types.



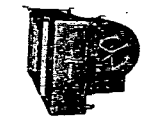
Down Flow Unit Heater



Cabinet Unit Heater



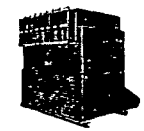
Basic Type Seasonmaker



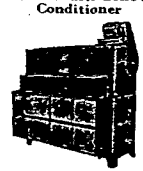
Hideaway Type Seasonmaker



Horizontal Heating and Ventilating Unit



Vertical Multi-Zone Air Conditioner



Vertical Type Seasonmaster

ILG ELECTRIC VENTILATING COMPANY

2880 N. Pulaski Road Chicago 41, Illinois

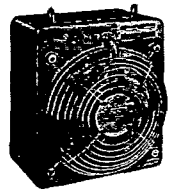
Offices in 57 Principal Cities

Member of *Air Moving and Conditioning Association, Inc. (AMCA)*

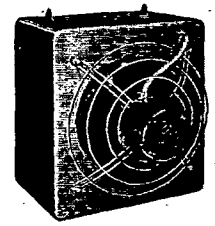
ILG UNIT HEATERS—STEAM, HOT WATER, ELECTRIC OR GAS

Tested and rated by *IUHA* and ASHVE Codes. "One-Name-Plate" guarantee.

ALL-ELECTRIC UNIT HEATERS



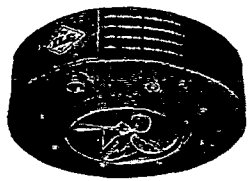
1½ to 7½ kw



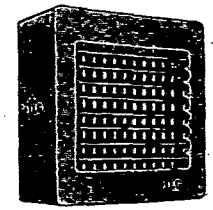
9 to 36 kw

ILG ALL-ELECTRIC UNIT HEATERS—new design for 1½ to 36 kw. features black heat type steel-sheathed elements and Ilg's quiet Type "Q" fan wheel. Built-in thermal cutout and direct-connected, totally enclosed motors. Heavy-duty construction throughout for heating *isolated areas, temporary* or auxiliary heating, stand-by freeze prevention, etc. Send for *Bulletin 808.*

STEAM AND HOT WATER UNIT HEATERS

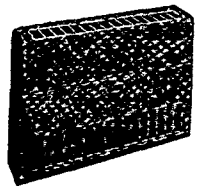


ILG VERTICAL-TYPE UNIT HEATERS—new design. Totally enclosed ball-bearing *motors.* Heating surface features similar to horizontal type. Type "V" for standard outlet temperatures, Type "VL" for low outlet temperatures. 15 capacity sizes. Send for *Bulletin 1202.*



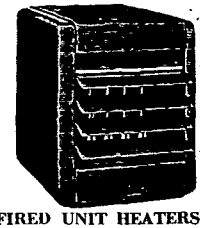
ILG HORIZONTAL-TYPE UNIT HEATERS—four side pipe connections for simplified piping. Suitable for hot water and steam to 150 lb pressure . . . tubes air-tested to 90 lb. Motor totally enclosed, continuous-duty type. 19 capacity sizes. Send for *Bulletin 1202.*

CABINET HEATERS



ILG SERIES "A" CABINET HEATERS—designed for maximum flexibility of arrangement and operation. Can be furnished for floor, wall, ceiling or inverted mounting. Available in seven basic unit sizes, 35 models. Blow-through type with direct fan assembly and draw-through type with belt-drive assembly. Equally suitable for steam or hot water applications. Send for *Bulletin 1506.*

GAS-FIRED UNIT HEATERS



ILG GAS-FIRED UNIT HEATERS—use natural, manufactured, mixed, butane-air mixtures; propane, or butane gas. Available in 10 basic model sizes, 2 types: Propeller Type (Series M); or Blower Type (Series MB) for use with duct work. Safety pilot, automatic control valve, electric limit switch. Motor has thermal overload protection. Send for *Bulletin 286.*

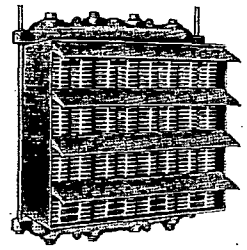
D. J. MURRAY MANUFACTURING CO.

WAUSAU • WIS.

OFFICES IN PRINCIPAL CITIES

MANUFACTURERS OF GRID CAST IRON

UNIT HEATERS, BLAST HEATERS, RADIATORS.



Corrosion IN HEATING EQUIPMENT Ends WHEN YOU INSTALL - - -

CAST IRON STEAM HEAT

GRID

TRANSFER SURFACE

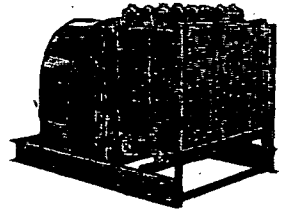
You once and for all eliminate internal electrolytic corrosion because all metals in GRID Cast Iron Steam Heat Transfer Surface that are in contact with steam are similar . . . not 2 or 3 different metals that promote corrosion to cause leaks and breakdowns. GRID Cast Iron Steam Heat Transfer Surface also largely eliminates external corrosion because its cast iron construction resists acids or fumes in the air. When corrosion in heat transfer equipment ends, high maintenance ends . . . it's as simple as that, no mystery, no magic, no "sales" talk. Your heating problems are no longer problems . . . you do away with high maintenance costs, repair work and the nuisance of leaky heating equipment.

GRID Cast Iron construction is designed to withstand steam pressures up to 250 p.s.i. . . . 450 deg temperature eliminating the use of reducing valves where high steam pressures are used. GRID Cast Iron Steam Heat Transfer Surface largely does away with replacement costs because GRID is built to last for years . . . operating successfully without maintenance in many plants since 1929. Sounds worth looking into? Then get the complete story today . . . come the heating season, you will be glad you did.

29 YEARS MAINTENANCE-FREE SERVICE ON ALL TYPES OF GRID EQUIPMENT

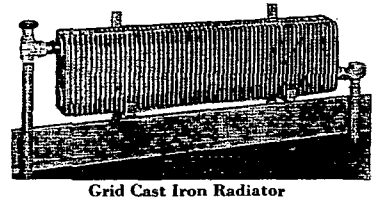
GRID HORIZONTAL BLOW UNIT HEATER DATA

Unit Heater Model	Casing Dimension Width	Casing Dimension Height	Over all Height	Center to Center Hanger Rod	Motor Hp	Motor Rpm	Vol. Fan Cfm	Capacities at 2 PSIG Steam 60° F Ent. Air Btu Per Hr	Capacities at 2 PSIG Steam 60° F Ent. Air Final Temp.	Pipe Size Supply	Pipe Size Return	Support Rod Dia.	Approx. Ship. Wt. Lbs.
CI-1000	13⅜	15 1/16	15 9/16	11¾	1/50	1550	570	27720	103	1¼	1¼	⅜	150
CI-1200	16⅛	15 1/16	15 9/16	15	1/12	1700	800	43320	109	1¼	1¼	⅜	210
CI-1500	15⅜	16	24⅜	17¾	¼	1750	1500	73000	104	1½	1½	½	280
CI-1520	15⅜	21	29⅜	17¾	¼	1750	1700	96730	112	1½	1½	½	390
CI-2000	20⅛	21	29	23⅜	⅓	1150	2600	134000	107	2	1½	½	490
CI-2025	20⅛	26	34	23⅜	½	1150	2578	163000	112	2	1½	½	620
CI-2504	25¾	26	35⅜	27½	½	1150	3200	196000	114	2	1½	¾	680
CI-2500	25¾	26	35⅜	27½	¾	1150	4350	213500	104	2	1½	¾	700
CI-2530	25¾	31	40⅜	27½	½	1150	4650	283630	111	2	1½	¾	900
CI-3000	30⅞	31	40⅜	32⅜	½	850	5900	305000	107	2½	1½	¾	1020
CI-3000	30⅞	31	40⅜	32⅜	1½	1150	3000	362000	101	2½	1½	¾	1070



Grid Blast Heaters in Fan Assembly

A rugged, strong, and economical heat transfer surface, GRID *Blast Heaters are designed for the tough job* . . . for low and high steam pressures . . . require less labor for installation, less cubical space, and being designed and made the same as GRID Unit Heaters are resistant to corrosive atmospheres . . . definitely proved for years of service and maintenance-free.



Grid Cast Iron Radiator

Can be furnished with protective grill

Designed for the tough applications in conjunction with high steam pressure systems, GRID Radiators eliminate the use of pressure reducing valves and because they are built with the same heating sections as GRID Unit Heaters, they will withstand steam pressures up to 250 psi . . . and wide fin spacing prevents collection of dust and dirt . . . particularly adaptable for factory, shop offices, store rooms, laboratories, etc. . . . any place where smaller heating capacities than required of a unit heater are needed.

THE TRANE COMPANY

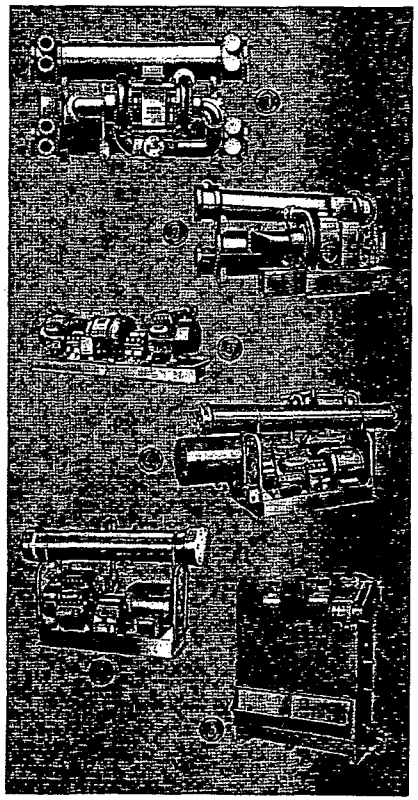
LA CROSSE, WISCONSIN

TRANE

manufacturing engineers of air conditioning, heating, ventilating, and heat transfer equipment

Akron, Ohio	Chattanooga, Tenn.	Fort Wayne, Ind.	Madison, Wis.	Pittsburgh, Pa.	Savannah, Ga.
Albany, N. Y.	Chicago, Ill.	Gainesville, Fla.	Memphis, Tenn.	Portland, Me.	Seattle, Wash.
Albuquerque, N. M.	Cincinnati, Ohio	Grand Rapids, Mich.	Miami, Fla.	Portland, Ore.	Shreveport, La.
Allentown, Pa.	Clarksburg, W. Va.	Greensboro, N. C.	Milwaukee, Wis.	Providence, R. I.	Sioux Falls, S. D.
Amarillo, Texas	Cleveland, Ohio	Greenville, S. C.	Morristown, N. Y.	Raleigh, N. C.	South Bend, Ind.
Appleton, Wis.	Columbia, S. C.	Harrisburg, Pa.	Nashville, Tenn.	Richmond, Va.	Spokane, Wash.
Atlanta, Ga.	Columbus, Ohio	Houston, Texas	Newark, N. J.	Roanoke, Va.	Syracuse, N. Y.
Aurora, Ill.	Dallas, Texas	Indianapolis, Ind.	New Orleans, La.	Rochester, N. Y.	Tampa, Fla.
Baltimore, Md.	Davenport, Iowa	Jackson, Miss.	New York, N. Y.	St. Louis, Mo.	Toledo, Ohio
Baton Rouge, La.	Dayton, Ohio	Johnson City, Tenn.	N. Tarrytown, N. Y.	St. Paul, Minn.	Trumbull, Conn.
Billings, Mont.	Denver, Colo.	Kansas City, Mo.	Oklahoma City, Okla.	Sacramento, Calif.	Tulsa, Okla.
Birmingham, Ala.	Des Moines, Iowa	Knoxville, Tenn.	Omaha, Nebr.	Saginaw, Mich.	Washington, D. C.
Boston, Mass.	Detroit, Mich.	La Crosse, Wis.	Pensacola, Fla.	Salt Lake City, Utah	West Hartford, Conn.
Buffalo, N. Y.	Duluth, Minn.	Lansing, Mich.	Peoria, Ill.	San Antonio, Texas	Wichita, Kans.
Canton, Ohio	Fargo, N. D.	Los Angeles, Calif.	Philadelphia, Pa.	San Diego, Calif.	Wilkes-Barre, Pa.
Charleston, W. Va.	Flint, Mich.	Louisville, Ky.	Phoenix, Ariz.	San Francisco, Calif.	Wilmington, Del.
Charlotte, N. C.					Worcester, Mass.

98 TRANE SALES OFFICES IN UNITED STATES



REFRIGERATION

1. HERMETIC CENTRAVACS

Hermetic centrifugal refrigeration unit—a complete water-chilling system. CenTraVac features automatic capacity control. No seals or gears. Capacities range from 45-1600 tons; 32 sizes. Unit performs efficiently at operating levels down to 10 percent of rated capacity.

2. TURBINE CENTRAVACS

Centrifugal water chillers combining CenTraVac compressor design and steam turbine drive. Integral assembly of compressor and turbine eliminates critical field alignment problems. Sizes from 300 to 1600 tons.

3. RECIPROCATING COMPRESSORS

Two cylinder sizes in 4, 6, and 8 cylinder units. Single or duplex direct drive arrangements with capacities from 10 to 150 tons.

4. COLD GENERATORS

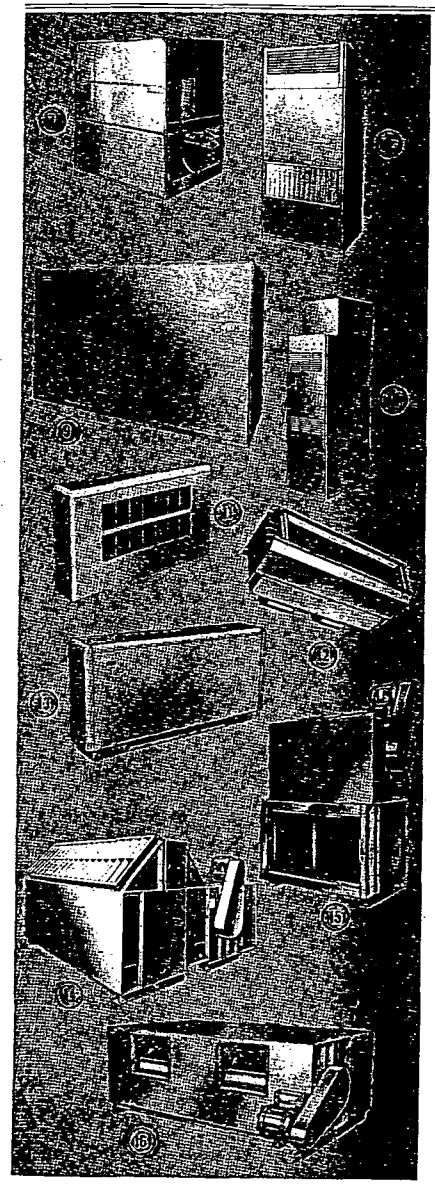
Complete factory-assembled packaged liquid chillers, designed to meet all normal requirements of chilled water systems. Only simple water and electrical connections are necessary for installation. No special foundations are required. Available with capacities ranging from 10 to 150 tons; using Refrigerant 12 and Refrigerant 22. Automatic chilled water temperature controls are also available.

5. CONDENSING UNITS

Trane Reciprocating Compressor complete with water cooled condensers. Capacities 10 to 150 tons; Refrigerant 12 and Refrigerant 22.

6. EVAPORATIVE CONDENSERS

For installations in localities where the use of water for condensing the refrigerant in the air conditioning system is restricted, or where disposal of large quantities of water is a problem. These units condense Refrigerant 12 and Refrigerant 22 by the evaporative process using a constantly recirculated supply of water. The only water lost is that which is evaporated to cool the water being circulated. Sizes from 3 to 100 tons.



The Trane Company

7-8-9. SELF-CONTAINED CLIMATE CHANGERS

Packaged air conditioning units for offices, shops, commercial or industrial air conditioning. Available in 8 sizes—20 models ranging from 3 to 30-ton. The 3, 5, 8, 10, and 15 sizes have deluxe cabinet design to be installed in the conditioned space and blend with any surrounding. The commercial units available in 10, 15, 20, 25, and 30-ton sizes are usually installed in a remote location, outside the conditioned area. Deluxe units available with water- or remote air-cooled condenser. 10, 15, and 20-ton commercial units available with integral evaporative condenser; 20, 25, and 30-ton sizes with either water-cooled or air-cooled condenser.

10. RESIDENTIAL CLIMATE CHANGERS

Heat, cool—or both—with this year-round unit. Cooling unit (top section) may be added later. 2, 3, and 5-ton sizes. Heating unit (lower section) comes in complete range of sizes from 77,000 to 154,000 Btu. Gas-fired. *A.G.A.* approved.

Residential Climate Changer Bonnet Cooling Unit fits atop Trane heating unit—or other modern makes. Compact 2, 3, or 5-ton capacity cooling coil.

Residential Climate Changer Fan-Coil Unit for cooling with independent heating system. Unit, with own quiet centrifugal fan, distributes cooled, dehumidified air. Built-in filter cleans air of dust and dirt. Fits anywhere. 2, 3, and 5-ton sizes.

Residential Climate Changer Duct-Cooling Unit for special applications. Same engineering, same capacity as bonnet type unit.

11-12-13. UNITRANE

UniTrane room air conditioners heat, ventilate, filter and circulate air in the winter. In summer, they cool, ventilate, filter, control moisture content and circulate air. Available in both fan-coil and induction types. Wide range of sizes and models—free standing, recessed, semi-recessed, ceiling-suspended—for exposed or concealed installation. Capacities for 100 to 600 cfm delivery. Automatic Air Flow Regulator provides permanent balancing for Trane high pressure induction air conditioning systems. Acts automatically to assure proper air delivery and performance. Factory-installed for either left-hand or right-hand air supply. System is balanced in a single initial operation; requires no periodic rebalancing thereafter.

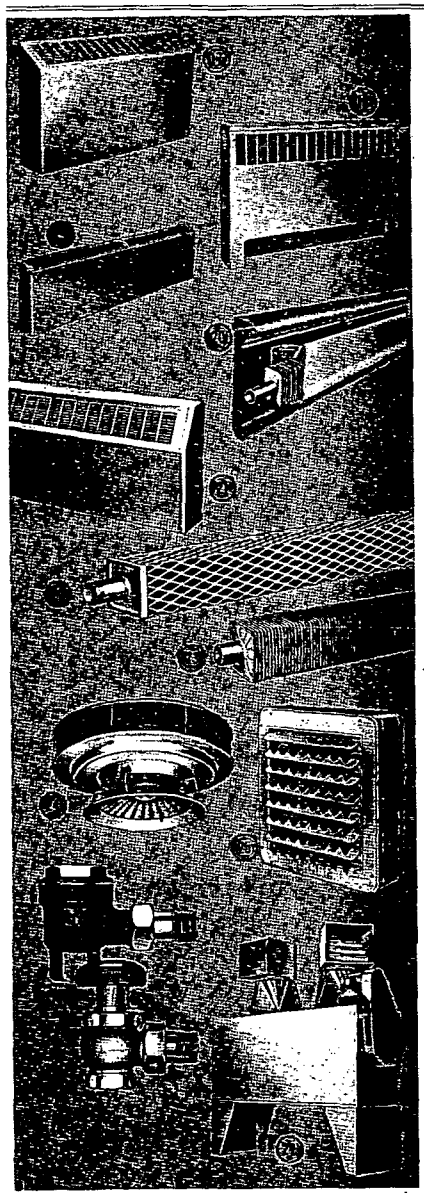
14-15-16. CLIMATE CHANGERS

Any or all phases of air conditioning available in one unit. Two models, horizontal and vertical, each in 12 different sizes. Capacities range from 600 cfm to 39,000 cfm for any application, comfort or process. Multi-Zone Climate Changers provide a number of zones with different air conditions at the same time. Temperature in each zone is automatically controlled by separate set of dampers, making it possible to heat in one zone, while cooling in another. Humidifiers and filters, optional.

EDUCATIONAL MATERIAL

"Trane Air Conditioning Manual" ($6.50), an unbiased 380-page textbook for the engineering profession. New revised edition now available. Many tables and charts included for making computations. In general, text covers heat gains, psychrometry, calculations for the conditioned air supply, refrigeration for air conditioning, water for air conditioning, and ducts and fans.

"Trane Refrigeration Manual," a reference for servicing and installing all types of refrigeration systems. Also available from The Trane Company, La Crosse, Wis. Price $2.00.



THE TRANE COMPANY

HEATING AND VENTILATING

17-18. CONVECTORS

For all steam and hot water systems. Type C—immediately available in many standard sizes. Other units for specific application problems.

19. WALL LINE CONVECTORS

High capacity radiation designed to run continuously below window expanses. Cabinet depths 4 in. and 6 in., heights—14 in., 20 in., and 26 in. Top and front outlet. Chain-type or dial-type knob dampers, optional.

20. BASEBOARD CONVECTORS

Non-ferrous fin-and-tube heating element. Attractive steel cabinets, 8¼ in. or 12 in. height. Damper or valve controls, optional.

21-22-23. WALL-FIN

Ferrous and non-ferrous finned radiation. Lengths 2 ft to 12 ft in 6 in. increments. Cabinets and expanded metal grilles available. Grille louvers two inches in length for greater strength. Ball bearing cradled elements minimize expansion-contraction noise. Spiral damper control for positive temperature control. Enclosures phosphatized and primed after spot welding to prevent rusting.

24. PROJECTION UNIT HEATERS

Model "P" taps the reservoir of ceiling heat, solves a multitude of heating problems. Available in 24 sizes, steam or hot water. Capacities from 40,000 to 560,000 Btu. Louver Cone diffuser, optional.

25. HORIZONTAL UNIT HEATERS

Combines Trane broad blade propeller fan and extended surface coil. In 24 sizes, 22,000 to 352,000 Btu. Louver Fin diffuser, optional.

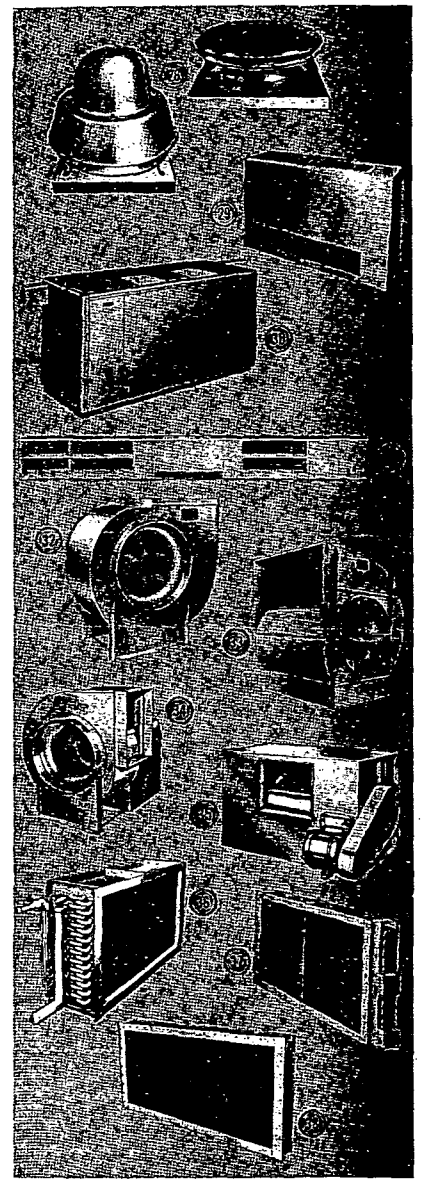
Gas-fired Unit Heaters available in six suspended type models, all *A.G.A.* approved for use with any type of gas—*natural*, manufactured, mixed or liquefied petroleum. One-piece Horizontal-Tube Heat Exchanger combines combustion chamber, heating surface, draft diverter and flue outlet in a single compact unit. Patented Louver Fin Diffuser, optional. All units carry *Underwriters' Laboratories* Seal of Approval.

26. STEAM SPECIALTIES

Complete line including thermostatic traps, valves, float traps, bucket traps, strainers, vents and temperature control valves.

27. TORRIVENT

Large capacity heating and ventilating units combining Trane Centrifugal Fans and extended surface coils for heating and ventilating large spaces and for process applications. Available in 1, 2, or 3-fan units. Choice of 3 coils. Designed for air deliveries ranging from 1150 to 36,000 cfm and for heating capacities from 50,000 to 1,900,000 Btu. Four models available in nine sizes.



THE TRANE COMPANY

28. ROOF VENTILATORS

Complete line of axial and centrifugal roof ventilators for wide variety of applications. Units for exhausting air, for supplying air—units that supply heated air for comfort conditions or process work in the winter season.

29. FORCE-FLO HEATERS

Deluxe cabinet-type unit heater with centrifugal fan. Capacities: 17,700 to 142,000 Btu.

30. UNIT VENTILATORS

Kinetic Barrier System with lateral extensions the length of the outside wall provides constant protection against window downdrafts. Conventional discharge type units with all air being delivered from the unit itself, also available. Automatic or manual temperature control. Quiet, belt-driven operation. Matched *components* and accessories.

31. SHELVING

Available for Kinetic Barrier and conventional unit ventilation in open or closed type. Three sizes with fill-in sections to make wall to wall installations.

32-33-34-35. FANS

Centrifugal fans in non-overloading backwardly inclined or slow speed forward curved types. All standard arrangements in sizes from 12 in. to 89 in. Capacities 668 to 330,048 cfm.

Utility fans, both belt-driven and direct connected, *in non-overloading* backwardly inclined or slow speed forward curved types. All standard arrangements in sizes 4½ in. to 30 in. *Capacities* 110 to 15,450 cfm. Cabinet fans in size 1-7 through 2-27. Capacities 587 to 39,499 cfm.

Propeller fans with direct or belt drive. Sizes 10 in. through 72 in. Capacities 475 to 58,560 cfm.

36-37-38. EXTENDED SURFACE COILS

Heating coils include a complete range of same and opposite end connection steam distributing tube coils, standard coils, heavy duty, high pressure steam coils, and equipment coils.

Cooling coils are manufactured for direct expansion or water cooling in thousands of sizes and types.

All extended surface coils have exclusive Delta-Flo Fin for greater heat transfer efficiency.

L. J. WING MFG. CO.

Office and Factory
LINDEN, N. J.

Canadian Factory:
MONTREAL

Division of Aero Supply Mfg. Co. Inc.

59 Vreeland Mills Road, Linden, N. J.

Branch Offices in
European Representative for Heaters:

Principal Cities
Wanson, Haren-Nord, Brussels, Belgium

WING REVOLVING UNIT HEATERS

ONLY WING OFFERS THESE ADVANTAGES

Distribution outlets slowly revolve—assuring

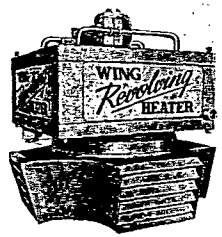
1—Complete distribution of heat over wide areas—around obstacles

2—No uncomfortable steady blasts of heated air

3—Increased worker productivity

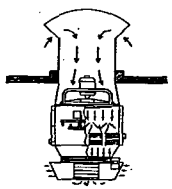
Multiple outlets direct heat in multi-diameter circles. Area entirely covered. Fewer heaters needed. Minimum installation, maintenance and fuel costs. For steam, hot water, electricity, gas. Summer operation gives refreshing breezes—closest approach to air conditioning.

Write for *Bulletin HR-6A*.



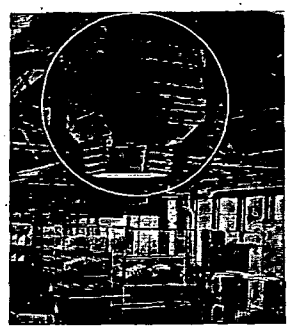
Four types of discharges permit mounting heights from 8 to 60 feet.

WING FRESH AIR SUPPLY HEATERS



Wing Fresh Air Supply Heaters (right and left) provide "make-up" for exhausted air and for health, comfort and efficiency of workers and for improving processes.

Illustrations show 4 way discharge for low ceilings. Other outlets for high and medium ceilings are available.



Coils cannot freeze . . . steam always at full pressure.
Temperatures automatically controlled.
For roof or window mounting . . . horizontal or vertical discharge.
Write for *Bulletin FAS-1*.

OTHER WING HEATERS



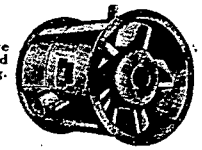
Left . . . Wing Door Heater. Blasts curtain of hot air over open doorways. Keeps cold drafts out.
Right . . . Wing Utility (Horizontal Discharge) Unit Heater. Steam, hot water, gas, electric.



Right . . . Wing Revolving Gas Fired Unit Heater—Self contained for overhead mounting and downward discharge. Output capacities to 368,000 Btu per hr.

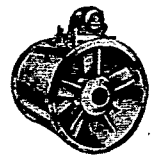


WINGFOIL VANEAXIAL FANS



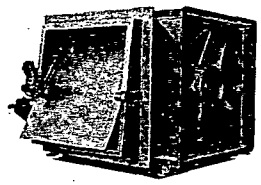
Wingfoil Direct Drive Vaneaxial Fan has sealed motor within the casing. Sizes 10 in. to 42 in.

Wingfoil Elbow Fan replaces elbow in run of duct. Motor, outside the casing, remains cool and clean. All deliver large volumes of air for size of fan without vibration. Capacities to 95,000 cfm. Write for Bulletins.
Wingjet Power Roof Ventilators discharge hot or contaminated air high above building. Two types available: motor inside or outside airstream. *Bulletin PV-1*.



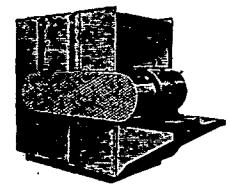
Wingfoil Straightline Vaneaxial Fan forms part of straight line of duct. Sizes 10 in. to 60 in.

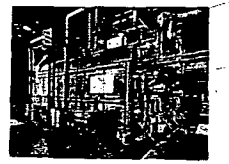
WING BOILER DRAFT EQUIPMENT
WING DRAFT INDUCERS





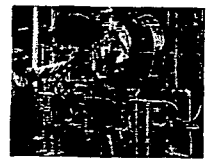
Wing Packaged Draft Inducers for Heating or Power Boilers furnish uniform, adequate draft regardless of wind or weather. Tall chimneys become obsolete. Factory assembled, easy to install in minimum space. Capacities available for all sizes of heating boilers. In power boilers up to 120,000 lb of steam per hour. Send for *Bulletin I-58*.





WING FORCED DRAFT BLOWERS

Wing Motor Driven Blowers. For volumes to 50,000 cfm. Statics to 24 in. Manual or automatic regulation through Voltrol Vanes down to 10 per cent of maximum. Static efficiencies to 80 per cent. Mountings horizontal or vertical. Send for *Bulletin FD-4*.



Wing Turbine Driven Blowers. Easily installed on burner fronts or boiler settings. Capacities to 50,000 cfm and statics to 15 in. Exhaust is oil free. *Bulletin FD-4*.

Wing Dual Drive Blowers provide both turbine and motor drive, either available at will of operator. Send for engineering data.

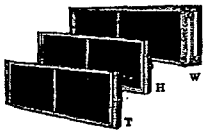
Wing Steam Turbines for driving pumps, fans, compressors, etc. are sturdy, dependable sources of auxiliary power. Capacities to 150 bhp. Pressures to 600 psi. Temp. to 750F. Speeds to 4000 rpm. *Bulletin T-54*.

JOHN J. NESBITT, INC.

State Road and Rhawn Street, Philadelphia 36, Pa.

SALES: HEATING AND COOLING SURFACE sold by representatives of *John J. Nesbitt, Inc.*;
SYNCRETIZERS, AUDICON, ROOMMATE, WIND-O-LINE & THERMOVENTS sold by Nesbitt and American Blower Corporation;
BASEBOARD RADIATION & GAS FIRED UNIT HEATERS sold through Wholesalers;
SILL-LINE RADIATION & UNIT HEATERS sold by *John J. Nesbitt, Inc.*

Products: Heating, Ventilating and Air Conditioning Equipment — Heat Transfer Surface (steam and water), Heating and Ventilating Units, Baseboard Radiation, Sill-line Radiation, and Unit Heaters (steam, water and gas-fired).



NESBITT HEATING AND COOLING SURFACE

Series T. Trombone shaped tubes distribute even smallest amount of steam uniformly under modulating valve control. Maximum freeze-up protection. Ideal for pre-heating outdoor air. Available with steam and condensate connections at same or opposite ends. Copper tubes. Aluminum or copper fins. *Lengths 12" to 120". Publication 305-1.*

Series H. Efficient blast-coil surface for general heating, ventilating and air conditioning, or drying in high and low pressure systems. Seven types; full range of sizes, capacities. *Publication 301-1.*

Series W. For cooling, dehumidifying or heating. Type WD has exclusive Nesbitt drain feature. Surface pitched in casting. Freeze proof. Type WB for booster heating or air cooling where drain is unnecessary. *Pub. 246.* Type WC features cleanable tube sections. *Pub. 255.*



Nesbitt Syncretizer with storage cabinet components

THE NESBITT SYNCRETIZER

The heating and ventilating unit for classrooms that provides quick, economic heat without overheating; draft-free ventilation and natural cooling when needed. A fully automatic, quiet operating unit adaptable to four cycles of control. Models to deliver from 750 to 1560 c.f.m. anemometer rating. *Publication 10-1.*



Wind-O-Line (circled) in rear channel of Nesbitt storage cabinets

NESBITT WIND-O-LINE RADIATION SYSTEMS

Finned radiation, operating in conjunction with the Syncretizer, solves special problems of cold windows and walls. May be concealed by wall hung enclosures or storage cabinets. Provides positive protection against downdraft. 450, 550 and 700 Btu ft. capacities. *SERIES WIND-O-LINE SYSTEM* combines the Syncretizer with Wind-O-Line Radiation on a hot water system to reduce mechanical costs and, at the same time, increase the efficiency and quality of the system. *Publication 10-1.*

NESBITT SERIES T THERMOVENTS

For heating and ventilating large auditoriums, gymnasiums, cafeterias, assembly halls and similar gathering places, Nesbitt Series T Thermovents are available in capacities up to 15,000 c.f.m. standard. For high- or low-pressure applications, steam or hot water. Acoustically "silenced" for *quiet* operation. *Pub. 20-1.*

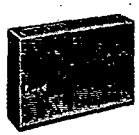
THE NESBITT YEAR-ROUND SYNCRETIZER

Quietly promotes classroom learning by providing balanced heating, ventilation and natural cooling during cold weather... ventilation and mechanical cooling during hot, humid weather. Fully automatic. One-piece motor and fan assembly. Models from 500 to 1500 c.f.m. *Publication 11-1.*



THE NESBITT AUDICON

Sound engineered for large occupancy spaces such as auditoriums, libraries, cafeterias. Provides heating, ventilating and natural cooling in winter; ventilating and mechanical cooling in summer. The latter may be installed initially, or added at a later time. Available in high or low pressure models from 1,250 or 15,000 c.f.m. Cooling from 3 to 60 tons. Heating capacities from 40,000 to 1,500,000 Btu. *Publication 22-1.*



THE NESBITT ROOMMATE

A year-round cabinet conditioner for both winter heating and summer cooling. 7 Standard models. Complete range of air volumes from 200 to 1400 c.f.m.; cooling capacities 1 to 5 tons; heating capacities 25 to 100 MBH. Large variety of recessed, semi-recessed, universal and built-in arrangements possible due to versatility of intake and discharge locations. Outdoor air dampers optional. Install for heating only, if desired... add mechanical cooling in future. *Publication 600-1.*



NESBITT SILL-LINE RADIATION

High capacity finned radiation for perimeter heating in commercial and institutional buildings. Attractive 16-gauge enclosure, rigid one-piece back. Five casing styles, 8 lengths in baked enamel finish. Code ratings, 2.9 to 12.1 sq. ft. EDR. *Publication 30-1.*

NESBITT UNIT HEATERS



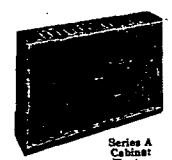
Propeller-Fan Unit Heater

Propeller-Fan Unit Heaters feature universal coils of continuous-tube design — perform equally well with either steam or hot water. Made for high water capacities and temperature drops. Male coil connections through casing rear provide maximum headroom. Sound rated in accordance with A.M.C.A. test code. Modern heavy gauge steel casing; green hammered-tone finish. Eight basic sizes, 24 models, 15,000 to 335,000 Btu/hr. *Publication 401.*



Little "Giant"

Little Giant Unit Heaters. Versatile, draw-through propeller fan units in 34 models from 34,000 to 684,000 Btu/hr; universal heating elements for steam or hot water; side piping for close-to-ceiling mounting; wide variety of outlets; code rated for sound and capacity. *Publication 402.*



Series A Cabinet Heater

Series A Cabinet Heaters for steam or hot water. Smartly styled, compact, quiet, trouble free. Ideal for offices, lobbies, corridors. 7 Standard models, 71 to 545 sq. ft. EDR; 265 to 1880 c.f.m. Four Blow-Through types have direct drive fan assembly. Three Draw-Through types available with belt drive fan assembly. Available for recessed, semi-recessed or non-recessed mounting. Full selection of intake and outlet locations. *Publication 403.*



"Giant" Unit Heater

"Giant" Blower-Fan Unit Heaters. Economical heat for large areas. Floor-mounted, wall-mounted, ceiling suspended types in wide capacity range. Giants of efficiency and endurance. Twelve sizes, choice of 10 heating elements, steam or hot water; 1, 2, or 3 fans; for heating and/or ventilating; from 2000 to 28000 c.f.m., 83,000 to 1,735,000 Btu. *Pub. 404.*



Gas Fired Unit Heater

Gas Fired Unit Heaters. New, light, compact Series K models provide clean, economical, automatically controlled heat for commercial and industrial applications. Ten sizes from 25,000 to 250,000 Btu per hour input; approved for all types of gases. Sound rated. *Publication 400.*

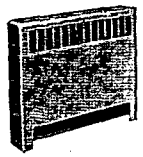
All ratings of Nesbitt Unit Heaters are based on tests made in accordance with the standard test code of the Air Moving and Conditioning Association and the American Society of Heating and Air Conditioning Engineers.

Young Radiator Company

DEPT. 549, RACINE, WIS.

HEATING, COOLING, AIR CONDITIONING PRODUCTS FOR HOME AND INDUSTRY, HEAT TRANSFER PRODUCTS FOR AUTOMOTIVE, AVIATION, AGRICULTURAL, INDUSTRIAL, GAS AND DIESEL ENGINE APPLICATIONS

General offices: Dept. 549, Racine, Wisconsin, U.S.A.
Factories at Racine, Wisconsin and Mattoon, Illinois
Sales and Engineering Representatives in All Principal Cities

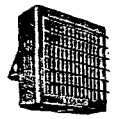


CONVECTORS & UNIT HEATERS

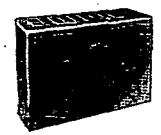
Convectors for steam or hot water systems are available in a wide range of sizes and six standard cabinet styles to fit all applications. Also available in institutional, low-level and bathroom models.



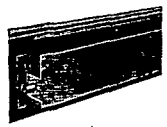
Vertiflow® Unit Heaters for use in steam or hot water systems are available in 22 models with sizes and capacities that range from 52,500 to 1,062,000 Btuh. Excellent for heating industrial buildings, garages, warehouses, and large high-ceiling airplane hangars. Outstanding Young feature is the through-stack motor ventilation which provides motor cooling. Units are fitted with Spreadaire diffusers, Anemostats, or Aerojet nozzles to meet specific air distribution requirements.



Type H Horizontal Unit Heaters are built in 15 models; with capacities from 19,000 to 325,000 Btuh; for steam or hot water systems.



Cabinet Type Unit Heaters are available in 3 sizes, with capacities from 29,900 to 117,300 Btuh, and in floor mounted, wall-hung, inverted wall-hung, and horizontally suspended models.

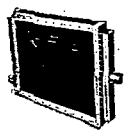


Perimaheat® Baseboard Convector for hot water service; available in 6 and 8 ft lengths; capacity with 190 deg average water 787 Btu per linear foot. Accessories include end caps, inside and outside corners, and optional splitters or dampers.



Gas-Fired Unit Heaters are available in eight sizes ranging in capacities from 50,000 to 230,000 Btuh input. Have welded corrosion-resistant aluminized steel combustion chamber and heat exchanger; other important design features.

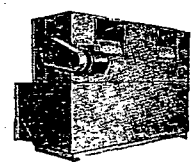
HEATING AND COOLING COILS

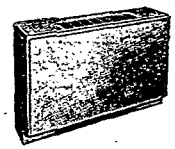


Young manufactures a complete line of heating and cooling coils for central heating or cooling systems. All steam coils are pitched in casing to allow proper condensate drainage.

AIR CONDITIONING UNITS

Young Type "CS" Central Station Air Conditioning Units are available in 10 sizes with capacities from 500 to 28,000 cfm, and in three models—Type H (Horizontal), Type V (Vertical), and Type MZ (Multizone). They can be assembled in any combination, from a simple ventilation installation to a complete air-conditioning system, including heating and humidifying, cooling and dehumidifying, and filtering. Their simplified sectional design makes it possible to select a unit to exactly fit the requirements of the job.





Young "Roomaire"® Conditioner is a remote-type room air-conditioning unit that provides year-round air-conditioning of individual rooms in multi-room installations. For use with hot water and chilled water systems. Individual controls. Four sizes—handling from 200 to 600 cfm with nominal cooling capacities of ½ to 1½ tons. Four models—F and CF for free standing or concealed floor installation; H for horizontal installation; CH for concealed horizontal installation.

ELECTROMODE
Division Of
COMMERCIAL CONTROLS CORPORATION

Complete line of ELECTRIC HEATING SYSTEMS for HOME - INDUSTRY - FARM

ROCHESTER 3, NEW YORK

When you need advice in any field, you consult a specialist. Deciding on a heating system is an important matter—one that you do not undertake very often. Therefore, it is only natural and in the course of wisdom, that you will look to a specialist in this field. For more than 25 years Electromode has devoted its research engineering and manufacturing facilities predominantly to the design and production of electric space heating equipment, and is one of the oldest in the business. The Safety Grid Heating Element is an Electromode feature. All electric wires are insulated, embedded and completely sealed inside a finned aluminum casting. No exposed hot wires, no glowing coils and no danger of fire, shock, or burn. This finned aluminum casting spreads the heat rapidly, resulting in higher efficiency and greater economy. The built-in safety switch automatically prevents overheating. The Safety Grid Heating Element is guaranteed for 5 years against any defect in material or workmanship.

For HOME and OFFICE



Fig. 1



Fig. 2

WALL-TYPE HEATERS

Electromode Wall-type Heaters are made in models and capacities suitable for all size rooms, ranging from 1320 to 4000 watts, with built-in thermostat or manual control. Heat is fan-circulated at floor level. All have automatic thermal safety switch. The big room heaters have silver grey finish and can be painted to match decor. The small room or bathroom heaters are available in gleaming chrome or white enamel and are used as complete or auxiliary source of heat. See figs. 1 and 2.

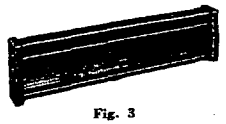


Fig. 3

BASEBOARD HEATERS

Designed so that a completely harmonious installation can be made which conforms to design and dimensions of any room. Matching blank sections, end pieces and corners for carrying out unbroken baseboard effect. The comfort of radiant warmth along outer walls, under windows and across the floor, thermostatically controlled at the comfort level you select for each room. There's no interference from Electromode Baseboard Heaters. They're so out of the way along the walls just like regular baseboard. 600, 900, 1200 watts; 120 or 240 volts. See fig. 3.

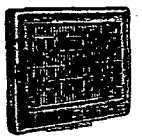


Fig. 4



Fig. 5

RADIANT HEATING PANEL

Combines radiant and convection heating. Modern design with silver grey finish. With or without built-in thermostat. Capacity 1100 watts. See fig. 4.

RADIANT CABLE HEAT

Coils of cable hidden in ceiling radiate clean heat downward. Each room temperature controlled by separate thermostat. Cables are a measured length of special insulated electric wires; waterproof; non-corrosive. Used in plaster, drywall ceilings, or concrete floors. Magic Seal coating prevents streaks on ceiling. See fig. 5.

For INDUSTRY



Fig. 6



Fig. 7

UNIT HEATERS

Unit Heaters available in Suspension-type and Combination Portable and Suspension models. Capacities from 1500 to 45,000 watts. Fine for auxiliary warmth in hard-to-heat areas and for complete heating in outlying buildings. Adjustable louvers for directing fan-circulated heat into working zone. Thermostat control available on all models. See fig. 6 for Suspension and fig. 7 for Portable and Suspension. The Electromode Commercial and Industrial heater is specifically designed for quick, comfortable, warmth in factory departments, offices, supermarkets and other areas where its quiet operation is an asset. It is easily portable, or it may be mounted on swivel-type brackets. 5KW—240 volts—single or 3 phase—17,075 Btu. See fig. 8.



Fig. 8



Fig. 9

EXPLOSION-PROOF HEATERS

Explosion-Proof heaters are specially engineered for installation in hazardous areas where heat is required and danger of explosion exists. They conform to rigid safety specifications and are fully tested and approved by *Underwriters' Laboratories* for use in areas covered in Class 1. Group D hazardous locations. Used in places where atmospheres contain gasoline, petroleum, naphtha, solvent vapors, natural gas and where flammable gases etc. are made or stored. Convection type. Three models range from 2000 to 6000 watts. See fig. 9.

BLAST COILS

Consists of a bank of Electromode cast-aluminum heating elements mounted in a 16 gauge formed steel frame having a 1½ in. wide flange with 5/16 in holes on approximately 6 in. centers for mounting to air ducts. Complete wiring diagrams for special applications furnished on request. See fig. 10.

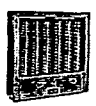


Fig. 10

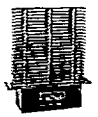


Fig. 11

Electromode Insert Duct Heater (fig. 11)

Safety and endurance is inherent in its construction. It was designed as a package unit ready for easy installation. The Safety-Grid heating element is your assurance of complete protection against fire, shock or burn. Safety switch automatically breaks the service to the heater in the event of stoppage of air flow through the heater from any cause. It is an automatic reset type.

Engineering help gladly given. For more information, see your supplier or write Dept. HVG-19, Electromode Division of Commercial Controls Corporation.

INDUSTRIAL ENGINEERING AND EQUIPMENT COMPANY

22 Hanley Industrial Court, St. Louis (Brentwood) 17, Mo.

INDEECO

Electric Blast Coil Heaters

for Installation in Ducts
and in Package Air Conditioners
Total and Supplementary Heat

Package Units
sized to specifications

SAVE YOU Design & layout work
Field time

Commercial, industrial, institutional, residential and governmental installations are heated by these patented units. They offer the following advantages:

1. **COMPLETE UNIT,** ready for installation, eliminating design and layout work
2. **SIZED TO FIT**
3. **THERMAL CUT-OUTS,** furnished with each unit
4. **WIDE SIZE RANGE**

INDEECO UNITS FOR PACKAGE AIR CONDITIONERS. Write for *Bulletin E97P* showing 704 sizes to fit 20 manufacturers' package units.

INDEECO UNITS FOR DUCT-INSTALLATION. Write for *Bulletin E97U* showing 422 sizes.

SPECIAL UNITS in both categories furnished at no extra cost. INDEECO Blast Coil Heaters are made to suit your specifications. You are not compelled to change your plans to meet our specifications.

A FEW EXAMPLES OF USES

1. Primary source of heat
2. Booster, supplementary or standby auxiliary heating
3. ZONE temperature control
4. Industrial applications
   (Example: for major oil refinery, seven pumping stations without attendants are heated by INDEECO blast coil heater installations)

Literature describes wide range of applications in detail. More than 10,000 specifications on file.

WRITE FOR CATALOGS
CONSULTATION ON ANY PROJECT
WITHOUT CHARGE

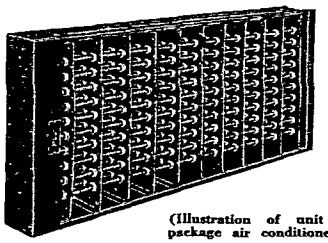


22,621 INSTALLATIONS to date
150 to 17,500,000 Watts

Typical examples include:

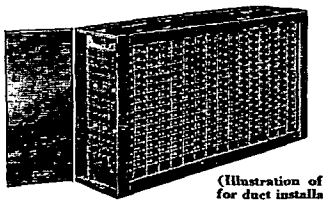
SUN-TIMES BUILDING, CHICAGO
CADILLAC MOTOR CO., DETROIT
GENERAL FOODS CORP., WHITE PLAINS
ROCKEFELLER CENTER, N. Y. CITY
LOYOLA UNIV., NEW ORLEANS
U. S. EMBASSY OFFICE, HONGKONG
I. MAGNIN DEPT. STORE, PALO ALTO, CALIF.
ST. LAWRENCE SEAWAY (U. S. AND CANADIAN SIDES)

AND OTHERS



(Illustration of unit for package air conditioners)

INDEECO Electric Blast Coil Heater for Package Air conditioners as shipped and ready for installation



(Illustration of unit for duct installation)

INDEECO Electric Blast Coil Heater for duct installation. Listed by *Underwriters Laboratories*. Above is part of installation totaling 1,680,000 watts.

Edwin L. Wiegand Company

7672 Thomas Boulevard Pittsburgh 8, Pa.

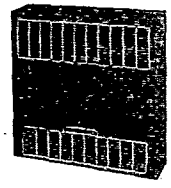
Representatives in all Principal Cities

PRODUCTS: Over 15,000 types, sizes and ratings of Chromalox electric heaters for Industry and Home. All heating elements are metal or alloy sheathed in metal for strength and corrosion resistance, fully insulated for safety.



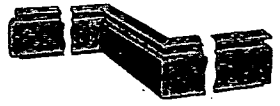
CHROMALOX ELECTRIC UNIT VENTILATOR

The Chromalox Electric Unit Ventilator has been specifically designed to meet the *requirements* for proper heating, ventilating and natural cooling of classrooms and similar rooms . . . wherever it is normal for many people to gather in a limited space. Fast response to a room's changing requirements . . . from heating and ventilating to natural cooling, give the versatility required for these applications.
Normally installed below a window sill, a louvered opening through the wall permits the introduction of fresh *outdoor* air into the Chromalox Unit for proper ventilation and natural cooling. The unit can be supplied with automatic room temperature controls.
To eliminate discomfort from window downdraft, Chromalox Electric Baseboard Heaters are recommended on both sides of the Chromalox Unit Ventilator extending the full window width.
Units are available in 500, 750, 1000 and 1250 cfm Standard Air Ratings and up to 18.0 kw (61,500 Btu/hr) capacities. *Bulletin D-52.*



CHROMALOX ELECTRIC CABINET CONVECTOR HEATERS

Chromalox Electric Cabinet Convector Heaters are completely enclosed, 18-gage metal casings, designed to give easy bottom access of air which is then heated by a bank of metal-sheathed strip heaters and convected outward and upward. Available in 6 in. deep free-standing models or semi-recessed models projecting 2¼ in. from wall surface. Heating capacities: 1, 2, 3 and 4 kw. *Bulletin 975.*



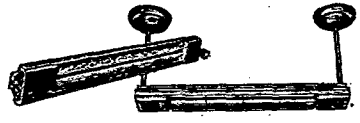
CHROMALOX ELECTRIC BASEBOARD RADIATORS

Clean, balanced radiant and convected heat supplied by rugged baseboard measuring only 8-⅞ in. high and 2-⅜ in. deep. Lengths in standards of 1 ft with 100 watts, 2 ft with 300 watts, 5 ft with 800 watts, 8 ft with 1250 watts. Extremely easy to install. Heating element is all metal fully enclosed, flat strip type which is break-proof, moisture-proof and shock-proof. Designed for maintenance-free heating in homes, offices and public buildings. *Bulletin 975.*



CHROMALOX AIR DUCT HEATERS

For use in square or rectangular forced air ducts. Construction allows free-flow of forced air circulation. Finned elements present maximum contact for fast heating and minimum air friction. For room heating and process work. Ratings from 6 kw to 100 kw. *Catalog 60.*



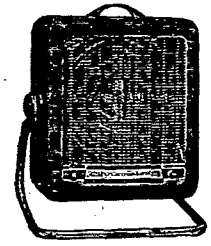
CHROMALOX ELECTRIC FAR-INFRARED COMFORT HEATERS

"Spot heating" for personal comfort without heating entire rooms or buildings. May be ceiling-, wall-, or floor-mounted as best suited for various conditions. Heater housing is aluminum extrusion. Stainless steel grill protects against accidental contact. Lengths from 24⅜ in. to 85¾ in. Individual temperature control by Chromalox thermostat, or several heaters may be controlled on automatic cycles with Chromalox input controller. *Bulletin F-1614.*

Edwin L. Wiegand Company

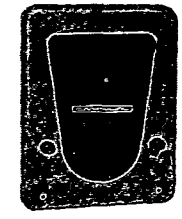
7672 Thomas Boulevard Pittsburgh 8, Pa.

Representatives in all Principal Cities



CHROMALOX ELECTRIC PORTABLE COMFORT HEATERS

Model shown is one of several available. Outputs range to 13,648 Btu's. Available with manual "on-off" switch or automatic thermostat. Swivel stand adapts for wall or ceiling mounting. *Bulletin 975.*



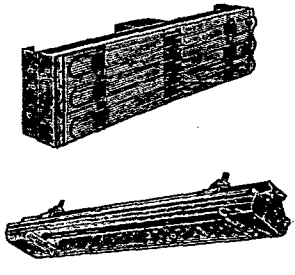
CHROMALOX ELECTRIC FORCED-AIR WALL HEATERS

Smooth molded contours of heavy-gage steel form the front grille. Recessed section contains a quiet, fully-enclosed motor and dustproof fan. Metal sheathed Fintube element and fan are controlled by manual on-off switch. Variable temperature control knob for setting built-in thermostat. 1500 to 4000 watts for 240 a-c supply for game rooms, living rooms, motels, reception rooms, etc. *Bulletin 975.*



CHROMALOX ELECTRIC CIRCULATION HEATERS

Circulation Heaters heat fluids in convection or forced heating systems; for washrooms, laundries, dishwashers, central heating systems, superheating and drying steam, preheating fuel oils and heat transfer fluids, gases. Temperature to 750F. *Catalog 60.*

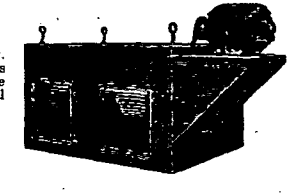


CHROMALOX FAR-INFRARED UNITS

Far-Infrared Radiant Heat for heating, drying, baking, dehydrating, etc. Compact pre-engineered panels, heaters and lamps can be made into banks, tunnels, ovens. All-metal construction protects against breakage, splashing; high safety against shock hazards. *Catalog 60.*

CHROMALOX ELECTRIC HIGH VOLUME HEATERS

For heating of larger areas or as central heating system. Ratings from 15 to 40 kw. Twin centrifugal fans, driven by a 3-phase motor, draw air into unit and across Chromalox Finstrip Heaters. Air deflectors furnished for each outlet to discharge warmed air slightly downward. May be connected to ducts of conventional central system when required. Fans may also be used for summer cooling. *Catalog 60.*



4965

WESIX ELECTRIC HEATER COMPANY

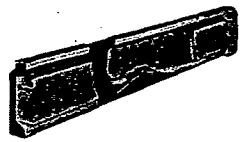
390 First Street

San Francisco 5, California

CHICAGO DALLAS DENVER DETROIT HUNTSVILLE
LOS ANGELES NEW YORK PORTLAND SEATTLE

Complete Line of Electrical Air and Liquid Heating Equipment for Residential, Commercial & Industrial Applications

BASEBOARD PERIMETER PANELS with monofin element available in standard 32 in. and 48 in. lengths for simple, inexpensive installation without special tools or metal cutting. 250 watts (853 Btu) per linear foot. Control section (thermostat and switch), Air-conditioning Supply Section and Receptacle Section, each 6 in. long. Corner section, hood and end pieces available. Approved by *UL*.





AUTOMATIC ELECTRIC BRACKET TYPE HEATER for commercial, institutional or industrial applications, these units are available in a large range of sizes and types including explosion-proof models for use In Class I, Group C and D locations. Approved by *UL*.



AUTOMATIC ELECTRIC WALL FURNACE sizes range from one to eight kilowatts for use in domestic and commercial installations. Available in radiant-convection (model illustrated), convector or blower type (5000 watt maximum). Approved by *UL*.



DUCT HEATERS utilizing either Wesix "dust free" or conventional transverse fin elements, these units may be ordered from a wide variety of self contained, standard units, or built to order. Units available complete with sensitive integral overtemperature control.



LOAD REGULATORS control electric heating capacity automatically by switching voltage supply from 240 volts to 120 volts by means of outdoor thermostatic control. Controls heating loads up to 24 kw. Approved by *UL*.

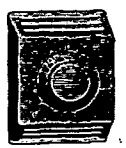
INDUSTRIAL UNIT HEATERS—BLOWER TYPE for horizontal or vertical mounting either as suspended unit heaters or duct system use in standard sizes to 60 kw. Larger sizes or models for special applications on order. Approved by *UL*.



QUIET AUTOMATIC SWITCH, a motor operated contactor for resistance loads to 24 kw, 1ϕ or 40 kw 3ϕ, 240 volts and 33 kw 1ϕ or 55 kw 3ϕ, 480 volts. Actuated by SPDT switch or thermostat. Available with sequence action for control from single switch of two or more contactors. Approved by *UL*.



COMMERCIAL UNIT HEATERS—AXIAL FLOW sizes from 3 to 40 kw for ceiling or pedestal mount. Feature Wesix longitudinal fin, "dust free" inclosed heating elements. Approved by *UL*.



HEAVY DUTY WALL THERMOSTAT AND SWITCH for control of electric heating apparatus directly up to 25 amp, this unit includes a lockout switch double pole in the "off" position. Thermostatic action is bi-metallic, sensitive to plus or minus 1 degree F. Fits standard wall box mounted vertically. Case color: Ivory. Approved by *UL*.



THE AMERICAN BRASS COMPANY

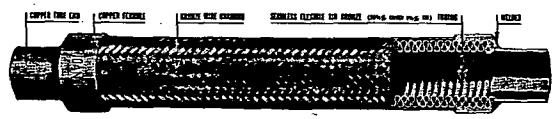
AMERICAN METAL HOSE DIVISION

WATERBURY 20, CONNECTICUT

Specify "AMERICAN" VIBRATION ELIMINATORS Stocked by leading wholesalers

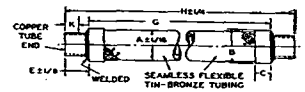
AN ANACONDA PRODUCT

AMERICAN VIBRATION ELIMINATORS
for the important job of CONVEYING REFRIGERANTS
and minimizing transfer of VIBRATION and NOISE



STANDARD SIZES

Standard copper tube "slip-fittings" welded on both ends. Suitable for any industrial or domestic air conditioning or refrigeration unit or any general piping encountering vibration.



V. E. DATA TABLE I

TO FIT COPPER TUBE Actual OD	Nominal	Flexible Tubing I.D.	FEMALE ENDS Part No.	H	U. L. No. on Ferrule*
1/4	1/8	5/16	3618F	7	W-36
1/4	1/8	1/4	1418F	7½	W-14
5/16	1/4	5/16	5614F	8¼	W-56
1/2	3/8	3/8	3838F	9	W-38
5/8	1/2	1/2	1212F	9¾	W-12
3/4	5/8	1/2	1258F	10	W-12
3/4	5/8	3/4	3458F	11¼	W-34
7/8	3/4	3/4	3434F	11½	W-34
1 1/8	1	1	1010F	13	W-10
1 3/8	1 1/4	1 1/4	5454F	14¾	W-54
1 5/8	1 1/2	1 1/2	6464F	17	W-64
2 1/8	2	2	2020F	20	W-20
2 5/8	2 1/2	2 1/2	5252F	24	W-52
3 1/8	3	3	3030F	27	W-30
3 5/8	3 1/2	3 1/2	7272F	32	W-72
4 1/8	4	4	4040F	33	—
5 1/8	5	5	5050F	41	—
6 1/8	6	6	6060F	49	—
8 1/8	8	8	8080F	62	—

Listed by *Underwriters' Laboratories, Inc.* in copper tube sizes through 3⅝ in. O.D. • I.D. of Copper Tube Ends for Depth "K" meets *ASA* Std. B16.18—1950 and *ASA* B16.22—1951. • 2½ in. I.D. and larger are double-braided with tin bronze wire for extra strength and protection. • Sizes above 3 in. I.D. are soldered under the ferrules after welding. • Copper Tube Ends on sizes above 3 in. are commercial parts and "E" and "H" may exceed tolerances shown. • "G" Tolerance: plus or minus ½ in. for sizes through 4 in. I.D. Above 4 in. I.D.: plus or minus ½ in. • VE's are sealed in polyethylene envelopes for internal and external protection. • Also available with male ends. Write for specifications. • **Underwriters' Laboratories (UL)* number on ferrule may appear with prefix "M" instead of "W." • Neoprene latex covered VE's are available for use when danger exists from condensed moisture on the outside of the unit

V. E. DATA TABLE II

I.D. Flexible Tubing	A	B	C	E	G
5/16	7/16	1/2	3/8	1/2	6
1/4	1/2	5/8	3/8	1/2	6½
5/16	5/8	1 1/16	3/8	5/8	7
3/8	3/4	1 3/16	3/8	3/4	7½
1/2	1	1 1/16	7/16	7/8	8
1/2	1	1 1/16	7/16	1	8
3/4	1 5/16	1 7/16	9/16	1 1/8	9
3/4	1 5/16	1 7/16	9/16	1 1/4	9
1	1 9/16	1 3/4	5/8	1 1/2	10
1 1/4	1 7/8	2	1 1/16	1 5/8	11½
1 1/2	2 1/4	2 3/8	1 3/16	2	13
2	2 7/8	3	1	2 1/2	15
2 1/2	3 5/8	3 7/8	1 3/16	3	18
3	4 1/4	4 1/2	1 3/8	3 1/2	20
3 1/2	4 3/4	5	2	4	24
4	5 1/2	5 3/4	2	4 1/2	24
5	6 3/4	7	2	5 1/2	30
6	7 3/4	8	2	6 1/2	36
8	9 3/4	10	2	7	48

The American Brass Company

Waterbury 20, Conn.

ANACONDA
from mine to consumer

District Offices in Principal Cities

IN CANADA: ANACONDA AMERICAN BRASS LIMITED, NEW TORONTO, ONTARIO

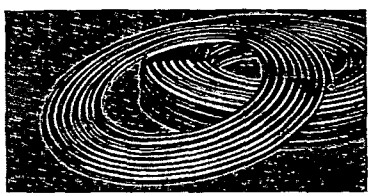
PRODUCTS—Anaconda Phosphorized Copper Tubes and Fittings; Anaconda "85" Red Brass Pipe; Everdur Metal for storage heaters, storage tanks, ducts and air-conditioning equipment

Anaconda Types K and L Phosphorized Copper Tubes, assembled with solder-joint Fittings, offer an unusual combination of advantages for hot-water and low-pressure-steam heating systems, including radiant panels. These advantages may be summarized briefly as follows:

Low Friction Loss—Smooth, nonrusting inside surfaces of copper tubes do not become rough, and offer an unchanging low resistance to flow.

Ease of Installation—The flexibility of copper tubes simplifies connections which would be awkward and expensive with rigid pipe, and solder-joint fittings can be installed in restricted places where the use of a wrench would be impossible.

They meet the requirements for these types of tubes in Federal Specification WW-T-799a and *ASTM* Specification B88, when so specified. Type K, the heavier, is recommended for heating lines and general piping.



Anaconda Copper Tubes, in standard sizes, are furnished soft in 60- and 100-ft coils; also hard and soft in 20-ft straight lengths.

Anaconda Type M Copper Tube is furnished hard drawn in 20-ft straight lengths in nominal sizes from 1¼ in. to 12 in. for drainage lines in certain types of air-conditioning installations. This tube has a thinner wall than Types K and L and provides economies in weight and cost when used in the larger sizes. Type M tube is assembled with wrought-copper or cast-brass solder-joint fittings.

***PG's (Pre-Formed Panel Grids)** Anaconda Panel Grids are machine-formed radiant-heating coils of Type L copper tube, supplied in two standard panel sizes and two nominal diameters (⅜ in. for ceilings, ½ in. for floors). They can be contracted or expanded by hand to obtain a range of C-C spacings. *Publication C-6* describes economies of PG's, gives simplified design suggestions and short-cuts to calculating heat output.

* Trademark Reg. U. S. Pat. Off.



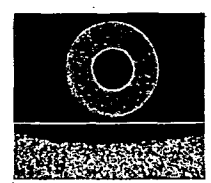
Heat for this basement game room will be provided by ½ in. PG's *imbedded* in the concrete. Troublesome hand bending and resulting sags and dips that increase friction loss are eliminated with pre-formed PG's.

CUSTOM-MADE CAPILLARY TUBES

Anaconda Custom-Made Capillary Tubes, for *restrictor* purposes, are made to specific mutually agreed-upon airflow limits. These limits alone are the basis for production.

Initial shipments contain Master Reference Sample Tubes with labels giving maximum and minimum flow capacities agreed upon. Duplicate Master Samples are kept in the mill files. Every subsequent shipment of tubes will be made to the limits of the Master Samples.

All Anaconda Restrictor Tubes are plug-drawn to finish to provide a smooth, round inside bore. Each length is chambered at both ends, thoroughly washed and dried, given a final air-flow test, and carefully paper-wrapped in a bundle.



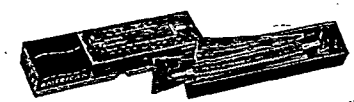
(Top) A cross-section of an Anaconda Copper Restrictor tube, .081 in. OD x .031 in. ID, magnified 10×. Note the roundness of the bore. (Bottom) Section of a photomicrograph at 200× magnification to show smoothness of the bore.

The American Brass Company



REFRIGERATION TUBING

Anaconda Dehydrated Copper Refrigeration Tubes are manufactured in accordance with *ASTM* Specification B280 in all standard sizes up to and including ⅞ in. O.D., in 50-foot coils. The tubes are sealed immediately after annealing and dehydrating.



VIBRATION ELIMINATORS

American Vibration Eliminators—Compressor vibration and noise are muffled in a line equipped with an American Vibration Eliminator. The corrugated tin-bronze tubing is *seamless*. Copper ferrules and tube ends are *braze welded*. Each unit is pressure-tested under water, is spotlessly clean and dry, and is packed in a polyethylene envelope.

ANACONDA "85" RED BRASS PIPE

Anaconda "85" Red Brass Pipe, in standard pipe sizes, is considered the highest quality corrosion-resistant pipe commercially obtainable at a moderate price, and is the material preferred by many engineers for steam condensate return lines.

EVERDUR*

Everdur Metal is the original copper-silicon alloy group. These high-strength, highly corrosion-resistant engineering alloys are manufactured by The American Brass Company in five standard compositions and in practically all commercial forms.

In addition to their nonrusting properties and high strength, Everdur alloys possess many qualities not usually found in metals of this character. They are unusually resistant to general atmospheric conditions and other normally corrosive factors. Everdur alloys have excellent machining and working characteristics, and can be fabricated into a variety of forms and shapes. Everdur alloys are available for oxyacetylene, carbon-arc and inert-gas arc welding.

Everdur Tanks—Everdur copper-silicon alloy is an ideal material for durable, nonrusting water tanks of every description—from domestic range boilers to large storage heaters for hotels, laundries, hospitals, textile plants, schools or breweries.

Everdur is made in all commercial shapes, including annealed tank plates which have physical properties as given in *ASTM* Specification B96.

Minimum specification requirements for hot-rolled-and-annealed tank plates are: Tensile Strength, 50,000 psi.; Yield Strength (at 0.5 percent elongation under load) 18,000 psi.; Elongation, 40 percent in 2 inches.

Welds made with annealed Everdur tank plates meet the requirements for construction in the *ASME* Boiler and Pressure Vessel Code.

For additional data and names of fabricators, address our nearest District Sales Office.

EVERDUR FOR AIR-CONDITIONING EQUIPMENT

Because of its strength and welding properties, Everdur may often be substituted for steel and fabricated by substantially the same methods, with much the same equipment as steel.

Everdur metal has been used with marked success for fans, blowers, ducts, humidifiers, cast and wrought parts of other equipment items subject to corrosive influences.

EVERDUR LITERATURE

Descriptive literature containing much pertinent tabular data will be sent on request.

* Trademark Reg. U. S. Pat. Off.

RESTRICTOR TUBE • FORMED TUBE PARTS • HARD COPPER TUBE CUT TO LENGTH • COPPER TUBE IN COILS AND STRAIGHT LENGTHS • PRE-FORMED RADIANT PANEL GRIDS • FITTINGS FOR TYPES K AND L TUBES • VIBRATION ELIMINATORS • FLEXIBLE REFRIGERATION TUBING CONDUIT • DIE PRESSED FORGINGS • COPPER, BRASS, BRONZE IN SHEETS, WIRE, RODS, TUBES AND SPECIAL SHAPES

CALUMET & HECLA, INC.
CALUMET DIVISION
URANIUM DIVISION
GOODMAN LUMBER DIVISION
WOLVERINE TUBE DIVISION
In Canada:
CALUMET & HECLA OF CANADA LIMITED
WOLVERINE TUBE DIVISION
CANADA VULCANIZER & EQUIPMENT CO. LTD.
UNIFIN TUBE DIVISION

WOLVERINE TUBE
DIVISION OF
CALUMET & HECLA, INC.
17286 Southfield Road
Allen Park, Michigan
Manufacturers of Quality-Controlled Tubing and Extruded Aluminum Shapes

PLANTS IN DETROIT, MICHIGAN AND DECATUR, ALABAMA. SALES OFFICES IN PRINCIPAL CITIES.



WOLVERINE ROLL-O-TUBE•

. . . copper water tube in a round carton that rolls

Wolverine Roll-O-Tube represents a new packaging concept for rolls of copper water and refrigeration tube. Roll-O-Tube for example, simplifies tube handling. It can be rolled, like a hoop, to job site or storage. A convenient center hole makes carrying easy. By holding the tube at one end and unreeling the required length, Roll-O-Tube functions as a reel. Also of importance to users is the fact that unused tube remains in the Roll-O-Tube carton protected against damage and dirt until needed again. Large easy-to-read print and color coding on the opening tape make identification fast and sure.

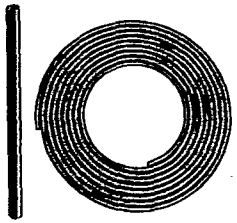
WOLVERINE VERSATUBE•

Aluminum Utility Tube used for air lines, connecting lines for pumps, stoves, oil burners, etc. Packaged in master cartons. Available in from ⅛ in. O.D. to ⅜ in. O.D.

WOLVERINE COPPER REFRIGERATION TUBE

Wolverine copper refrigeration tube is well known throughout industry. It, too, is available in straight lengths and rolls. In roll form, Wolverine copper refrigeration tube comes packaged in the convenient Roll-O-Tube carton. Wolverine refrigeration tube is dehydrated, has a clean, mirror-bright inside surface and is consistent in quality and temper. As a further service to its customers, Wolverine refrigeration tube is provided with unique plastic plugs for positive end-sealing. These plugs are removable and may be used again and again; because their O.D. is no larger than that of the tube there is no need for removing the plug when threading through partitions.

• Patent Applied For.



WOLVERINE COPPER WATER TUBE

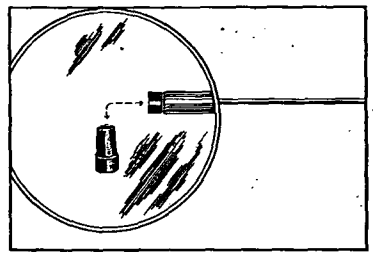
Wolverine copper water tube is available in three types:

Type K—in straight lengths and rolls—is recommended for general plumbing where severe service conditions may be encountered as well as for heating, gas, steam, and oil lines. It is also used for underground service. It is available in nominal sizes from ¼ in. through 6 in.

Type L—in straight lengths and rolls—is recommended for general plumbing involving less severe conditions and for internal applications for heating, gas, steam and oil lines. It is available in nominal sizes from ¼ in. through 6 in.

Type M and Type DWV—available in straight lengths only—are ideally suited for DWV (drainage, waste, vent) installations and for other non-pressure applications where soldered fittings only are used. Type M is available in nominal sizes from ⅜ in. through 6 in.—Type DWV, 1¼ in. through 4 in. Wolverine also manufactures standard and extra heavy S.P.S. pipe in copper and red brass. Pipe is available in standard lengths of 12 ft and 20 ft. Sizes range from ⅛ in. through 3 in. Standard, industry-approved color coding identifies the various types of tube.

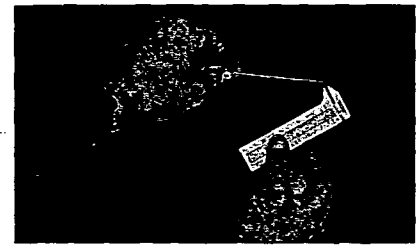
TUBE SEAL



CALUMET & HECLA, INC.
CALUMET DIVISION
URANIUM DIVISION
GOODMAN LUMBER DIVISION
WOLVERINE TUBE DIVISION
In Canada:
CALUMET & HECLA OF CANADA LIMITED
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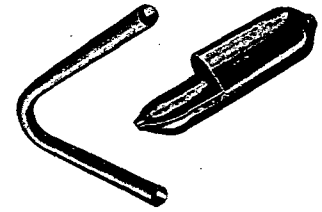
WOLVERINE CAPILATOR•

Wolverine copper Capilator tube is designed for precision control in the metering of gases, liquids or air. Tiny, plug-drawn Capilator has a smooth, mirror-bright bore held to exceptionally close limits. It is available in sizes from .026" to .090" inside diameter. Capilator is washed, deburred, flow-tested and its ends are paper wrapped to insure absolute cleanliness. For general metering or control purposes, Wolverine also manufactures rigidly quality controlled small diameter copper tubing, .062" O.D. and larger.

WOLVERINE TRUFIN•

Wolverine Trufin is an extended surface tube with fins extruded directly from the tube wall. Because of this, Trufin provides increased heat transfer surface—extracts more Btu's per foot of tube. Trufin's integral fins also resist vibration, temperature changes, and fluctuating pressures. Trufin is available in five different types in a wide range of sizes, alloys, and fin spacings—in copper, aluminum, and seamless steel. Illustrated is Trufin Type H/R—a high finned tube ideally suited for air cooled heat transfer operations.





FABRICATED TUBULAR PARTS

Through its Copper† and Aluminum‡ Spun End Processes Wolverine is able to produce one-piece, tubular-shaped parts (below) with a wide variety of contours and end treatments in one fast, economical operation. Other Wolverine fabrication techniques include finning, bending (such as the hairpin return bend at left above), coiling, flaring, expanding—to name but a few.



ALUMINUM TUBE AND SHAPES

Wolverine produces strong, lightweight aluminum tube—plain or finned—drawn or extruded—in a wide variety of sizes in popular alloys. Wolverine extruded aluminum shapes are rigidly quality controlled to meet customer specifications—can help save machining and assembly time and reduce material requirements.

FIELD ENGINEERING SERVICE

For the convenience of our customers Wolverine maintains its Field Engineering Service. Highly skilled tubing technicians are ready at all times to help you solve problems in design, corrosion or fabrication.

† Patented Process RE 22465.

‡ Process Patent Applied For.

Copper & Brass Research Association

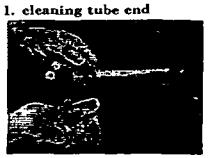
420 Lexington Avenue, New York 17, N. Y.

Soldering Joints in Copper Tube

The two most common types of solder used in joining copper tube are 50-50 tin-lead solder and 95-5 tin-antimony solder. 50-50 solder is generally used for moderate pressures with temperatures up to 250° F. For all temperatures up to 250° F where higher strengths are required, the 95-5 solder should be used. In general, 95-5 solder melts at a higher temperature than does 50-50, has less of a pasty range, and is therefore somewhat more difficult to handle, particularly for a vertical joint to be filled upward.

The fluxes best suited to the 50-50 and 95-5 solders are in paste form and consist of a petrolatum-base, impregnated with zinc and ammonium chlorides. It should not be the function of the flux to clean the copper, but assuming that the copper has been cleaned the flux may be reasonably expected to remove residues of oxide, in addition to protecting the surfaces from oxidation during the heating process. The flux also floats out the remaining oxides ahead of the molten solder and promotes wetting action of the solder. In general, paste-type fluxes are recommended rather than liquid fluxes.

1. cleaning tube end



2. cleaning fitting socket



3. fluxing tube end



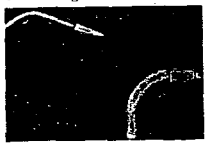
4. fluxing fitting socket



5. assembly



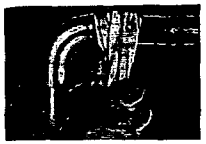
6. heating



7. applying solder

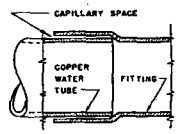


8. removing surplus solder and flux



It cannot be too strongly emphasized that for strong, tight, long lasting joints, thorough cleaning of the tube *surface* and the inside of the cup of the fitting is of utmost importance. Pretinning is not necessary and generally is not recommended if the metal is properly cleaned and fluxed.

Fittings that are oversize will result in a loose fit. The capillary action is dependent on a fairly tight fit although a certain amount of looseness can be tolerated. Looseness is more apt to cause difficulties in working with the larger size copper tube, especially for horizontal joints where an *oversize fitting* may allow the tube to rest directly on the socket, resulting in a double width space at the top. Wherever possible, such joints should be made up in a vertical position, before assembly. For best results always use fittings conforming to accepted standards.



Making up a joint

There are six simple steps that, when followed, will result in a well-made joint. These steps should be taken after first measuring the tube to proper length so that it will run the full length of the socket of the fitting, cutting the tube end squarely and removal of all small burrs.

1. Clean tube end and socket of fitting.
3. Apply flux to the clean areas.
3. Assemble.
4. Apply heat and solder.
5. Remove residual solder and flux.
6. Allow joint to cool.

Safe Strength of Soldered Joints

Pressure Ratings

Solder used in Joints	Maximum Service Pressure, PSI — Water†			
	Service Temperatures Deg. F	¼ to 1 in. Incl.*	1¼ to 2 in. Incl.*	2½ to 4 in. Incl.*
50-50 Tin-Lead**	100	200	175	150
	150	150	125	100
	200	100	90	75
	250	85	75	50
95-5 Tin-Antimony	100	500	400	300
	150	400	350	275
	200	300	250	200
	250	200	175	150
Brazing Filler Metal melting at or above 1000 F††	250	300	210	170
	350	270	190	155

*Standard copper water tube sizes
**ASTM B32, Alloy grade 50A
†Including refrigerants and other non-corrosive liquids and gases
††ASTM B260, Brazing Filler Metal

Air Devices Inc.

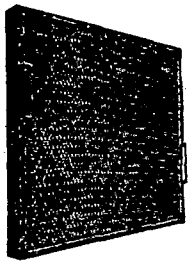
Air Diffusers • Registers • Grilles • Exhausters • Air Filters • High Velocity Mixing Boxes • Punkah Louvres • Filter Holding Frames • Industrial Furnaces

185 Madison Ave.
New York 16, N. Y.

AGITAIR

Agents in All Principal Cities

AGITAIR AIR FILTERS



Type FM
HIGH VELOCITY
•
ALL METAL
•
PERMANENT
•
CLEANABLE

HOW IT WORKS

High turbulence of many finely divided air streams is the keynote of Type FM air filters. The media divides the air into countless fine streams and throws those streams into violent cyclonic turbulence against countless viscous-coated "wiping surfaces" which virtually scrub the air clean by catching and holding the dirt.

HIGH VELOCITY

The Agitair FM Filter is designed to perform at highest efficiency at an approach velocity of 432 fpm—or 1200 cfm through a 20 x 20 in. filter panel.

1/3 LESS SPACE REQUIRED

The ability of the FM to filter, with greater efficiency, 50 percent more air at the high velocity of 432 fpm reduces the number of filter panels required. Now TWO FM's will do the work of THREE ordinary filters . . . ⅓ less space required . . . fewer units to be installed . . . fewer units to be serviced . . . overall installation and maintenance costs reduced to a minimum.

HIGH EFFICIENCY

At the recommended velocities, the Agitair FM has a high dust arresting efficiency, which increases as the dust load is applied.

LOWER RESISTANCE

The sustained low resistance of the Agitair FM means sustained peak volume of air for longer periods of time . . . no loss in air volume . . . no danger of unloading . . . clean filtered air at all times.

HIGHER DUST HOLDING CAPACITY

Employing a new formula for air filtration the new Agitair FM holds more dirt, from two to six times as much as ordinary 2 in. permanent, cleanable filters. No early clogging of air passages . . . less frequent servicing . . . lower maintenance cost.

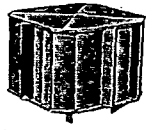
TWO TYPES OF HOLDING FRAMES

Individual type: Designed and constructed for easy handling in single unit installations, and to facilitate "on the job" assembly, into a multiple unit bank. **Pre-Fabricated Type:** Made of heavy gage steel and delivered completely knocked down. They can be quickly and easily bolted together to form a sturdy panel bank.

GREASE FILTERS

Strong Expanded Metal Media recommended for use in kitchens where grease-ladened air is a fire hazard and a maintenance problem. Prevent grease from entering exhaust ducts, eliminate frequent duct cleaning, protect fans, greatly reduce fire hazard, and help maintain good ventilation.

AGITAIR WIND-ACTUATED EXHAUSTERS



Type CNO

Provide proper ventilation regardless of wind direction, and with positive elimination of down-draft. Functions at peak efficiency at average low wind velocities. Will not restrict the exhaust of air or gases when there is no movement of outdoor air across the head.

AIR FILTER CORPORATION

4554G West Woolworth Ave. Milwaukee 18, Wis.

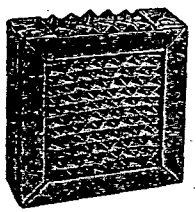
Douglas Engineering Co., Ltd. Canadian Representative Toronto, Ont.

AIRSAN

Reg. U. S. Pat. Off

- **AIR FILTERS**
- **GREASE FILTERS**

Permanent—Cleanable

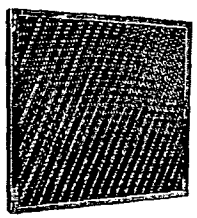


Airsan

VIRO-CRIMP FILTER
(Type W)
Pat: No. 2,653,676

A specially designed high velocity filter media, viscous type. Constructed of horizontal layers of galvanized wire mesh assuring a large filter-area with no appreciable pressure drop. Designed for velocities of 300-600 fpm at minimum resistance. Viro-Crimp edges are hemmed, an important safety feature that provides a smooth media surface.

Type W, Airsan High Velocity Air Filter is of all galvanized metal construction throughout. Drain slots provided to hasten drying and aid in cleaning. Available in all standard and special sizes (2 in. and 4 in.). *Bulletin W802.*



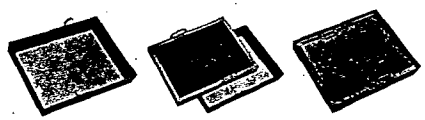
Airsan

AIR FILTERS
Low Velocity
(Type AF-1, AF-2)
Domestic Industrial Commercial

Expanded metal face plate acts as a lint arrestor to provide easier cleaning and servicing. It distributes air evenly over entire filter area providing high filtering efficiency and dust-holding capacity with low resistance. Media is viscous type, permanent, cleanable, and is constructed of multiple layers of galvanized wire mesh. Galvanized steel frames and drain slots for quicker easier cleaning. Available in standard 1 in. and 2 in. thickness. *Bulletin 302.* Also HEAVY DUTY filters (Type AD-2, AD-4) for industrial and special applications—2 in. to 4 in. thickness. *Bulletin 402.*

Airsan

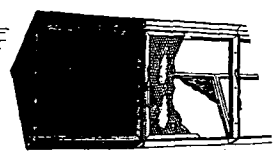
FILTER RETAINER
(Holding Frames)



For Air and Grease Filters. Features include positive snap locks, fire resistant felt seals etc. Particularly flexible to any type of installation of flat or "V" banks. Easily built up to any height or width by progressive addition of Airsan Filter Retainers. Available in 2 or 4 in. sizes. *Bulletin L605.*

Airsan

GREASE FILTERS



Permanent cleanable type Airsan Grease Filters are specially designed for range canopies, galleys, and kitchens.

Removes grease at source, reduces fire hazard in exhaust ducts, and prolongs life of fans, motors. Assemblies for mounting on ceiling or wall, single or multiple units—include holding frames, supporting angles and end seals. *Bulletin L604.*

Write AIR FILTER Corp. for Complete Bulletins

AIR FILTER CORPORATION-4554G W. WOOLWORTH AVE. MILWAUKEE 18, WIS.

Airsan

AIR-MAZE CORPORATION

25000 Miles Rd., Cleveland 28, Ohio

THE FILTER ENGINEERS

Representatives in all principal cities

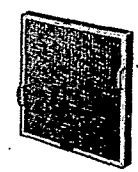
Because filtering air is still an art rather than an absolute science, proven experience itself is even more important in this field than in other technical areas. We at Air-Maze have specialized entirely in the art of filtration for well over 30 years. As a result, there are few filtering problems that we have not encountered during this time. Through first hand experience, we have learned that what not to do is a highly important factor in successfully engineering a filter for a particular job.

There is a saying in the filter business that, "In most cases one never knows he has the wrong filter until it's too late". Consequently, as a buyer or specifier of filters, our years of proven experience, we believe, are invaluable to you.

The prime function of our application engineers is to guide you in selecting the right filter . . . even though it means suggesting an item we do not make, as is sometimes the case. By experience, we recognize that future business depends on your confidence in us. So we endeavor to justify your complete confidence in every recommendation that Air-Maze makes.

A fully staffed organization, backed by adequate plant facilities with a well-equipped testing laboratory, have helped Air-Maze become one of the largest filter manufacturers in the country.

Send for catalog sheets on specific products. General catalog also available on request.



AIR FILTER PANELS

Available in metal washable, viscous impingement types, and dry disposable types. Dry filters also available with washable strainer-type filter element.

ELECTROMAZE ELECTRONIC AIR FILTER



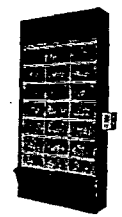
The Electromaze 2-stage electronic air cleaner performs an air cleaning function using the electrostatic phenomenon that no other commercial device can do.

- It is one of the two practical methods for removing particles of sub-micronic size. These particles include smoke, fumes, and carbonaceous matter which authorities agree is the principal cause of discoloration.
- The electronic air cleaner does this air cleaning job with extremely low pressure loss and high dirt-holding capacity. For all practical purposes the pressure drop is so low as to be negligible.
- Its operating costs and maintenance are unusually low.

Electromaze employs high-voltage direct current to ionize the air (and entrained dirt contaminant therein) by means of a corona discharge. Ionized particles are readily collected on plates having opposite positive and negative charges on alternate plates.

Has semi or completely automatic vertical spray washing system. Removes particles as small as 0.1 micron diameter, has rated efficiency over 95 percent, as tested by the *National Bureau of Standards'* Discoloration Method.

AUTOMAZE AUTOMATIC AIR FILTER



Where continuous operation without servicing is required, the Automaze "pulse action" automatic self-cleaning filter should be used. It combines a double filter panel curtain with a positive effective panel cleaning action to provide a better viscous impingement air filtration with a minimum of maintenance. New self contained drive mechanism and unitized control system simplifies installation. Standard Automaze filter media is satisfactory for most installations, but other media types are available for special applications.

ACCESSORIES

Cleaning tanks, holding frames, complete assemblies for range canopies, viscous adhesive coatings, spray guns, etc., are available for complete installation and service requirements.

American Air Filter Company, Inc.

673 Central Avenue, Louisville 8, Ky.

AMERICAN AIR FILTER OF CANADA, LTD., MONTREAL, P. Q.

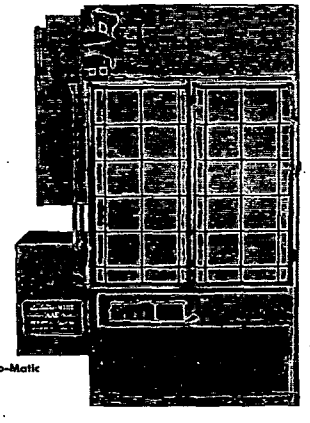
THE COMPANY: The American Air Filter Company, Inc., is recognized internationally as an authority on air filtration and dust control. In 30 years, its leadership and scientific "know how" have been responsible for installations of AAF equipment in every industrialized part of the world. Because its continuous research, high engineering standards and exclusive specialization have been consistently maintained, AAF equipment is used by leading companies in nearly every industry and is specified by leading architects and engineers for use in commercial and industrial air conditioning.

The American Air Filter Company, Inc., manufactures a complete line of air filtering and dust control equipment. A few representative types are shown here. AAF's years of experience and the scope of its knowledge dealing with the elimination of air-borne particles in every form insures maximum efficiency in application, design, manufacture and installation . . . with a minimum of field engineering. Write AAF on all problems of air filtration and dust control, or its Herman Nelson Division on heating or ventilating problems.

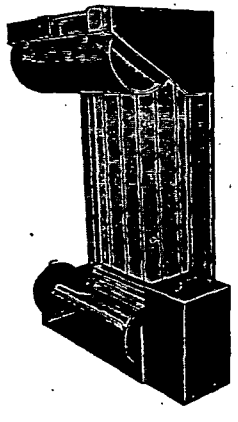
AAF AIR CLEANING PRODUCTS

Electro-Matic Self-Cleaning Electronic Precipitator—The Electro-Matic Precipitator is an automatic electronic precipitator combining advanced principles of electronic air cleaning with a special self-cleaning principle. It eliminates the necessity of shutting down the filter for manual cleaning, minimizes the need for personal attention and permits continuous high-efficiency operation. It also allows the Electro-Matic filter to be built in standardized self-contained sections, easy to install and with all exposed parts of the filter casing electrically grounded for the protection of operating personnel. Send for *Bulletin No. 250.*

ROLL-O-MATIC Renewable Media Filter—An automatic air filter featuring a low cost, renewable media filtering curtain of bonded glass fibers supplied in convenient roll form. The media, mounted at top of filter, moves down the face of the ROLL-O-MATIC and is re-rolled on a driven spool at bottom. Introduction of clean material to maintain desired operating resistance is controlled by an automatic time switch. ROLL-O-MATIC's maintenance-free operation provides continuous, high efficiency cleaning at less than half the operating cost of a disposable type filter of equal capacity. Send for *Bulletin 248.*



Electro-Matic

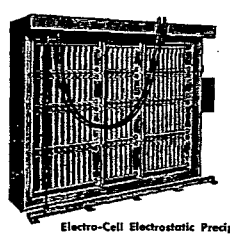


Roll-O-Matic

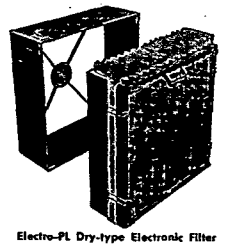
American Air Filter Company, Inc.

673 Central Avenue, Louisville 8, Ky.

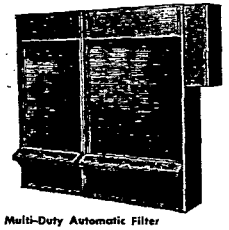
AMERICAN AIR FILTER OF CANADA, LTD., MONTREAL, P. Q.



Electro-Cell Electrostatic Precipitator



Electro-PL Dry-type Electronic Filter



Multi-Duty Automatic Filter

The Electro-Cell is a stationary plate-type electronic precipitator available in both conventional and high-velocity designs for application on a wide range of dirt concentrations. Cell construction assures simplified installation and ease of maintenance. Collector assemblies may be removed for individual cleaning—washed in place manually—or, with the addition of Type "H" Washer, cleaned automatically by traveling water sprays. Write for *Bulletin No. 252.*

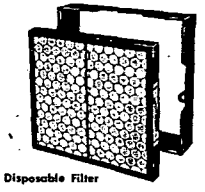
Electro-PL—A special dry-type electronic air filter with charged Airmat collector element which combines air filtration and electronic precipitation in a single unit. Gives intermediate efficiency at lower cost, when the need for superclean air is not indicated. However, it has twice the efficiency of uncharged dry-type filters. Available in Straight Bank or "V" arrangement to meet virtually any space or capacity requirement. Send for *Engineering Bulletin No. 257.*

Multi-Duty Automatic Filter provides outstanding features of performance and design and will accommodate either armored screen panels or die stamped louvre panels available in three types. Offers advantage of uniformly constant air supply, fixed operating resistance and automatic operation. Ideal for ventilation and air conditioning service. Available in any size or capacity. Send for *Bulletin No. 241-A.*

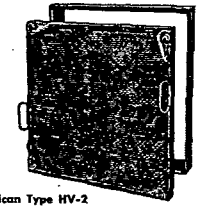
AMER-glas Replaceable Unit Filters. A new viscous impingement type for eliminating atmospheric dust from forced air heating systems. The highly efficient filtering media consists of continuous, curled and interlaced, extremely fine glass filaments, bonded with thermoplastic to form a thick resilient pad. The pad is sprayed with a special Viscosine and placed in a fiberboard casing between perforated metal grilles. The AMER-glas is nonflammable, sanitary and odorless and the Viscosine remains in a fluid-jell state for the life of the filter. Available in 15 sizes. Write for *Bulletin No. 211-A.*

Airmat Type PL-24—Airmat filters use standard Airmat medium, renewable after collecting dust load. Used both for comfort and industrial air conditioning. Available with unit frames to be set up to meet any capacity requirement or space condition. Send for *Bulletin No. 230.*

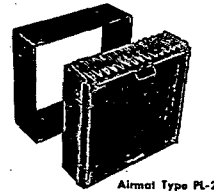
Type HV-2 Filter: A high capacity, low-resistance unit designed for velocities up to 500 fpm. Special pyramid pocket media design eliminates through-air passages and gives uniformly high efficiency over wide range of air velocities. HV-2 has large dust capacity, long life, and decided advantages where space is limited. Available in three designs. Write for *Engineering Bulletin No. 203.*



AMER-glas Disposable Filter



American Type HV-2



Airmat Type PL-24

American Air Filter Company, Inc.

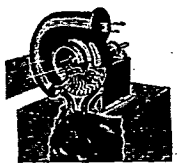
673 Central Avenue, Louisville 8, Ky.

AMERICAN AIR FILTER OF CANADA, LTD., MONTREAL, P. Q.

AAF DUST CONTROL EQUIPMENT

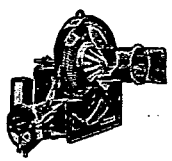
All AAF Dust Control equipment—dynamic precipitators, hydrostatic precipitators, dry centrifugals and fabric arresters—combine the functions of exhausting, separating and storing dust in one simple, compact, self-contained unit. All have these basic design features: (1) small space requirements, (2) maintained performance over a wide range of operating conditions, (3) constant exhaust volume, and (4) high collection efficiency. Available in a wide variety of types and sizes, AAF equipment has a proven record of economy and efficiency covering thousands of production line and individual applications. Write for profusely illustrated 35-page application and *Engineering Bulletin No. 274-A.*

Type D ROTO-CLONE —A dynamic precipitator designed for dry collection of granular industrial process dust. Combines functions of exhauster, dust separator and storage facilities in one compact unit. High collection efficiency remains constant over entire pressure volume range. Adapted for individual unit or central system use. Capacities — from 100 to 15,000 cfm. Write for *Bulletin No. 274.*



Type D ROTO-CLONE

Type W ROTO-CLONE offers high dust separation efficiency where extreme fines and heavy dust loads are involved. Combines dynamic precipitation with integral water sprays. Delivers a constant air volume and collects dust as a sludge. Compact design makes for easy, low-cost installation. Capacities from 1,000 to 50,000 cfm. Write for *Bulletin No. 274.*



Type W ROTO-CLONE

Type N ROTO-CLONE is a hydrostatic collector, with induced water scrubber. Requires minimum space—easy to install. No moving parts, spray nozzles or pumps. Completely self-contained with dust discharged as a sludge. Capacities—from 1,000 to 50,000 cfm. Write for *Bulletin No. 277.*



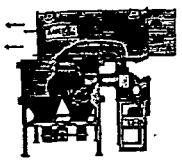
Type N ROTO-CLONE

AMERjet—A reverse jet type fabric arrester offering high collection efficiency, constant pressure drop (constant air volume), compact design and continuous operation. Other features are (1) unique method for pressure air supply, (2) rugged mechanism for dislodging collected material with high pressure air jets and (3) venturi-shaped inlet for dust-laden air which extends life of cloth. Available in range of capacities. *Bulletin No. 279.*



AMERjet

AMERclone —A dry centrifugal dust collector designed to handle large exhaust volumes containing dust in high concentrations. Unit requires small space due to compact design and high cleaning capacity of the AMERclone tubes. Abrasion resistant construction featured throughout. Collection efficiency remains practically constant over wide range of exhaust volumes. Available in wide range of capacities. Write for *Bulletin No. 291.*



AMERclone

BARNEBEY-CHENEY

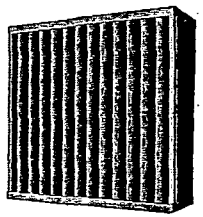
Cassady at Eighth, Columbus 19, Ohio

Los Angeles • San Antonio • St Johns, Quebec

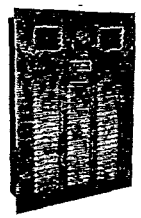
Barnebey Cheney

Activated charcoal purification equipment

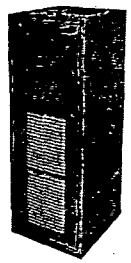
Barnebey-Cheney manufactures activated charcoal and purification equipment for air, water and chemical products. Used in air purification, AC saves on heating-cooling costs by permitting recirculation of air. AC adsorbs odors, irritating vapors, smoke and smog. Completely revitalizes stale or stuffy air.



FilterFold adsorbers Pleated filter with ½ in. bed of charcoal. Installed in multiples, these cells give large air handling capacity in a small space. Each cell will handle up to 1,000 cfm at 0.2 wg. Heavy guage steel frames take rough treatment (stainless frames available). Five types available.

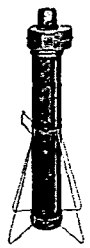


Wall unit Flush mounted circulator for semi-light duty use. Contains centrifugal fan, charcoal filter, and dust filter. Steel cover plate furnished with prime coat. Designed to fit between studding. Capacity: 150 cfm.



Cabinet circulators Complete units contain panel adsorbers, dust filter, centrifugal blower, and motor assembly. They are fully assembled, ready to plug in. Designed for long service and quiet operation 24 hours a day. They will purify up to 12,000 cubic feet of space.

Heavy duty units are available for warehouse or industrial operation. These may be ceiling mounted or portable.



Canister circulators Complete unit contains removable canister and blower-motor assembly. Designed for small rooms and enclosed areas. Four models, with capacities from 15 to 120 cfm. Either floor stands or wall mounting available.



Panel adsorbers Flat perforated metal cell with 1 in. bed of activated charcoal. Use it when the contaminant load is heavy and you want long service life. Frame assemblies and panels are available for any air flow. Capacities up to 180 cfm at 0.3 wg. Three models.

Canister adsorbers Cylindrical perforated metal with ⅜ in. to ¾ in. bed of charcoal. Use these for small jobs and for expansion of existing installations. They are mounted in manifold plates in multiples, depending on cfm, and are interchangeable with other types of canisters. Four models.

Other products and services Disposable charcoal filters in paperboard frames are available for use in air conditioners, forced air furnaces, and ventilating systems. Breeze circulators are attractive units for homes, offices, clubs.

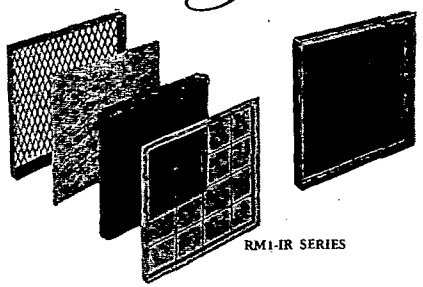
Barnebey-Cheney's complete line of activated charcoals is available in bulk.

Ask for free literature.

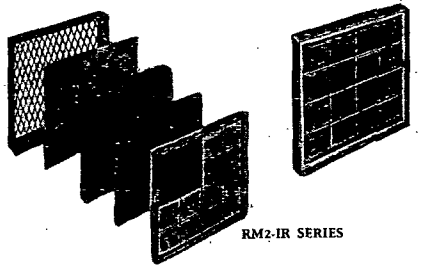
BURKE AND COMPANY

2902 Hyde Park Boulevard, Los Angeles 43, California

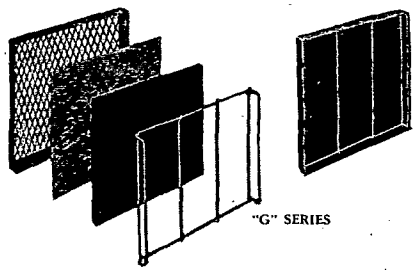
ROTO*Aire*® REPLACEABLE MEDIA AIR FILTERS GAS AND VAPOR ADSORBERS



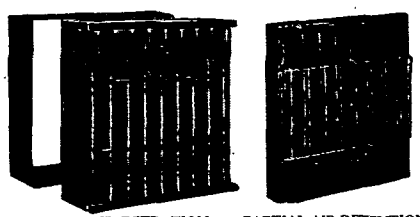
RM1-IR SERIES



RM2-IR SERIES



"G" SERIES



TOTAL AIR DETENTION PARTIAL AIR DETENTION

Roto Aire RM1-IR Series Impingement-Strainer Type Replaceable Media Air Filters with Integral Retainer

The Roto Aire RM1-IR Series impingement-strainer type replaceable media air filter with integral retainer is the "advanced design" air filter which combines the filter holding device and the filter itself. It is comprised of a heavy gauge retaining frame which is faced with a sheet of 1" diamond, flattened, expanded steel and equipped with filter case centering dimples and spring loaded positive filter media case face-locking devices. The filter media case is likewise constructed of heavy gauge steel and is faced with a 4" mesh 12 gauge welded steel rod grid. The spring loaded positive filter media case face locks permit the installation of filters on any angle and press the filter case against the retainer so securely that by-pass of dirt laden air is eliminated. The filtering medium is glass fiber and consists of one filter pad that has a *bulk thickness* of 2" and one strainer mat that is .025" in thickness. Each medium *is treated with Type 201 non-inflammable* binder, and the 2" filter medium is treated with TCP (Tri-Cresyl Phosphate adhesive), the surface chemistry of which is cationic.

Roto Aire RM2-IR Series Impingement-Strainer Type Replaceable Media Air Filters with Integral Retainer

Roto Aire RM2-IR Series air filters differ from the RM1-IR Series to the extent that the filter media case is 2" in depth to accommodate three distinct types of filtering media namely large particle preload paper in addition to the media employed in the RM1-IR Series. These media perform the various functions of *retaining airborne* substances over the wide range from lint collection to retention of 0.5 micron particles. Greater economies in operation are to be realized because of their high dirt holding capacity and low face loading characteristics.

Roto Aire G Series Impingement-Strainer Type Replaceable Media Air Filter with Integral Retainer

The Roto Aire G Series air filter is comprised of a retainer, a gate lock grid and two types of fiber glass filter media that combine to form an integral filter cell. The *retainer consists* of an "L" *type* welded frame that is faced with flattened expanded steel to form a case for the filtering medium. Clips for hinging and locking the grid are also provided in the retainer. The grid is fabricated from steel wire and shaped so that when opened it sets off to permit easy installation *of replacement filter media.* Like the Roto Aire I.R. Filter, the G Series filter may be installed at any angle, and the grid, when locked in place, presses the filter media against the retainer so securely that by-pass of dirt laden air is eliminated. The filtering medium is glass fiber and consists of one filter pad *that has a bulk thickness of 2" and one* strainer mat that is .025" in thickness.

Roto Aire Gas and Vapor Adsorbers

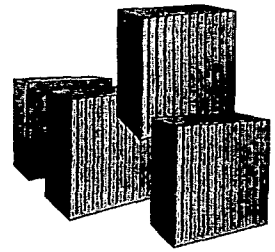
ROTO AIRE GAS AND VAPOR ADSORBERS increase comfort and physical efficiency because they purify air by removing noxious gases and vapors. The air purifying agent in Roto Aire Gas and Vapor Adsorbers is granular carbon that has been processed at high temperature to expel hydrocarbonic impurities and increase its porosity. The surface of carbon granules so refined becomes super active and contains the remarkable power of arresting noxious gases and vapors from ventilation and recirculated air. This phenomenon is known as adsorption and is easily likened to flies clinging to fly paper. ROTO AIRE GAS AND VAPOR ADSORBERS make it possible to greatly reduce, or even eliminate, con*taminated outside air for ventilation purposes. Air conditioning* installation and operational expenses therefore decrease because there is no need to heat or cool this additional air. There are no anoxia concerns because unavoidable infiltration factors exceed by many times the air volume necessary for oxygen replenishment. Partial air detention type Roto Aire Adsorbers are designed to process air at high efficiency on a by-pass basis. Therefore only a part of the air passes through the carbon cartridges where up to 25% of air borne contaminants are removed. Their use is recommended to process ventilation and recirculated air when ventilation air requirements are less than 10% of the total air volume. They are also recommended for use in systems where outside air is excluded and the total quantity of supply air is recirculated.

CAMBRIDGE FILTER CORPORATION

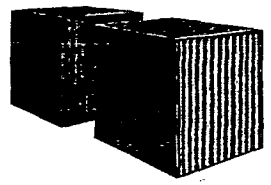
736 EAST ERIE BOULEVARD, SYRACUSE 1, N. Y.

Cambridge®

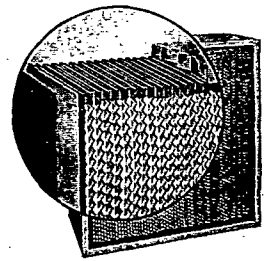
Representatives in Principal Cities



Cambridge AEROSOLVE® Filter
Efficiencies 35% to 95%
Discoloration Test, Atmos. Dust



Cambridge High-Velocity
AEROSOLVE® Filter



Cambridge ABSOLUTE® Filter
Guaranteed 99.97% Efficient on 0.3
Micron Particles

High-Efficiency Air Filters

Cambridge air filters are positive, strainer-type, high-efficiency filters which provide trouble-free air cleaning without electric, water or drain connections. They are economically installed and, except for infrequent replacement, require no maintenance. In relation to efficiency, initial pressure drops are exceptionally low.

Cambridge AEROSOLVE® Filters

Consist of permanent cadmium-plated steel frames with easily replaceable and interchangeable cartridges available in efficiencies of 35 percent, 85 percent and 95 percent efficiency (discoloration test with atmospheric dust). This test was developed by the National Bureau of Standards and is used for rating electric precipitators. Most commercial filters test 5 percent to 15 percent by this method. Changes in efficiency requirement are easily met without major change in the system simply by substituting cartridges. Cartridges slide easily into frames and are secured by quick-acting fasteners. Integral prefilters are available where desired.

Standard units are designed for face velocity of 250 fpm and can be used at higher velocities by V-ing the filter bank. High-Velocity units, face velocity 450 fpm, can be installed in the plenum in a straight bank, for even greater economies.

	High Velocity	Standard	
Face Area	24" × 24"	24" × 24"	24" × 12¾"
Rated Face Velocity	450 fpm	250 fpm	250 fpm
Rated Capacity	1800 cfm	1000 cfm	500 cfm
Area, Filter Medium	86 sq ft.	43 sq ft.	22 sq ft.

Low Pressure Drop

In relation to efficiencies, the initial static pressure drop of AEROSOLVE Filters is exceptionally low:

	AEROSOLVE 95	AEROSOLVE 85	AEROSOLVE 35
Average Discoloration Efficiency	*95 percent*	*85 percent*	*35 percent*
Initial Δp (Standard)	0.35" w.g.	0.22" w.g.	0.16" w.g.
Initial Δp (High-Vel.)	0.45" w.g.	0.32" w.g.	0.25" w.g.
Final Δp (Both Types)	0.80" w.g.	0.60" w.g.	0.60" w.g.

The great area and thickness of AEROSOLVE filter cartridges provide long life which, with low initial cost, guarantees economical owning and operating costs.

Cambridge ABSOLUTE® Filters

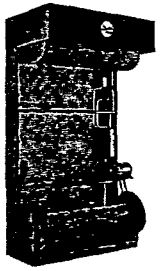
New *glass-asbestos filter medium* now used in ABSOLUTE Filters is an improvement both in efficiency and in air-handling capacity over the medium which Cambridge originated for removal of *radioactive particles from exhaust systems in AEC* installations. 99.97 percent removal of 0.3 micron particles is guaranteed. Essentially removes all smoke, dust, bacteria and *mold spores and is widely used by industry* in applications where the highest degree of air cleaning is required. Every filter individually tested. Maximum penetration of 0.03 percent guaranteed. Initial pressure drop *1.0 in. w.g.* at rated air flow. Each filter self-supporting—no permanent frame required. Easily installed. Standard size units from 30 cfm to 1375 cfm in stock for immediate shipment. Special sizes and shapes available on order. 1100 cfm units 24 in. × 24 in. × 11½ in.; filter area over 200 sq ft. Standard A-series withstands 220F and 85 percent RH. Other series available for extremes of temperature, humidity, fire resistance and corrosion.

Table of Sizes

Model	Size, in.	Rated Capacity CFM at 1" wg.
1–25	8 × 8 × 3¹⁵⁄₁₆	30
1–50	8 × 8 × 5¾	50
1–600	24 × 24 × 5¾	650
1–1000	24 × 24 × 11½	1100
1–1250	24 × 30 × 11½	1375

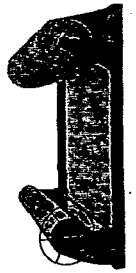
CONTINENTAL AIR FILTERS, Inc.

P.O. BOX 1647 — LOUISVILLE, KY.



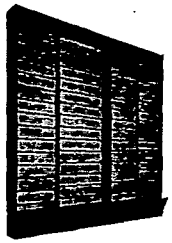
CONTINENTAL CONOMATIC Roll-Away Type Air Filters... provide maximum filter area for filter size. Automatically advances viscous coated, glass-fiber filter media as needed. **Media Saver** minimizes cost. Trouble-free direct drive saves maintenance.

Write for *Bulletin No. 802.*



CONTINENTAL CONOMANUAL ... manually operated dry-type filter similar to Conomatic, for use where low installation and maintenance cost limits preclude automatic equipment. Simple hand-wheel advances ½-inch-thick disposable synthetic media blanket which has high dust-holding capacity, is moisture-, fungus-, and fire-resistant and will not part or "shed" when handled.

Write for *Bulletin No. 850.*



CONTINENTAL ANTOMATIC, SELF-CLEANING Air Filters ... viscous impingement type. **Features:** Patented E-Z-Wash media, "Ferris Wheel" action of continuous rotating filter curtain, and positive automatic flush-cleaning of curtain in each rotation cycle.

Ferris Wheel action passes filter sections from front to rear curtain **without reversing** them to the direction of air-flow ... thus eliminating problem of trapped dirt being blown into clean air stream.

Cam action tilts and lowers each filter section while in oil reservoir so as to forcefully flush filter oil through the media **counter to air-flow.**

Write for *bulletin No. 810.*



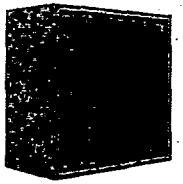
CONTINENTAL DYCON Dry-Type Air Filters ... have excellent dust-holding capacity. Efficient media blanket of specially prepared synthetic fiber. **Low maintenance cost:** quickly and easily restored to original resistance by cold-water rinsing. No oil involved. Standard sizes. Type CA-24, 8 in. thick. Type CA-20, 4 in. thick. Type DY, 1 in. thick.

Write for *Bulletins Nos. 460 and 670.*



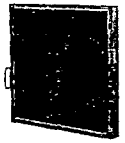
CONTINENTAL E-Z-WASH Unit Filters ... use same advantageous patented media as discussed above. Can be **washed easily** by brisk cold water hosing, and quickly re-oiled with E-Z-Oil spray bomb or dip tank. Standard sizes, 2 in. or 4 in. thick.

Write for *Bulletin No. 425.*



CONTINENTAL TYPE LV Air Filter... crimped galvanized hardware cloth and screen wire in highly rust-resistant, hot-dipped galvanized steel-channel frame assure long service-life for the LV viscous impingement air filter. Bale handles optional.

Write for *Bulletin No. 600.*



CLEERALL Air Filters ... to approach **totally dust-free** conditions ... provide efficiency in excess of 99.97 per cent on 0.3-micron particles as tested by DOP standard procedure required by *Atomic Energy Commission.*

Available in standard sizes ranging from 30 to 1200 cfm rated capacities at 0.9 in. WG. Size 24 in. x 24 in. x 11½ in. has 1000 cfm capacity.

Three standard types meet operating conditions in temperatures up to 1000F or high humidities. Special filters for use up to 2300F or in highly corrosive fumes also available.

Write for *Bulletin No. 900.*

Dollinger Corporation

6 CENTRE PARK • ROCHESTER 3, N. Y.

AIR PROTECTOMOTOR — STAYNEW FILTERS

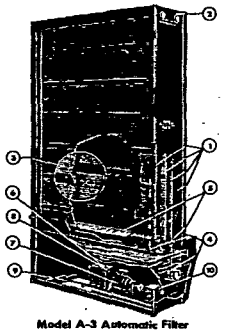
Filters for Building Ventilation, Air Conditioning, Engine Intake, Pipe Lines and Many Other Special Applications

REPRESENTATIVES IN PRINCIPAL CITIES

STAYNEW MODEL A-3 AUTOMATIC FILTER

An endless curtain type oil-bath filter for handling large volumes of heavily dust-laden air at low cost. The efficiency of Staynew Model A-3 is outstanding among mechanical self-cleaning filters.

Patented, exclusive Staynew Compressed Air Curtain Cleaners (5) are available for special conditions.



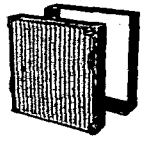
Model A-3 Automatic Filter

SPECIFICATIONS

Model A-3 Filters are sectional and may be bolted together to obtain any required capacity. Sections come in two widths, 4 ft. 3 in. and 2 ft. 9 in. Curtain drive and control mechanism (6) arranged either as an integral part of filter unit or for remote mounting, includes a ¼ hp motor (7) driving through a reduction gear and a momentary contact time switch (8) for testing and checking curtain travel and compressed air control. All are mounted on a common base plate (9) on clean air side of filter. Shear pin (10) is provided for protection of moving parts from accidental damage. Drive Motor operates for a few seconds at 15 minute intervals, and compressed air curtain cleaners can be arranged to operate simultaneously.

Write for Bulletin 500.

STAYNEW PANEL TYPE FILTERS



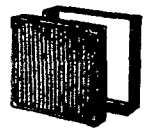
Model WKE Panel and Frame

Model WKE: Dry-type finned panel filter for use in ventilation and air conditioning systems. Extremely large filtering area in relation to overall size. Adaptable to wide variety of filtering media—in fact, almost any medium obtainable in sheet form that can be crimped. Steel mesh on both sides of medium prevents sagging and makes the WKE fire-resistant (models available to meet Class I Fire Underwriters approval), and cleanable without possible damage from vacuum cleaning tool or cleaning nozzle. It may also be washed or dry cleaned when and if necessary. There are no cross bars, spacer bars, or other obstructions to interfere with the cleaning operation. Unaffected by temperature changes.

Filter cells are held in rigid box-type supporting frames of heavy gage metal by spring-loaded cam-type locking latches. Two lifting handles are provided on each cell. Filtering medium supplied already crimped and cut to size. It may be inexpensively replaced in 2 to 5 minutes right at the filter bank—no special tools required.

Frames are drilled so that they can be riveted together to form a flat bank, or by the addition of angle uprights into a "V" or staggered arrangement.

Write for Bulletin 600.



Viscous Model DPV Panel and Frame

Viscous Panel (Model DPV): A permanent type panel for air conditioning systems used in heavy duty industrial service. Filtering media consist of a series of layers of crimped galvanized screen cloth and woven mesh. These media when coated with PD-10 Pingene Filter Oil form an unusually efficient filter. Model DPV filters are cleaned easily with live steam or by washing in a suitable solvent. Spring-loaded locking latches and lifting handles are provided as in Model WKE.

Both Model WKE and DPV cells are furnished in 2 in. and 4 in. depths in various standard sizes.

Write for Bulletin 700.

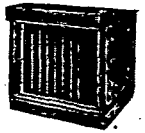
ELECTRONIC FILTERS



1200 CFM Electro-Staynew Air Cleaner

Electro-Staynew Air Cleaner: Ionizer and Collector Cell built into one compact unit. Highly efficient—removes 90 percent of all air-borne dirt, including smoke, dust, pollen, oil mist. Available in single units of 1200 and 1800 cfm; any capacity in multiple units.

Write for Bulletin 400.



1500 CFM HV Electro-Staynew Air Cleaner

HV Electro-Staynew Air Cleaner: Has same features as standard model but requires less space. High velocity design permits 40% reduction in face area for any given capacity. Available in 1000 and 1500 CFM units, any capacity in multiple units.

Face dimensions:

1000 CFM unit—20" x 17"

1500 CFM unit—20" x 23"

Write for data sheet.



Electro-Staynew Mist Collector

Electro-Staynew Mist Collector: Eliminates smoke and oil mist at its source. Mounted on or near high speed machine tools, an Electro-Staynew Mist Collector draws contaminated air from around grinding or cutting operations and returns clean air to shop. A completely self-contained ventilating system for machine tools.

Write for Bulletin 420.

Electro-air Cleaner Company Inc.

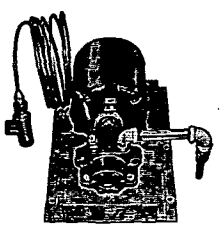
Olivia & Sproul Sts., McKees Rocks, Pa.

"In Greater Pittsburgh" • Representatives in Principal Cities

Electro-air Hi-C, "High Capacity" Electronic Air Cleaners have been developed for ease of installation, maintenance-free operation and high efficiency. The entire line features a self-adjusting, ballasted, single circuit and selenium rectifiers. All parts, including selenium rectifiers are guaranteed for five years.

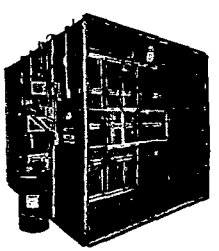
All Electro-air equipment incorporates the patented "Inside Curve" Ionizer which means higher efficiency, reduced size of equipment, increased capacity and face velocity, and service-free operation. All sizes of Electro-air units are available with either vertical traveling manifold or horizontal fixed manifold washing systems.

The latest Electro-air engineering development is the new *Detergent-Adhesive System, Type "X", with "double action" DAX solution* which is available with all sizes of Electro-air equipment. It is supplied as a "package" and is used to promote thorough washing and adhesive application. DAX solution is injected into the wash water during the last ⅓ of the washing cycle, acts as a strong detergent, then dries and leaves an adhesive coating on collecting plates and electrodes.



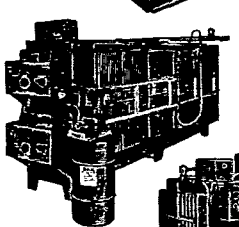
Electro-air Detergent-Adhesive System Type "X

"Built-up Line" equipment is manufactured in capacities from 10,350 cfm to 200,000 cfm for larger building areas. Each unit is made "to order" according to the specified height, width and cleaning capacity required, is assembled and tested at the factory, then match-marked, disassembled and shipped in well marked crates. "Built-up Line" units are also furnished with "enclosures" complete with drain pans, fore and aft ducts, windowed access doors and complete wiring.



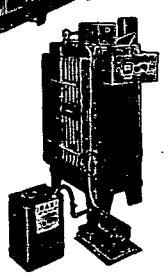
Electro-air "Built-up Line" Enclosure Unit

"Custom Line" units are furnished in capacities from 1,150 cfm to 21,360 cfm as complete "packages" and are designed for use in those cases where space is limited and access doors in ducts are not practical. Each unit is complete with the required number of collecting cells, drain pan, washing system and power pack. "Custom Line" Electro-airs are furnished in either horizontal or vertical (up or down) air flow, and may be floor mounted or suspended.



Electro-air "Custom Line" Unit

"Compact Line" Electro-airs are moderately priced and are designed for easy installation in homes and smaller commercial or industrial buildings which have forced air ventilation. They are available in fifteen models, both horizontal and vertical (down) flow in capacities from 800 to 9,600 cfm. Horizontal flow models may be either floor mounted or suspended. "Compact Line" units include most features of larger "made-to-order" Electro-air models.



Electro-air "Compact Line" Unit

Electro-air Cleaner Company, Inc. also manufactures low cost "Tec Line" units which are true electronic air cleaners (90 percent efficient as determined by the National Bureau of Standards Dust Spot Test) for installation in small and medium size homes.

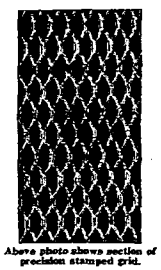
THE GEORGE EVANS CORPORATION

MOLINE, ILLINOIS

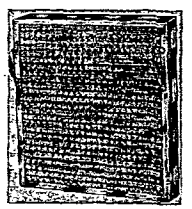
Evans

Evans Lifelong Air and Grease Filters

All Aluminum . . . Every Part Precision Made



Above photo shows section of precision stamped grid.



FILTER PACK HOLDS MORE GRAMS OF DIRT PER CUBIC INCH.

Strands and grids are scientifically sized and placed for holding *maximum dirt load. Filter pack has larger grid openings on* air inlet side progressing to smaller grid openings on air outlet side. Prevents face loading and choke-off of air stream. Accomplishes uniform depth loading all through the pack. Every grid, every strand does its part.

TURBULENCE AIR CLEANING ACTION.

Dirt is slammed into the flat strands of mesh where it is caught and held by talon like edges of these grids. Precision stamped openings force air to travel many times filter depth from inlet to outlet side . . . for high efficiency air cleaning.

LOWER AND SLOWER STATIC PRESSURE RISE PER GRAM OF DIRT LOAD.

Filter pack catches and holds dirt without strangling air stream. Filter remains in service longer: (1) without sharp increase in resistance (2) with fewer cleanings (3) with power costs held down.

HIGH VELOCITY APPLICATION . . . UP TO 1,000 FPM.

Using turbulence air cleaning action (instead of screening or straining action) makes Evans filters particularly well suited to extremely high velocity requirements. The higher the velocity . . . the harder dirt is slammed into the mesh where it is caught and held by the talon like edges of the strands. *Filter's efficiency rises as velocities rise.*

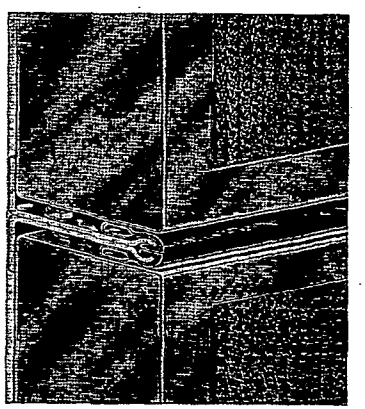
DOUBLE ACTION ADHESIVE.

Works as adhesive while filter is in service. Becomes a detergent and helps wash filter when water is flooded on. IMPORTANT . . . When Evans filter fluid is used filters can be washed with tap water.

FILTER CAPACITY CHART—CFM DELIVERY

(Velocity) fpm	Pressure Loss Inches H2O	cfm Delivery Standard Size Filters 10 × 20	16 × 20	20 × 20	20 × 25	20 × 30	25 × 30
303	.044"	815	848	700	894	1055	1393
345	.054"	860	626	800	1021	1339	1578
390	.069"	466	706	900	1151	1396	1778
432	.077"	430	784	1000	1277	1550	1978
478	.094"	495	862	1100	1404	1704	2171
519	.113"	540	939	1200	1531	1858	2387
563	.132"	588	1019	1300	1661	2016	2567
606	.156"	630	1097	1400	1789	2170	2763
649	.180"	675	1175	1500	1915	2323	2959
693	.206"	721	1254	1600	2044	2481	3160
736	.234"	765	1332	1700	2171	2635	3356
779	.262"	810	1410	1800	2298	2789	3553
823	.291"	856	1490	1900	2428	2946	3753
866	.325"	901	1567	2000	2555	3100	3949

SEAL TIGHT HOLDING FRAME WITH PVC GASKET GIVES PERMANENT DUST TIGHT SEAL.



Gasket pressure locks filter into frame. No springs, latches or locking attachments. Slide filter into frame. Gasket automatically grips it securely.

Gasket is permanent (Polyvinyl Chloride). Stands up for years *against compressive stresses without breaking down.* Unaffected by oil or moisture.

HOW TO SPECIFY EVANS SEAL-TIGHT FRAMES.

Filter holding frames shall be heavy gage steel. Polyvinyl *chloride gasketed* on front of air inlet side. Gasket shall provide both positive grip for holding filter in frame and positive seal between filter and holding frame. Frames shall be double coat finished in black enamel. *Frames shall bolt together.*

HOW TO SPECIFY EVANS LIFELONG AIR FILTERS.

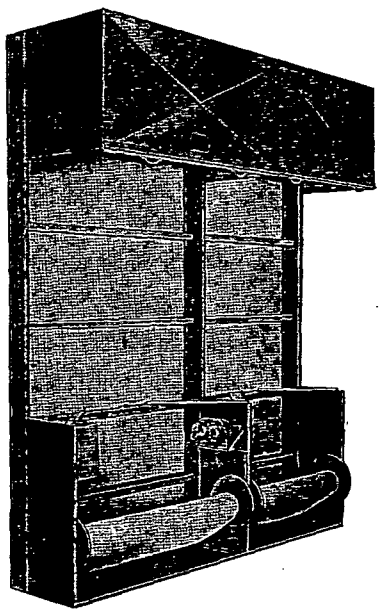
Filters shall be aluminum. Frame holding media shall be one-piece precision die-pressed heavy gage aluminum with edges rounded. *Corners machine fitted. Frame shall have drain holes* in 3 corners of frame member. Filter pack shall be made of die-cut flat and corrugated aluminum grids packed with larger openings on inlet side progressing to *smaller openings on* outlet side.

FARR COMPANY

P. O. Box 90187, Airport Station, Los Angeles, California

NEW YORK LOS ANGELES CHICAGO

AUTOMATIC FILTERS

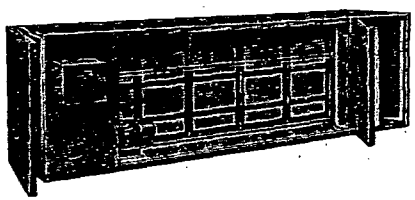


ROLL-KLEEN AUTOMATIC FILTER

A disposable media filter which automatically changes the media in the face of the filter. Designed for Roll-Kleen media, an adhesive coated, glass fiber mesh, bonded to and reinforced by a lenoweave fabric backing. Use of this reinforced media permits a simple, direct drive media movement system. Auto-Control system includes prewired control boxes, electric gear motor and signal lights. Pressure operated media control is standard. Timer or combination control systems also available. Factory assembled in Sections 3, 4, and 5 ft wide and from 5 to 15 ft high. Write for *Bulletin B-1400-2*

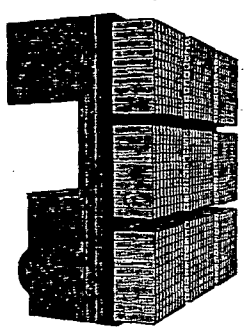
LINT KLEEN

An automatic filter similar to the Roll-Kleen in appearance and operation. Designed for textile mill applications. Uses special, extra long media rolls. Write for *Bulletin B-1400-5*.



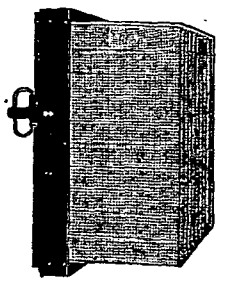
ROLL-KLEEN TYPE H

Incorporates the basic Roll-Kleen features in a compact assembly which can be easily attached to most makes and models of air-handling units. Media moves horizontally across the air stream. The Type H uses standard Roll-Kleen media. Write for *Bulletin B-1400-3*.



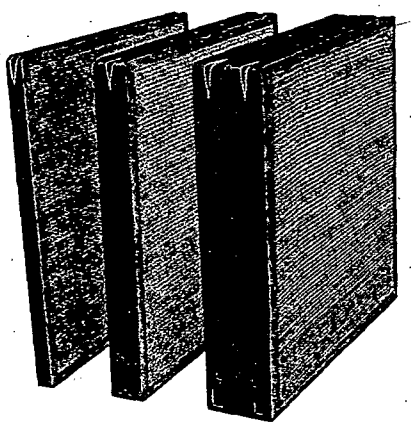
HI-KLEEN

A complete, two-stage, air-filtering system combining the Far-Air Roll-Kleen and HP filters in a compact unit. Total depth is only 38¾ in. Both stages use convenient, completely disposable media. Offers exceptional efficiency, especially in the 0-5 micron range. Write for *Bulletin B-1400-4*.



Far-Air HP FILTER

A deep-pleated, flameproof, disposable filter which is preformed, then folded before shipment for convenient handling and storage. The HP is popped open at the filter bank and easily inserted into a matching metal holding frame which remains permanently in place. After use, the HP will refold trapping all collected dirt in the deep, pocket-like pleats. Filter weighs approximately one pound depending on size. Highly efficient in the small (0-5 micron) particle size range. Available in 16 x 20 in., 20 x 20 in., 20 x 25 in., and 24 x 24 in. sizes. All sizes are 8 in. deep. For use with or without prefilters at high or low velocities. Write for *Bulletin B-1500-2*.



Far-Air HIGH VELOCITY FILTER

A high velocity, low resistance, viscous coated, impingement air filter for industrial and commercial applications. Available in 1, 2 and 4 in. thickness and all standard and special sizes. One piece frame; rod-reinforced construction. Specify handles. Special materials and finishes for marine, fume and other applications. Write for *Bulletin B-100-2B*.

F/S FILTER

The type F/S Filter is an all metal, cleanable, viscous coated impingement type panel air filter. Performs efficiently at either low or high (350 to 520 fpm) velocity.

The F/S filter is especially effective in air containing a high percentage of lint contamination. The filter is available in ⅛ and 1-⅜ in. thicknesses, standard and special sizes. Write for *Bulletin B-1502-1*.

SELF-WASHING FILTERS

These completely automatic units wash and re-oil themselves. Washing water and contaminated oil is immediately flushed away, eliminating messy sludge and oil sumps. There is no oil entrainment. Each unit has a safety deluge valve that automatically prevents fire from passing the unit. Installation is simple and inexpensive. Write for *Bulletin B-1000-3*.

MICROLOC FILTER

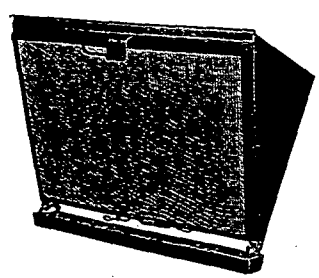
The FAR-AIR Mieroloc is a dry-type, renewable media filter which offers a selection of high efficiencies on the D.O.P. test method. Each Microloc unit consists of 5 inserts to hold the disposable media pads.

The Mieroloc filter units are interlocking and include a panel filter holding frame. A single bank of Microloc filters provides for 2 stage filtration.

Recommended for use after impingement filters where high efficiency is required. Write for *Bulletin B-1303-1*.

MAGNAMEDIA FILTER

Ultra-High Efficiency filters which meet *AEC* specifications for high efficiency and construction. Used to trap radioactive or other fine particle contamination. For laboratory and special industrial applications. Write for *Bulletin B-1304-1*.



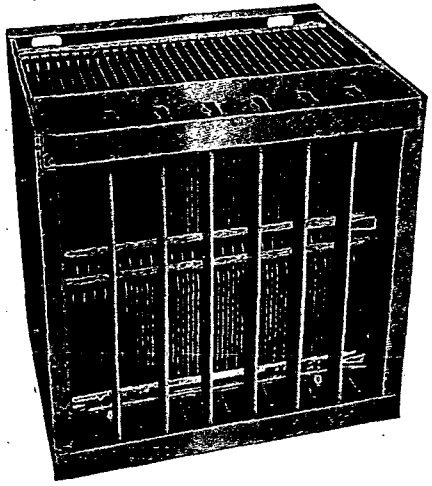
GREASE ELIMINATORS

FAR-AIR Grease Eliminators halt the deposit of grease and lint in the air ducts. Available in one-sided and two-sided models in sizes to meet any size requirement. For restaurants and commercial kitchens. Write for *Bulletin B-201-5*.

MINNEAPOLIS-HONEYWELL REGULATOR CO.

Minneapolis, Minnesota

NEW SELECTOR SERIES OFFERS AN ELECTRONIC AIR CLEANER FOR EVERY REQUIREMENT



Hi Velocity Cell

Provides air cleaning efficiency of 98 percent at an air velocity of 600 fpm. A greater efficiency will result at lower speeds. Honeywell Hi Velocity cleaner should be specified when maximum cleaning is desired, or when unusually high velocities are common. ¼-in. plate interval.

AC/ME Cell

Offers advanced capacity for dirt loading and maximum cleaning efficiency at moderate air flow rates. 90 percent efficient at 600 fpm. This cell should be used in most normal air-conditioning applications. Reduced velocity will result in greater efficiency if desired. ⅝-in. plate interval.

Hi-Load Cell

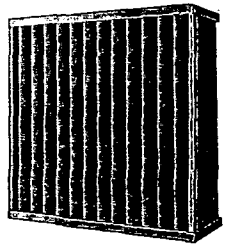
Designed to provide great capacity for dirt loading. Outstanding for air-conditioning applications requiring wider plate interval to reduce frequency of plate washings. Honeywell Hi-Load cell is given rating of 90 percent cleaning efficiency at an air velocity of 400 fpm. ⅜-in. plate interval.

HONEYWELL'S Selector Series of Electronic air cleaners offers you a choice of three air-cleaning cells. The cells are identical except for the number of dirt-collecting plates per cell and the spacing between plates. Particles as small as .001 microns can be stopped by a Honeywell electronic air cleaner. After being charged positively in a powerful electrostatic field, particles are attracted by grounded plates where they are collected. Periodically, the unit is flushed and cleansed by a built-in water-washing system.

Honeywell electronic air cleaners can remove cigarette smoke, bacteria, spores, dusts and corrosive mists.

They protect large investments in machines, merchandise, fixtures and lessen cleaning and redecorating costs.

ALL HONEYWELL ELECTRONIC AIR CLEANERS ARE WARRANTED FOR ONE YEAR AGAINST FAULTY WORKMANSHIP OR DEFECTIVE MATERIAL



Honeywell Activated Charcoal Filters

Where it is necessary to remove odors as well as to clean dirt from the air, we recommend Honeywell Activated Charcoal Filters should be used with the Electronic Air Cleaners above.

In addition to removing odors, these Activated Charcoal Filters cut air conditioning costs by reducing the amount of outdoor air intake normally required to diffuse odors. Four sizes available. Capacity of the filter shown is 1000 cfm, resistance 0.2 wg.

FOR MORE INFORMATION, CALL YOUR LOCAL HONEYWELL SALES OFFICE

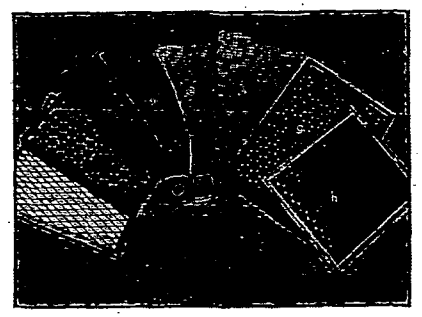
Owens-Corning Fiberglas Corporation

Toledo 1 — **Ohio**

OWENS-CORNING FIBERGLAS

DUST-STOP* AIR FILTERS

AN AIR FILTER PRODUCT TO FIT MOST OF YOUR COMMERCIAL NEEDS



a. Filter Adapter Frame for filter banks. b. Mat-Faced Filter for paint spray booth applications. c. Pad Media Filter for room air conditioners. d. Bonded Mat for strainer type air filtering. e. PF-115 for special high efficiency filtering applications. f. Filter wool, type TWL, for dry type specialty applications. g. Commercial and Industrial Filter for velocities of up to 500 fpm nominal. h. Dust-Stop Filter for velocities of up to 300 fpm nominal. i. Die-cut Aerocor for liquid fuel applications. j. Special molded shaped Aerocor for Jet Fuel-Water Stripper Elements.

THE FILTER

A replacement type filter that since 1931 has offered dependable air filtration, easy installation and convenient maintenance at low cost, for filter banks and other commercial or industrial filtering applications. Standard Dust-Stop Filters are designed for use in filter banks and equipment where velocities do not exceed 300 fpm. The Dust-Stop 450 series may be used with velocities up to 500 fpm. Care should be taken to see that fan capacity is adequate to overcome increased resistances. Rated capacities are 800 cfm at 300 fpm for standard 20 × 20 Dust-Stop Filters and 1000 cfm at 300 fpm for 20 × 25 filters.

DUST-STOP FILTER FRAMES

Frames are designed for the convenience of the engineer in constructing filter banks. Metal frame is universal style, suited for use in flat or "V" banks. Designed-in, self-locking device permits fast, economical assembly. Made to take 2 in. or 4 in. depth of standard 20 in. × 20 in. or 20 in. × 25 in. filters. Black painted or galvanized finish. Front, back and side-angle uprights for Y-bank assembly are also offered in 20 in. increments up to 160 in. length.

DUST-STOP RESISTANCE INDICATOR

Dust-Stop Resistance Indicator acts as a guide to economical bank maintenance. With adjustable filter-change marker set at resistance level specified by engineer as point where filters should be changed, this indicator provides a guide to when filters should be changed and insures operation at rated capacity and efficiency. Available direct from H. O. Trerice Co., 1420 W. Lafayette, Detroit, Mich. For technical data on Dust-Stop Filters and Frames, as well as Fiberglas Flexible Media, write: Owens-Corning Fiberglas Corporation, Dept. HVAC-1, P.O. Box 901, Toledo 1, Ohio.

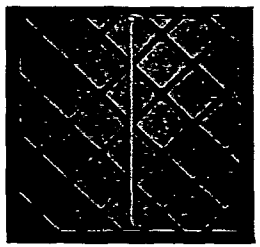
* Fiberglas and Dust-Stop are trade-marks (Reg. U. S. Pat. Off.) of Owens-Corning Fiberglas Corporation for a variety of products made of or with fibers of glass.

Pittsburgh Plate Glass Company

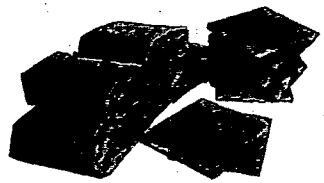
FIBER GLASS DIVISION

One Gateway Center Pittsburgh 22, Pennsylvania

Where clean air is a necessity for either residential, commercial or industrial applications specify GLASFLOSS filters. Products of the Pittsburgh Plate Glass Company, these filters are composed of long and fine glass fibers. Because of this fineness, they furnish a greater dust-collecting area and provide unusually complete filtering action. GLASFLOSS filters are available in three forms, to suit all filtering requirements:

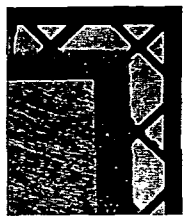


GLASFLOSS STANDARD AIR FILTER—Millions of fine, bonded glass fibers, treated with a special fire-resistant, dust-catching adhesive, trap and hold more dirt. New **Safety-Grille** design makes filters stronger, eliminates danger of cut hands from old-type metal grilles. Excellent for home heating and air conditioning equipment.



GLASFLOSS ROLL-PAK—For bulk users of air filters who are faced with the problem of continuous replacement. The Glasfloss Roll-Pak provides substantial savings . . . as high as 30 percent. It offers quick replacement of filter media, and is available in rolls of 40 in. wide and 10 ft. long and in ½ in., 1 in. and 2 in. thicknesses, packed five rolls per carton, to suit every size commercial air filter. Also available in cut pads.

GLASFLOSS I-S FILTER—To the long, fine glass fibers used in the STANDARD Air Filter a special strainer mat has been added on the air-exit side. This means that the I-S filter is actually two filters in one. For industry, where clean air is essential, or in the meticulous home equipped with forced air heating, Glasfloss I-S provides new high performance air filtering.



	Nominal Sizes	Volume of Air Cleaned	ACTUAL DIMENSIONS Width	Height	Thickness 1" Filter	Thickness 2" Filter
			TOLERANCES Plus 0.00" Minus ⅛"		Plus or Minus 1/16"	
GLASFLOSS STANDARD AIR FILTERS	20" x 25"	1000 CFM	19 13/16"	24 13/16"	13/16"	1 13/16"
	20" x 20"	800 CFM	19 13/16"	19 13/16"	13/16"	1 13/16"
	16" x 25"	800 CFM	15 13/16"	24 13/16"	13/16"	1 13/16"
	16" x 20"	640 CFM	15 13/16"	19 13/16"	13/16"	1 13/16"
Special sizes available	15" x 20"	600 CFM	14 13/16"	19 13/16"	13/16"	1 13/16"
	10" x 20"	400 CFM	9 13/16"	19 13/16"	13/16"	1 13/16"
	10" x 10"	200 CFM	9 13/16"	9 13/16"	13/16"	1 13/16"
	12" x 30⅜" x 1"	735 CFM	11 13/16"	30 3/16"	13/16"	1" only
	20" x 22¼" x 1"	890 CFM	19 13/16"	22 3/16"	13/16"	1" only
	16" x 22¼" x 1"	715 CFM	15 13/16"	22 3/16"	13/16"	1" only
GLASFLOSS I-S AIR FILTERS	20" x 25"	1000 CFM	19 13/16"	24 13/16"	1 13/16"	1 13/16"
	20" x 20"	800 CFM	19 13/16"	19 13/16"	1 13/16"	1 13/16"
	16" x 25"	800 CFM	15 13/16"	24 13/16"	1 13/16"	1 13/16"
	16" x 20"	640 CFM	15 13/16"	19 13/16"	1 13/16"	1 13/16"
	15" x 20"	600 CFM	14 13/16"	19 13/16"	1 13/16"	1 13/16"

RESEARCH PRODUCTS CORP.

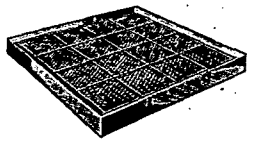
MADISON 1, WISCONSIN

Air Filters for Heating and Ventilating—Grease Filters for Kitchen Exhaust Systems—Paint Arrestors—Automatic Humidifiers

R P AIR FILTERS

Viscuous impingement, with controlled turbulence caused by scientifically staggered baffles is a feature of every R P Air Filter. In addition to high efficiency dust, dirt and pollen removal, R P washable air filters offer a bonus feature—ODOR REMOVAL—through the use of Super Filter Coat Adhesive.

E Z KLEEN AIR FILTERS

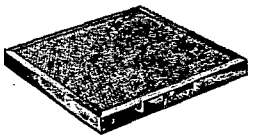


The aluminum, washable air filter that features easy cleaning and moderate cost. Available in ½ in., 1 in. and 2 in. thicknesses—all designed for velocities up to 350 fpm.

Multi-Velocity E Z Kleen Air Filters

For commercial-type installations, where a strong, yet light weight filter is required. Designed for both low and high velocities (350 to 520 fpm) and available in 1 in. and 2 in. thicknesses.

Industrial Air Filters

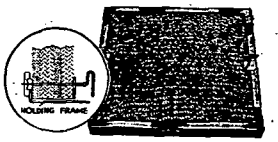


R P Industrial Air Filters are Multi-Velocity filters (250 to 600 fpm) with extremely low initial resistance, and slow build-up resistance. Outstanding where a rugged, heavy duty filter is required for industrial installations. In 1 in., 2 in., 3 in. and 4 in. thicknesses. Holding frames for filter banks, combination lift and lock handles, lift or bail handles are available.

R P ADHESIVE

R P Super Filter Coat adhesive performs all the normal dust and dirt trapping functions, but also is scientifically compounded to adsorb and absorb odors—and prevent their re-circulation in the air stream. A "built-in" detergent simplifies the filter washing process. Available both as a concentrate and in ready-to-use form.

GREASE FILTERS

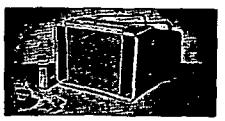


Efficiency in excess of 99 percent on grease vapors, gleaming appearance, ease of cleaning and light weight have made the R P Grease Filter a volume performance leader in its field. Inset illustration shows handle lock—also available with lift or bail handles. A free, comprehensive data booklet on kitchen exhaust systems is available on request.

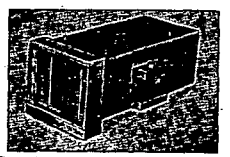
PAINT ARRESTORS

Specially designed, inexpensive, replaceable R P Paint Arrestors trap paint over-spray from spray booth exhaust air. Efficient in operation, R P Paint Arrestors are flexible and versatile—can be used in existing booths and in new installations. Easily installed (with Snap-in Grid to hold Paint Arrestor in frame), and maintained—reduce down-time and floor-space requirements.

APRILAIRE AUTOMATIC HUMIDIFIERS



Residential—3 models available . . . all featuring controlled high capacity (up to 9.2 lb of water per hour). Plenum Model illustrated; Universal Model for buildings with wet heat; Portable Model, requiring no installation.



Industrial-Capacities from 2½ to 10 gph . . . dust-free operation . . . positive control, models for all types of heating.

TRION, INC.

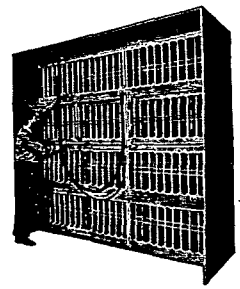
1000 Island Avenue McKees Rocks, Pa.
in Metropolitan Pittsburgh
Toronto, Canada Zurich, Switzerland
Dortmund, Germany Tokyo, Japan
Representatives in Principal Cities
Designers and manufacturers of equipment for cleaning and purifying air and other gases

TRION ELECTRONIC AIR CLEANERS

High efficiency air purification equipment. Electronically removes up to 97 per cent (*National Bureau of Standards Dust-Spot Test* method using normal atmospheric air) of dust, dirt, smoke, pollen, and germs from air. All airborne particles are electronically charged (positive) as they enter the Trion. They are then attracted to and collected upon grounded plates An integral washing system flushes the dirt to a sewer.

FIELD ASSEMBLED UNITS

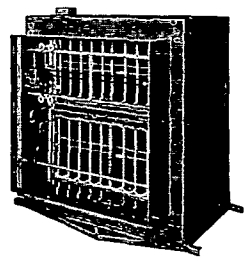
The specific use of field assembled units is for installations requiring the cleaning of air in comparatively large volumes. An example would be multiple floors of a building. Units are sized to specifications for handling any volume of air. The unit consists of all basic components including power pack(s) for supplying the necessary current; required number of dirt collecting cells; and a built-in water spray system for removing collected dirt from cells.



Field Assembled Units

CUSTOM-BUILT *PACKAGED* UNITS

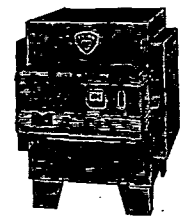
As an aid to the architect, engineer and installer, Trion will supply equipment constructed as a complete *package. Packaged* equipment is available for installation where space is limited. Units are designed for floor mounting or ceiling suspension and for horizontal or vertical air flow. Complete unit includes all basic units housed in a corrosion-resistant metal cabinet including drain pan. Sizes range from 1,300 to 10,000 cu ft of air per minute.



Custom Built Package Unit

PACKAGED RESIDENTIAL UNITS

The standard *packaged* units are designed primarily for residential and small commercial installations. Because of ease of installation, they are also being widely used for industrial and commercial applications where the volume of air to be cleaned is relatively small. Units are available in suspended models or for floor mounting in sizes to handle from 800 (average 5–6-room home) up to 9,600 cu ft of air per minute. Units are sized to the capacity of the furnace or air-conditioning blower.



Package Residential Unit

OIL MIST PRECIPITATOR UNITS

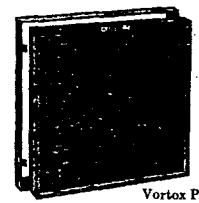
These units are specifically designed for the elimination of coolant oil mist and smoke emanating from high-speed cutting, grinding, and machine operations. Flexibility is provided in that they can be mounted on the floor, on the machine, or suspended from roof trusses. The basis components are the same as for field-assembled units. Under normal machining operations the daily oil recovery from a Trion Oil Mist Precipitator may be as much as 3 to 5 gal.

VORTOX COMPANY

Claremont, California

Panel Air Filters for all Industrial Applications

Vortox Panel Air Filters are of the cleanable, impingement, viscous-coated type with exceptionally low air flow resistance. They are made for use in air conditioning ventilation, range canopies, engine intakes, and general air filtration.



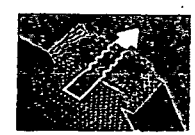
Vortox Panel Filter with Frame

Vortox Type VR Panel Air Filter illustrated above is the type used on engine intakes and in car bodies of diesel locomotives. It is constructed to withstand the *severe usage of locomotive service.* It has exceptional characteristics as in all types of Vortox Panel Air Filters.

FILTER ELEMENT. Fabricated of elastic units of fine steel wire positively interlocked to provide a permanent filter, the element offers many advantages: (1) In a 2 in. x 20 in. panel there are from 9,300 to over 15,000 feet of fine steel wire, depending upon the type. (2) Even distribution of filaments exposes innumerable viscous-coated surfaces to the air stream, thus obtaining the most effective cleaning for space occupied. (3) Proper spacing of filaments prevents clogging and assures maximum dust removal efficiency. (4) Structural strength and permanent resilience combine to withstand "packing" effect of vibration and pulsation.

EFFICIENCY. *Vortox* Panel Air Filters are efficient at both high and low air flows. Consequently to reduce costs of the original installation or to save space, fewer Vortox Filters may be used at higher air velocities. To assure a longer service life, more Vortox Filters may be used at lower velocities. In either case, Vortox Panel Air Filters provide better cleaning at lower cost.

OPERATION. Numerous changes in direction of dust laden air cause the dust particles to impinge on the adhesive viscous-coated surfaces of the filter element. Coarser particles collect near the entrance, while finer



particles penetrate to a greater depth. As the entrance becomes saturated, the cleaning action takes place deeper in the filter and the restriction is increased slightly. Increased restriction in certain sections of the filter diverts the air to cleaner sections, thus distributing the dust load for a higher over-all efficiency and a greater total dust-holding capacity. EASY TO CLEAN. All dust and lint are easily and completely flushed from the filter by water or steam sprays.

Nominal Size of Panel Inches*	Actual Outside Dimensions Inches	Capacity Range In CFM**	Outside Dimensions of Holding Frames Inches
16 x 20 x 2	15⅝ x 19½ x 1⅜	640 to 950	16¼ x 20⅛ x 2
16 x 25 x 2	15⅝ x 24⅝ x 1⅜	800 to 1200	16¼ x 25⅛ x 2
20 x 20 x 2	19¼ x 19½ x 1⅜	800 to 1200	20⅛ x 20⅛ x 2
20 x 25 x 2	19¼ x 24⅝ x 1⅜	1000 to 1550	20⅛ x 25⅛ x 2

* These filters are also made in 4-in. thickness.

** Some filter manufacturers specify very high velocities which are applied to certain internal dimensions of the filter. The range of capacities stated above are computed on the basis of average to high velocities applied to the total filter area using the actual outside dimensions of Vortox Panel Air Filters.

Westinghouse Electric Corporation

Sturtevant Division

**Heating, Ventilating, Cooling, Dehumidifying,
Electronic Air Cleaning and Mechanical Draft Equipment**

Main Office—Boston 36, Mass. Offices in Principal Cities

PRECIPITRON® ELECTRONIC AIR CLEANERS

By catching and holding dust, dirt, smoke, pollens and bacteria as small as one-tenth of a micron, the Westinghouse **PRECIPITRON** provides clean air for homes, hospitals, public buildings, commercial and mercantile establishments and industrial plants. A product of research and development by Westinghouse engineers, **PRECIPITRON** is available in unitary and field assembled models, all easily installed and with capacities to meet exacting requirements.

HIGH CAPACITY UNITS—available from 2500 to 120,000 cfm for use in central ventilation and air conditioning systems. Units feature automatic washing—built-in high voltage connections—simplified erection.

IN-LINE AIR CLEANER SECTIONS—match standard Air Distributing Units for Air Conditioning, Heating, Ventilation and Air Cleanliness control. Sizes from 2000 to 28,800 cfm—built-in traverse washing, in economy or automatic models.

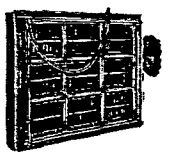
HEAVY DUTY UNITS—especially for industrial applications where dust concentrations are greater. Available with one side or dual side automatic washing, remotely controlled in sizes and capacities from 6000 cfm up.

CABINET UNITS—available in high capacity and heavy duty models. Self-contained—built-in traverse washing—suspended or platform mounting. Capacities 2000 to 28,000 cfm in high capacity, 2000 to 12,000 in heavy duty models.

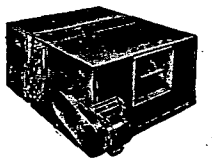
RESIDENTIAL UNITS—for home heating and air-conditioning systems. Save on household cleaning and redecoration—cost little to operate.

OIL MIST CONTROL UNITS—collect coolant oil mists generated by high speed machinery. Fan is built-in for individual machine ventilation.

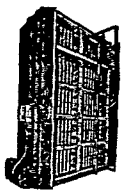
(J-80677)

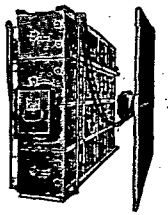


High Capacity Units
Catalog 1460

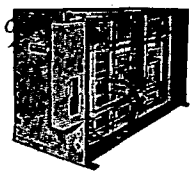


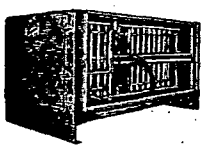
In-Line Air Cleaner
Sections
Catalog 1636



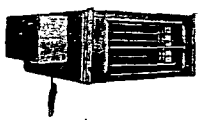


Heavy Duty Units
Catalog 1413





Cabinet Units
Catalogs 1465 and 1425



Residential
Bulletin B-5203



Oil Mist
Catalog 1450

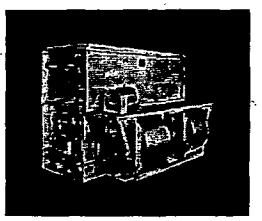
BALTIMORE AIRCOIL COMPANY, INC.

P.O. Box 7322, Baltimore 27, Maryland

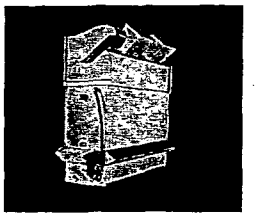
EVAPORATIVE CONDENSERS — COOLING TOWERS

B.A.C. EVAPORATIVE CONDENSERS

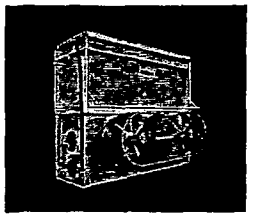
10 to 350 Tons for Freon 12, 22 or Ammonia



Blow-Through
Centrifugal Fans
Maximum Accessibility



Draw-Through
Centrifugal Fans
Minimum Floor Space



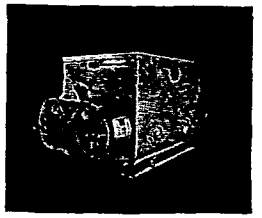
Blow-Through
Propeller Fans
For Horsepower Saving

One of the most complete lines of Evaporative Condensers offered. From 3 different models, you can select the unit that exactly suits your installation requirements.

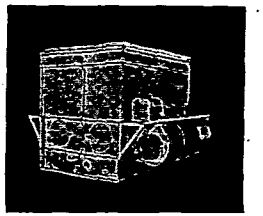
Baltimore Aircoil Company has specialized in the manufacture of Evaporative Condensers for 20 years and is a recognized leader in this field.

B.A.C. COOLING TOWERS

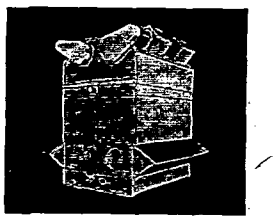
Up to 400 Tons in a single, factory-assembled unit



Flexi-Tower
Propeller Fans
Adjustable Capacity



Blow-Through
Centrifugal Fans
Quiet Operating



Draw-Through
Centrifugal Fans
Ideal for Indoors

A new line of adjustable capacity Flexi-Towers is now available. Without any mechanical changes to the tower, Flexi-Tower capacities can be varied over a wide range to meet changing load requirements.

All B.A.C. units require as little as ½ the space needed for field-erected cooling towers. This, plus their quiet operation, makes them ideal for today's modern buildings where the cooling tower must be inside or otherwise hidden from view. These units are of rugged, hot-dipped galvanized construction, and even the largest sizes require only a simplified and inexpensive base.

Binks Manufacturing Company

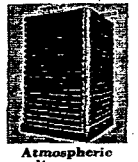
3130-36 Carroll Avenue, Chicago 12, Illinois

Representative in all principal cities

BINKS NATURAL AND MECHANICAL DRAFT COOLING TOWERS

Binks atmospheric spray cooling towers

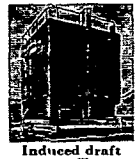
Small sizes, in a variety of standard units with capacities ranging from 10 to 125 gpm—larger units handle from 600 to 1,200 gpm. Special designs furnished in sizes of exceptionally large capacity. Standard tower capacity and temperature performance are based on nozzle pressure of 7 lbs per sq in. Ask for *Bulletin 512*.



Atmospheric cooling towers, Type "T"

Binks steel cased induced draft cooling towers

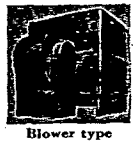
Towers of this type are of the spray filled or deck filled type—made in 20 standard *sizes of sq ft rated* area. Larger models are engineered to specification. Air propulsion assemblies for either type can be arranged for V-belt or reduction gear drive, as required. Ask for *Bulletin 533*.



Induced draft towers, Types "2-K-S", "2-K-W"

Binks blower type mechanical draft cooling towers

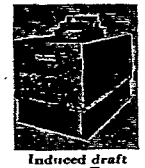
The squirrel cage blower provides extremely quiet operation. Available in single or multiple types, capacity range from 9 to 800 gpm of cooling water. Ask for *Bulletin 477-A*.



Blower type towers, Type "3-B"

Binks multi-finned, plastic decked induced draft cooling towers

New POLYETHYLENE plastic decking provides efficient evaporative cooling surface in small area. No nozzles needed for initial water break-up. Decking slides out for easy cleaning maintenance. Capacities from 15 to 180 gpm. All components pre-assembled for easy installation—hot-dipped galvanized. Fan is stainless steel. V-belt drive. Ask for *Bulletin WT-501*.



Induced draft towers, Type "Watertemp"

BINKS SPRAY NOZZLES FOR EVERY PURPOSE

Binks non-clogging Rotojet spray nozzles

Non-clogging Rotojet nozzles are the heart of every Binks water cooling system. They account largely for the efficiency and satisfactory operation of Binks water cooling installations. In addition to cooling tower applications, Binks Rotojet nozzles have found a wide number of uses in brine-spray and quick-freeze refrigerating systems, air-washing equipment, metal cleaning and treating machines, chemical plants, etc. Rotojets produce a uniformly fine fluid breakup in a hollow cone pattern.

Binks small and medium capacity Rotojet nozzles

To fit ⅛ to ¾ in. pipe connections. Regularly *supplied in brass*, with male or female threads, as specified. Discharge orifices are available over a considerable range for each size. Rotojet nozzles of this type are designed on the side inlet whirl chamber principle, which produces a fine fluid breakup and a uniform spray pattern.



Binks heavy-duty Rotojet nozzles

To fit 1 to 2½ in. pipe connections. Female threads only. Discharge orifices *available in various sizes*, from $^{9}/_{16}$ to $1^{13}/_{16}$ in. The totally unobstructed involute type of whirl chamber produces a uniformly fine water breakup at low pressures (5 to 7 lbs).



Binks pneumatic atomizing nozzles

Series 50 nozzles are designed for use wherever conditions of controlled humidity must be maintained, as in the storage of perishable products, paper storage and printing plants, textile mills, greenhouses, etc. Nozzles of all brass construction deliver round or flat spray and are designed for use with automatic siphon or pressure feed installations.



Standard small and medium Rotojet nozzles are machined from brass bar stock, but can be made on special order from monel, stainless steel, or other machinable metals. Large, heavy-duty Rotojet nozzles for use in large cooling towers and spray pond installations, are cast from high quality brass with precision machined threads and orifices. These nozzles may be cast in other metals for special purposes.

Binks Spra-Rite nozzles

Produce a solid mass cone spray pattern. Small sizes for ¼ to ¾ in. connections are widely used for air washing, cooling, brine refrigeration, rapid evaporation processes, filtering systems, chemicals, etc.



Binks large capacity Spra-Rite nozzles

To fit 1 to 3 in. connections meet a variety of heavy-duty uses in blast furnace gas washers, vibrating and revolving screen coal and gravel washers and water cooling.



Free technical bulletin and engineering service

Binks 28 page *Bulletin No. 5600 contains* a wealth of useful information for anyone specifying spray nozzles for air conditioning, refrigeration, humidifying, heating and ventilating, or similar heat transfer processes. Nozzles shown above and all other nozzles in Binks line are illustrated and described. Data given include nozzle dimensions, capacities and spray angles. Cutaway drawings explain *nozzle operation* in many cases. "Blue print" drawings show installation details for heating and cooling uses where this information is required. Write for a free copy without obligation.



Binks engineering service in planning and installing cooling systems is available *without obligation or cost to architects*, heating and ventilating engineers and builders.

FLUOR PRODUCTS COMPANY

Fluor Products Company — Air Conditioning and Refrigeration Cooling Towers

AC & R Division: Meadows Bldg. Dallas 6, Texas

General Offices: 12000 E. Washington Blvd. Whittier, California

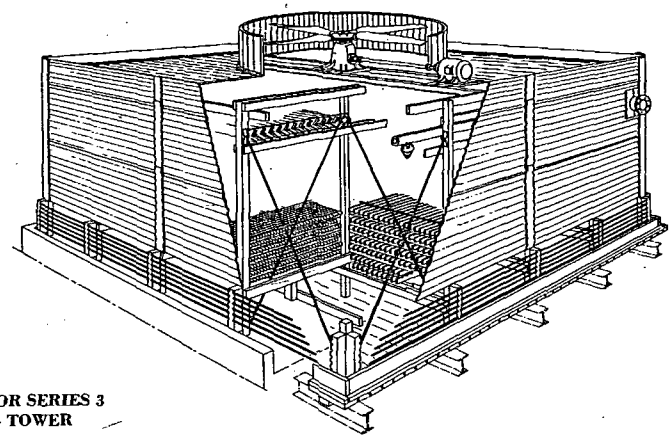
A Division of the Fluor Corporation, Ltd.

District Sales Offices, Agents & Representatives in:
Albuquerque, El Paso, Detroit, Philadelphia, Shreveport, Phoenix, Amarillo, Louisville, Mission, Kan., Memphis, Atlanta, Jacksonville, St. Louis, Cincinnati, Austin, Dallas, Houston, Charlotte, Minneapolis, Washington, D. C., Pittsburgh, Miami, Orlando, Daytona, Tampa, Birmingham, Richmond, New Orleans, Whittier, Corpus Christi, Denver, Salt Lake City, Oak Park, New York, San Francisco, Cleveland

SERIES 3 COOLING TOWER

Offers complete coverage for the Air Conditioning, Refrigeration System field with appeal for *architectural design*. The sheathing material is $^{5}/_{8}$ in. corrugated asbestos-cement, having the corrugations in the horizontal plane. Compact counterflow design requires minimum plan area, and fits rigid space requirements.

Supporting structure may be oriented in any direction. Perimeter anchorage and uniform column spacing simplify design of supports. Water inlets may be located on any wall. The tower is offered with either V-Belt or gear drive. Single and multiple cells cover any required capacity of 40 tons and up.



THE FLUOR SERIES 3 COOLING TOWER

Gives you these extra features . . .

Rugged Construction The entire tower structure with its *tension type bracing* and thru bolting of all members is designed to withstand 30 lb wind load (100 mph). Wood stress values are in accordance with the *U. S. Forest Products Handbook*.

Poly-Grid Fill Fluor patented polystyrene Poly-Grid *fill* gives more cooling per cubic ft than any other fill available, reduces total operating weight by ¼ to ⅓ over conventional wood *fill* towers. *Wood Fill (illustrated in right of cutaway)* also available.

Redwood Distributing Systems Header and laterals of redwood stave construction are completely non-corrosive. Low pressure splash plate units of non-ferrous material are non-clogging and self-draining.

Standard Accessories Galvanized steel hardware, wood ladder, and redwood basin with sump, sump cleanout, overflow-drain and makeup float valve.

Optional Accessories Handrails, Fan guard, non-ferrous hardware.

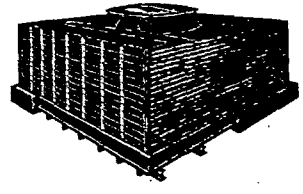
Fluor Air Conditioning and Refrigeration Cooling Towers are available thru exclusive Representatives in all major market areas. Experienced service and erection personnel are *strategically located* to render service to the owner when and where needed.

Write for additional information.

SERIES 2 COOLING TOWER

This tower is offered for installations requiring minimum height, with less critical limitation on plan area. The capacity range is similar to that of the Series 3 Cooling Tower.

Write for additional information.



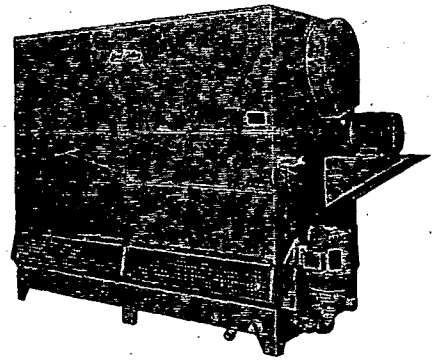
E. D. Goodfellow Co., Inc.

Memphis, Tennessee

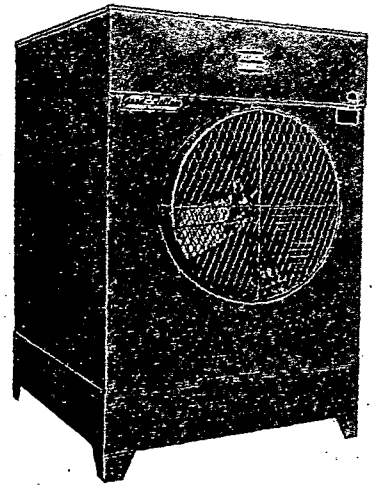
Good-Fellow COOLING TOWERS

GOOD-FELLOW CF Series: These models are produced in capacities from 2 to 100 TR. They are available with our standard corrosion resistant finishes or hot-dipped galvanized (after fabrication). Standard units have double air inlet and vertical discharge. Front or rear horizontal discharge and/or front or rear air inlet are available on order. This series is available with fan motor only, or with separate arrangement for close-coupled pump mounting. These models are leakproof. Models 15 tons and larger are constructed in three sections to facilitate erection and installation. They are normally shipped completely assembled.

We believe the CF Series is one of the most versatile types of cooling tower available.



CF Series



P Series

GOOD-FELLOW P Series: These models are manufactured in tonnage from 3 to 180 TR. All sizes of this series are designed to withstand a wind pressure of 30 lb per square foot. Large diameter slow speed fans produce efficient and quiet operation. Decibel ratings are furnished upon request.

Clear heart Redwood is used for deck filling. Inlet and discharge louvers are also of Redwood. All models P8 and larger can be completely taken apart, the largest section being the cold water basin. Motors on all models through P30 are mounted inside of tower casing. All models of this series are **HOT-DIPPED GALVANIZED** after fabrication. They can also be supplied with corrosion resistant Epon epoxy exterior and Epon Coal-tar pitch interior finishes.

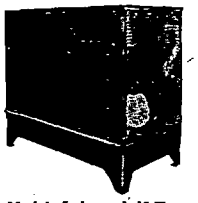
Havens COOLING TOWERS

Division of
Havens Structural Steel Company

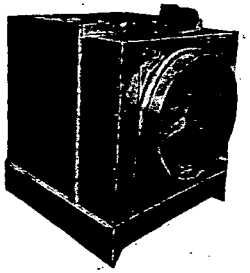
1713 Crystal Kansas City, Missouri

Standard and Custom-engineered Cooling Towers from 5 to 600 Tons

Representatives in Principal Cities of the U. S.



Models 5 through 15 Tons



Models 20 through 75 Tons

HAVENS "33 Series" 5 to 75 Tons

FEATURES

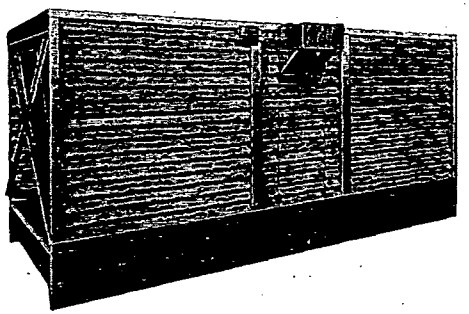
- All steel parts hot dipped galvanized after fabrication.
- Heavy Fan Screen
- Belt Guard
- Weather protected motors
- Stainless Steel Fan Shaft (our own bearing design)
- Stainless Steel Fan Blades
- Corrosion-resistant cast-iron sheaves
- Steel sheets, heavy gauge hot rolled mild steel

The "33 Series" is recommended for residential and small commercial installations. The units are shipped assembled, ready for installation. Water tray distribution system assures ample spread over all-heart, California redwood fill. All steel parts are hot dipped, double galvanized before assembly. Towers can be disassembled and reassembled with ease when necessary. Special fan bearing design assures quiet, trouble-free operation for years; it requires little maintenance.

HAVENS "V SERIES" Models 80 thru 600 Tons

(Models above 200 tons are field erected.)

- Recommended for medium and large commercial and industrial installations.
- Low, modern silhouette for pleasing appearance.
- Transite casings (corrugated cement asbestos board)
- Unique bearing design cuts maintenance costs.
- Models 80 to 200 tons are factory-assembled units—larger tonnages are field erected.



Havens "V Series" Cooling Towers are ruggedly constructed to stand years of service under the most adverse weather conditions.

Wood fill is California clear, all-heart red wood with unique tray arrangement to gain most advantageous water distribution.

The variable pitch fan provides versatility. To minimize installation costs detachable hoisting yoke is furnished with each tower.

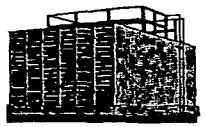
Product bulletins and specification data available on request.

Lilie-Hoffmann Cooling Towers, Inc.

Exclusive Builders of Cooling Towers Since 1898

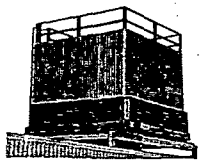
1450 So. Vandeventer Ave., St. Louis 10, Mo.

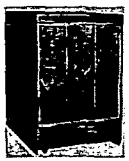
Two Modern Plants—St. Louis, Mo., and Plainview, Texas



LILIE-HOFFMANN CROSS FLOW TOWERS . . . Outstanding for its "Low-Hite" and low pumping head. Ruggedly constructed to serve intermediate sized air conditioning and refrigeration installations. Appearance designed to fit and complement modern architecture. Economical in first cost as well as operational cost, yet same high standard of quality of materials and mechanical equipment are maintained. *Bulletin CM-456*

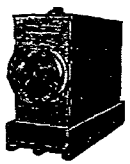
LILIE-HOFFMANN COUNTER-FLOW TOWERS . . . Highly functional with conservative, highly efficient spray distribution. Choice of casing materials will fit any architecture—may be installed inside of masonry enclosure with louvered openings. Low cost, space saving tower with very high performance record. This tower is particularly useful for high efficiency designs. Cast aluminum variable pitch multi-bladed fans are supplied. *Bulletin LM-456*

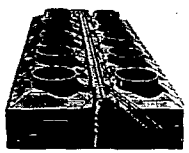




LILIE-HOFFMANN NEW WATER-MASTER COOLING TOWER . . . Steel packaged units distributed by dealers. Built to Lilie-Hoffmann high standard quality with heavy duty motor. Standard equipment includes air inlet and outlet screens. Have many years of built-in maintenance free operation with a high performance record. Size from 3 to 60 ton —shipped assembled. Large units are easily disassembled for special handling. *Bulletin WM-757*

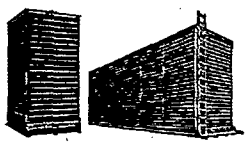
LILIE-HOFFMANN HORIZONTAL DRAFT PACKAGE TYPE TOWER . . . Designed for 4 to 100 ton air conditioning and refrigeration installations. Side walls are of corrugated and/or flat cement asbestos with stainless steel trim. Long service life with no painting problem means conservative operation. All towers equipped with either 4 or 6 bladed, deep pitched, low speed fans for delivery of maximum air at minimum horsepower and low decibel rating. *Bulletin CU-857-5M*





LILIE-HOFFMANN INDUSTRIAL INDUCED DRAFT TYPE . . . A very heavy duty tower providing the maximum in high efficiency for large capacities. Designed with conservative ratings. Provides independent operation of individual cell or group of cells. Cells can be arranged in-line or back to back. Pumping head design is low. Installations are world wide. *Bulletin MD-94-Z*

LILIE-HOFFMANN ATMOSPHERIC SPRAY TOWERS . . . Manufactured in two models. Smaller CS series for air conditioning and refrigeration up to 50 tons which are distributed by dealers. Larger S series for heavy duty and large capacity industrial installations. Heart quality redwood throughout. Galvanized wrought iron headers with bronze nonclogging spray nozzles supplies on both models. Completely prefabricated with simple instructions for easy erection. *Bulletins CS and S.*

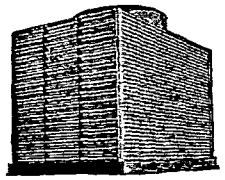


THE MARLEY COMPANY

222 W. Gregory, Kansas City 13, Missouri

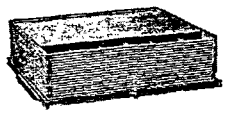
Representatives in All Principal Cities (Consult Classified Phone Directory)

Water Cooling Towers of All Types and Capacities. Spray Nozzles



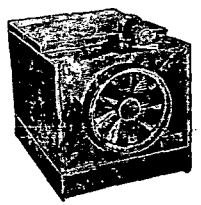
MARLEY DOUBLE-FLOW AQUATOWER®

MARLEY DOUBLE-FLOW AQUATOWER . . . For performance —and professional acceptance . . . there is nothing like the Double-Flow Aquatower in the intermediate-capacity cooling field. This "low silhouette," low pumping head cooling tower is designed for air conditioning and refrigeration jobs of 50 tons or more. It has open gravity water distribution, full height louvered walls and all fittings and mechanical equipment are readily accessible. Available in wood or steel structure with asbestos cement board casing and in single or multi-cell arrangements. Write for *Bulletin DFA-58.*



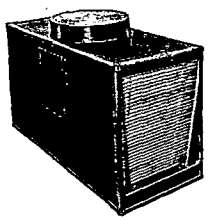
MARLEY UNDERFLOW® AQUATOWER

MARLEY UNDERFLOW AQUATOWER . . . First intermediate capacity cooling tower with no louvered sides or ends. Mechanical equipment is located beneath tower which inherently baffles operating sounds and facilitates inspection and service. Structural framework of steel cold water basin acts as grillage; only corner support columns required. Hot water basins at top are covered, leaving minimum air discharge area at ends exposed from above, greatly increasing fire-safety. Casing and basin covers asbestos cement board. Write for *Bulletin UA-58.*



MARLEY AQUATOWER®

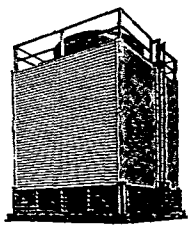
MARLEY AQUATOWER . . . Available in a complete range of 11 sizes, rated for servicing from 3 to 60 tons of air conditioning or refrigeration. Thousands of steel packaged units have proved themselves in years of maintenance-free operation. They are stocked by dealers throughout the country—are shipped completely assembled and require no field construction. Large units may be readily disassembled for easy handling. Write for *Bulletin AQ-58.*



MARLEY PERMATOWERS®

MARLEY PERMATOWERS . . . Available in 11 sizes, are the newest packaged cooling towers designed for maximum resistance to corrosion. They combine the time-proved design principle of Marley double-flow cooling with many new design features, including all-plastic water distribution piping assembly, plastic depressed sump, plastic fan cylinder, resin-impregnated-fiber surfaced laminated casing and basins. Write for *Bulletin PT-58.*

MARLEY COUNTER-FLOW SERIES, STEEL AND WOOD . . . This new series of Marley counterflow cooling towers is designed especially for heavy duty air conditioning and refrigeration service. The all-steel model (with corrugated asbestos cement board casing) is designed to meet fire codes in all metropolitan areas. Both the steel and wood models blend well with modern architecture. May be installed with any combination of louvered walls open providing unusual flexibility and maximum utilization of usable space. Write for *Bulletins CS-58 and CW-58.*



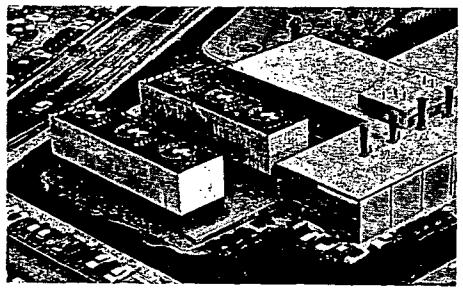
COUNTER-FLOW STEEL

MARLEY CROSS-FLOW . . . For large capacity industrial cooling, Marley offers 2 towers that utilize the patented Marley Cross-Flow principle—The Double-Flow and Single-Flow. Write for *Bulletin CF-58.*

MARLEY SPRAYTOWERS . . . Complete line of all-redwood natural draft cooling towers; capacities from 2 tons up. Write for *Bulletins ST-58 (up to 35 tons) 200-58 (above 35 tons).*

Phillips Cooling Tower Co., Inc.
220 DUPONT ST., BROOKLYN 22, N. Y.

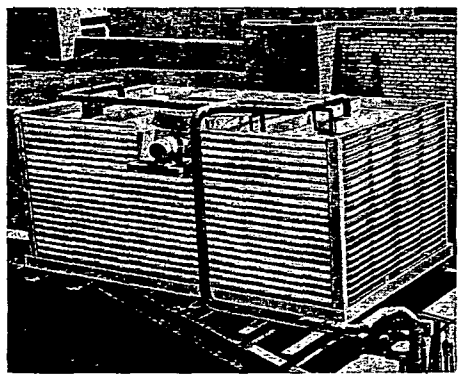
HEADQUARTERS FOR COOLING TOWERS



(150 TONS and UP)

Tower on left is redwood framed Type WT with cement asbestos casing and louvers. Type WT towers are available from 500 tons and up.

Counterflow Type Towers available with either redwood or steel framing. Steel framing members are all hot dipped galvanized after fabrication in accordance with ASTM specifications. Partitions, louvers and siding of cement asbestos board are standard on this type of tower.



Series "DA" towers are available in California redwood, or Hot Dipped Galvanized framing. Corrugated cement asbestos board casing, laid horizontally, is fastened to frame with Galvanized bolts and Neoprene washers. Slipfit louvers of flat cement asbestos board provide easy access. All hardware is Hot Dipped Galvanized.

SPECIAL FEATURES ON ALL TOWERS

Corrosion Resistant Hardware. Aluminum or Stainless Steel Fans. Nailless Redwood Filling Removable at Tower Base. Nailless Three-Pass Redwood Eliminators. Removable Louvers.

Monarch Manufacturing Works, Inc.

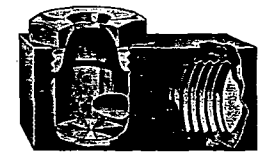
2509 E. Ontario St., Philadelphia 34, Pa.

SPRAY NOZZLES FOR WATER AND OIL

NON-CLOG AIR WASHER NOZZLES

Produce an exceptionally efficient, evenly distributed hollow cone spray. Single large tangential inlet to swirling chamber minimizes any possibility of clogging. Also available in ⅜ in. to 1 in. pipe sizes inclusive, and of Brass, Stainless and Monel.

Fig. 631



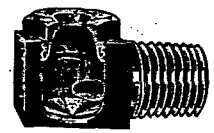


Fig. 629

Water Capacities: Gallons per Hour

Pipe Size and Fig. No.	Orifice	Lead	Operating Pressure: Lb per Square Inch							
			5	10	20	30	40	60	80	100
⅛ in.	69	69			2.7	3.3	3.8	4.7	5.4	6.0
Fig. 629	61	61			4.2	5.1	5.9	7.2	8.3	9.2
	61	53	2.7	3.7	5.3	6.5	7.5	9.2	10.5	11.8
or	53	53	4.0	5.7	8.2	10.0	11.5	14.1	16.3	18.2
	49	49	5.6	7.9	11.1	13.6	15.7	19.3	22.2	24.8
¼ in.	3/32	3/32	9.6	13.5	19.2	23.4	27.0	33.0	38.4	42.9
Fig. 629	⅛	3/32	12.2	17.3	24.4	29.9	34.6	42.4	49.0	54.9
or	⅛	7/64	14.6	20.6	29.1	35.7	41.0	50.4	58.4	65.1
Fig. 631	⅛	⅛	15.9	22.5	31.9	39.2	45.1	55.2	63.7	71.5
	11/64	⅛	21.7	30.6	43.2	53.0	61.1	75.0	86.4	98.0

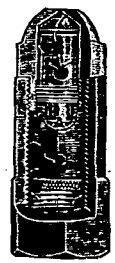


Fig. F-80

AIR CONDITIONING AND OIL BURNER NOZZLES

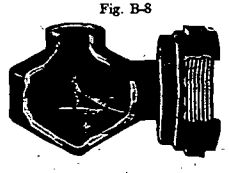
Water Capacities: Gallons per Hour

Nozzle No.	Operating Pressure, Lb per Square Inch					Min. Pressure (PSI)
	25	40	60	80	100	
1.35		.57	.70	.81	.90	40
1.65		.75	.92	1.06	1.18	40
2.00		.94	1.14	1.32	1.49	35
2.50		1.13	1.39	1.60	1.80	30
3.00	1.00	1.27	1.55	1.79	2.00	25
3.50	1.30	1.64	2.01	2.32	2.60	25
4.00	1.50	1.90	2.33	2.69	3.00	20
4.50	1.90	2.40	2.94	3.40	3.80	20
5.00	2.20	2.78	3.41	3.94	4.40	20
5.50	2.40	3.04	3.72	4.30	4.80	15
6.00	2.60	3.28	4.03	4.66	5.30	15
7.00	3.00	3.80	4.65	5.37	6.00	15

Produce very effective breakup with direct pressure only. Capacities above are on water. "Nozzle No." is capacity on 34 second Saybolt viscosity oil at 100 lb pressure. Larger sizes up to 60.00 gph and smaller sizes down to 0.60 gph.

Furnished of all Brass for Water—Stainless Steel tip and disc for Oil. Standard with ⅛ in. or ¼ in. female or ¼ in. male pipe Brass adapter and Monel gauze strainer.

Fig. B-8



SPRAY POND NOZZLES

For recooling condenser water, etc. Operate on pressures from 5 lb upward. Made of Cast Red Brass and in pipe sizes ¾ in., 1 in., 1½ in., 2 in., 2½ in. and 3 in. Capacities from 4.1 to 134 gpm at 7 lb pressure.

Write for Detailed Catalogs

AEROFIN CORPORATION

101 Greenway Avenue, Syracuse 3, N.Y.

Standardized Light-Weight Heat Exchanger Surface

There is a factory-trained Aerofin application and service engineer located near you to give you prompt service. Write, wire or phone the factory.



Available in a wide range of sizes and types for heating, cooling, air conditioning and process, Aerofin is the modern standardized light-weight encased fan-system surface designed by *fan engineers* to meet present and future requirements of this highly specialized field.

All standard Aerofin units are furnished as completely encased units, ready for pipe and duct connections. The patented casings are built of pressed steel and are exceptionally strong and rigid to protect the unit from all strains of pipe connections and expansion or contraction in service. The casings are flanged on both faces, top and bottom, and template-punched for bolting adjacent units together or for easy duct connection.

Guarantee

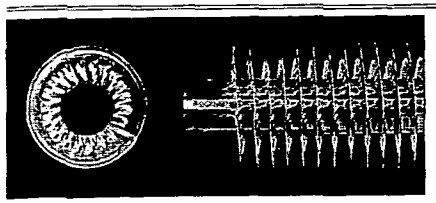
Aerofin is guaranteed, when properly installed and operated at pressures and temperatures not exceeding those specified, against failure or leakage due to faulty material or manufacture. The Aerofin Corporation will furnish a new unit, fob car Syracuse, N. Y., or Kingston, Ontario, for replacement of any unit found defective in either material or workmanship within one year from date of shipment of original unit, when such defective unit is returned to the factory at Syracuse, N. Y., or Kingston, Ontario.

Extended Surface Exclusively

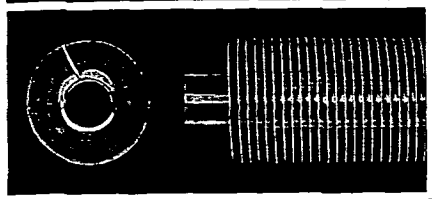
Aerofin manufactures light-weight extended-surface heat exchangers exclusively – not as a side line or accessory. Modern manufacturing, research and engineering facilities give real meaning to the Aerofin guarantee. Aerofin ratings are accurate – units may be safely installed to the full rated capacity.

Aerofin heat exchangers are rated according to accurate research laboratory test data and operate efficiently at full rated capacity.





Old Type Aerofin Surface with Crimped Fins



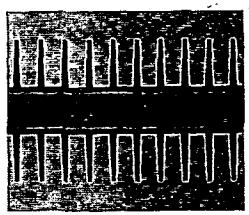
New Patented Aerofin Surface with Smooth Fins

Smooth-Fin Construction Provides Greater Capacity – Less Air Resistance

With air resistance and turbulence caused by the corrugations in the old-style fins now eliminated, note how many more fins can be used per inch of tube length. Note the extremely low air friction of this new design.

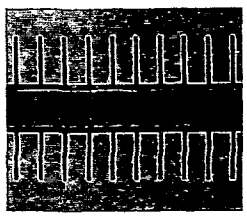
For optimum service, aluminum fins are spaced 14 fins per inch; copper, 12 fins per inch. Because of the low friction losses, high air velocities are entirely practical. This, combined with maximum heat transfer per sq ft of face area, results in extremely high heating or cooling capacity in relation to the available space.

For lower temperature differences, the tubes can be spaced on wider centers as required, or fewer rows may be used.



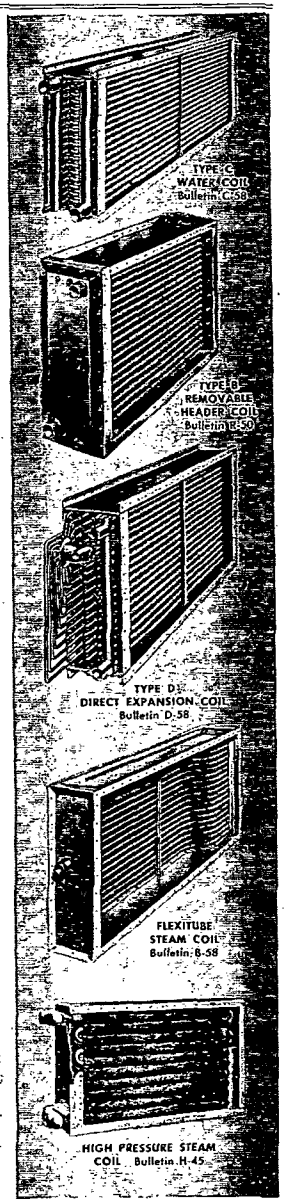
Aerofin Tapered Fin

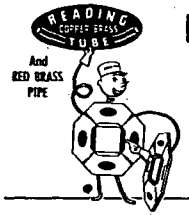
The wide contact base of the new Aerofin tapered fin conducts sufficient heat between fin and tube to make the entire fin effective transfer surface. It is not necessary to crimp the fin to provide the required contact surface.



Conventional Straight Fin

The relatively small amount of heat conducted by the small contact surface means that only a portion of the fin is effective heat transfer surface. Such fins are generally crimped to provide greater contact surface.





READING TUBE CORPORATION

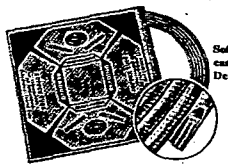
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DALLAS, TEXAS 9000 Sovereign Row Brook Hollow Industrial District
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READING "LEKTROSEAL"® COPPER REFRIGERATION TUBE



Soft tempered easy forming—Dehydrated

Handy, 50 ft coil in convenient protective carton

Crimped ends seal out all moisture and dirt

Reading refrigeration tube is made to the highest standards. Composed of 99.90% pure deoxidized copper, every coil is thoroughly dehydrated and sealed at the mill before being packaged. Its immaculately clean inside surface is the result of the most modern production methods and exhaustive inspection operations.

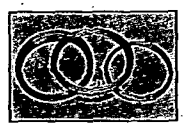
All sizes of refrigeration tubing are regularly shipped in 50 foot lengths. Coils are individually packaged inside master shipping cartons.

ACTUAL OUTSIDE DIAMETER	⅛"	3/16"	¼"	5/16"	⅜"	½"	⅝"	¾".
WALL THICKNESS, INCHES	.030	.030	.030	.032	.032	.032	.035	.035
LBS. PER LINEAR FOOT	.035	.058	.080	.109	.134	.182	.251	.305
WEIGHT PER 50 FT. COIL	1.74	2.88	4.02	5.45	6.70	9.10	12.55	15.25
NO. COILS TO MASTER CARTON	10	10	10	10	10	5	5	3
TOTAL FEET TO MASTER	500	500	500	500	500	250	250	150
TOTAL NET WT., EA. MASTER	17.4	28.8	40.2	54.5	67.0	45.5	62.8	45.8

MACKENZIE WALTON SMALL SEAMLESS QUALITY PRECISION TUBING

in Copper and Copper Alloys

- Famous for Accurate and Fine Finish Tubing
- Quality Controlled from Start to Finish
- Produced to Your Exact Specifications



CAPILLARY



SHAPE

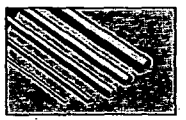
CAPILLARY—Available in coil and straight lengths from 0.031 in. OD to 0.250 in. OD and 0.009 in. to 0.055 in. in wall thickness, depending upon alloy and form specified. We guarantee uniformity of size, cleanliness and smoothness of bore, and consistency of temper. Modern methods and equipment under constant control by skilled metallurgists, assure quality from start to finish.

SHAPE—Whatever shape you specify . . . rectangular, square, oval, fluted, reeded, embossed, specialty or architectural, all Mackenzie Walton shape tubing is produced to meet the most exacting requirements. Available to your specifications within our size range in copper and copper-base alloys in random mill or exact cut lengths. Consult us on all applications.

THIN WALL—Available in Copper (DHP or OFHC), Brass (70/30, 85/15, 80/20), Bronze (95/5, 90/10) and Nickel Silver (18 percent, 15 percent, 10 percent) in size ranges from 1/32 in. to ⅜ in. OD, and wall thickness 0.007 in. and upward, depending on alloy and form specified. Noted for fine quality and cleanliness of OD and ID, our thin wall tubing is always made to close tolerance.

BRAIDED—Where armor or reinforcement of capillary tubing is an important factor, we will braid to your exacting requirements. Choice of B & S wire gage size in eye-appealing copper, tinned copper, stainless steel, galvanized steel, and aluminum with one to four layers over capillary tubing from 0.031 in. to 0.125 in. outside diameter, in wall thickness from 0.009 in. to .040 in.

BOURDON—Our Bourdon tubing assures you absolute, minimum tolerance variation for instrument accuracy. Quality controlled through rigid inspections means consistent spring properties, accurate size and constant wall thickness. We can furnish six standard size groups in random mill or exact cut lengths, in wall thicknesses from 0.0039 in. to 0.068 in., or to your specifications.



THIN WALL



BRAIDED



BOURDON

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COPPER BRASS
TUBE
And
RED BRASS
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READING "READI-FIN" Type W/H IS EASILY FABRICATED

That same rugged construction and design that gives Type W/H finned tube its high performance characteristics, also readily lends itself to being coiled, bent or formed into the particular type of bundle you desire. This feature is important. Since the fins are an integral part of the tube, any bending or coiling does not affect the strength of the concentricity. The performance remains high and the durability lasting. "Readi-Fin" Type W/H is available in the "as finned" or "annealed" temper. Annealed temper is suitable for bending.

Types of end finishing—"Readi-Fin" type W/H

Type A—Full-finned tube Type B—Both ends plain



"READI-FIN" INTEGRAL FIN TUBING SPECIFICATIONS

Specified O.D.-In.	Catalog Number	Material	PLAIN END O.D. in Inches t_1	PLAIN END Wall Thickness in Inches $2t_1$	FINNED SECTION Fin O.D. in Inches tf	FINNED SECTION Root Dia. in Inches $2f$	FINNED SECTION Wall Thickness in Inches $4f$	Outside Area Ft. 7/Ft.	Ratio of Outside to Inside Areas	Approx. Weight in Lbs.
⅜	3423	Copper	.622-.628	.056-.064	.714-.724	.518-.528	.022-.028	.537	4.49	.348
½	3427	Copper	.622-.628	.071-.079	.751-.761	.485-.495	.024-.030	.723	6.18	.401
⅝	7430	Copper	.747-.753	.061-.069	.846-.856	.842-.852	.027-.033	.671	4.36	.464
1	1435	Copper	.872-.878	.071-.079	.970-.980	.784-.794	.032-.038	.775	4.12	.590
			11-Fin Tubing Specifications							
½	3128	Copper	.622-.628	.062-.070	.730-.773	.520-.530	.023-.031	.554	4.57	.375
⅝	7132	Copper	.747-.753	.066-.074	.875-.900	.620-.630	.029-.035	.707	4.79	.490
1	1138	Copper	.872-.878	.072-.078	1.000-1.025	.745-.755	.035-.041	.821	4.65	.620

READING READI-FIN Type S/T EXTENDED SURFACE CONDENSER TUBE

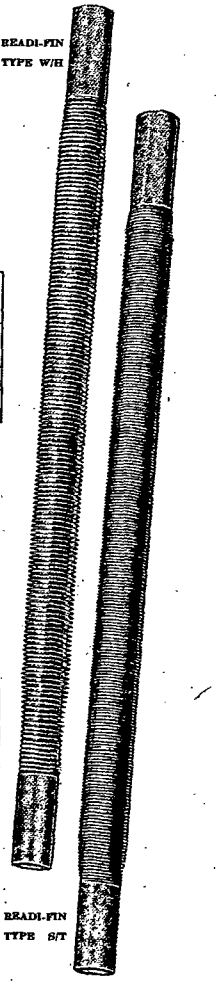
"Readi-Fin" Type S/T—the extended surface condenser tube—is designed primarily for shell and tube heat exchangers. Tube ends may be finned, stripped or plain. The outside diameter of the plain end is approximately the same as the outside diameter of the fin. "Readi-Fin" Type S/T, with plain ends, is fabricated in shell and tube heat exchangers in the same manner as plain condenser tubing. "Readi-Fin" Type S/T is available in "as finned" or "annealed" tempers. The "as finned" temper has plain ends suitable for rolling-in operation. The "annealed" temper is suitable for bending.



Type S/T Condenser Tubing

"READI-FIN" INTEGRAL FIN TUBING SPECIFICATIONS—19-Fin Tubing

Specified O.D.-In.	Catalog Number	Material	PLAIN END O.D. in Inches	PLAIN END Wall Thickness in Inches	FINNED SECTION Fin O.D. in Inches	FINNED SECTION Root Dia. in Inches	FINNED SECTION Wall Thickness in Inches	Outside Area Ft. 7/Ft.	Ratio of Outside to Inside Areas	Approx. Weight in Lbs.
⅛	2932	Copper	.497-.503	.044-.052	.488-.492	.348-.379	.029-.033	.312	3.84	.233
¼	2949	Copper	.497-.503	.059-.066	.488-.492	.348-.379	.045-.053	.312	4.52	.304
¼	29498	85-15 Red Brass	.497-.503	.065-.078	.484-.492	.348-.379	.045-.053	.310	4.30	.304
⅜	3933	Copper	.622-.628	.049-.053	.611-.617	.494-.504	.033-.038	.485	3.68	.316
⅜	3942	Copper	.622-.628	.058-.063	.611-.617	.494-.504	.029-.045	.485	3.72	.325
⅜	3949	Copper	.622-.628	.056-.063	.611-.617	.494-.504	.046-.053	.485	3.84	.357
⅜	3963	Copper	.622-.628	.068-.075	.611-.617	.494-.504	.046-.053	.485	4.19	.462
⅜	39498	85-15 Red Brass	.623-.628	.063-.070	.609-.617	.492-.504	.045-.053	.483	3.98	.350
⅜	39618	85-15 Red Brass	.623-.628	.075-.088	.609-.617	.492-.504	.061-.069	.483	4.50	.468
½	3933	Copper	.747-.753	.051-.059	.736-.742	.619-.629	.032-.038	.596	3.81	.397
½	3942	Copper	.747-.753	.054-.062	.736-.742	.619-.629	.039-.045	.596	3.90	.448
½	3949	Copper	.747-.753	.063-.071	.736-.742	.619-.629	.046-.052	.596	3.99	.489
½	3963	Copper	.747-.753	.073-.080	.736-.742	.619-.629	.063-.068	.596	3.84	.585
½	3983	Copper	.747-.753	.091-.098	.736-.742	.619-.629	.080-.086	.596	4.14	.680
½	39338	85-15 Red Brass	.747-.753	.052-.059	.734-.742	.617-.629	.032-.038	.596	3.81	.395
½	39498	85-15 Red Brass	.747-.753	.063-.073	.734-.742	.617-.629	.045-.053	.596	3.99	.494
½	39538	85-15 Red Brass	.747-.753	.073-.083	.734-.742	.617-.629	.061-.069	.596	3.81	.591
½	39638	85-15 Red Brass	.747-.753	.093-.103	.734-.742	.617-.629	.078-.087	.596	4.14	.685
⅝	7949	Copper	.872-.878	.061-.069	.861-.867	.742-.754	.046-.052	.558	3.64	.487
⅝	7963	Copper	.872-.878	.080-.086	.861-.867	.744-.754	.063-.068	.558	3.63	.585
⅝	7983	Copper	.872-.878	.095-.100	.861-.867	.744-.754	.080-.086	.558	3.84	.680
⅝	79538	85-15 Red Brass	.872-.878	.083-.087	.859-.867	.742-.754	.061-.069	.558	3.62	.578
⅝	79938	85-15 Red Brass	.872-.878	.098-.103	.859-.867	.742-.754	.078-.087	.558	3.84	.868
1	1949	Copper	.997-1.003	.062-.072	.986-.992	.865-.879	.046-.052	.774	3.51	.597
1	1963	Copper	.997-1.003	.080-.091	.986-.992	.865-.879	.063-.068	.678	3.48	.683
1	1983	Copper	.997-1.003	.095-.106	.986-.992	.865-.879	.080-.086	.678	3.66	1.001
1	19438	85-15 Red Brass	.997-1.003	.083-.093	.984-.992	.867-.879	.061-.069	.676	3.47	.868
1	19618	85-15 Red Brass	.997-1.003	.092-.097	.984-.992	.867-.879	.078-.083	.676	3.65	1.005



The G & O Manufacturing Company

140 Winchester Avenue **New Haven 8, Connecticut**

G&O

INDIVIDUAL AND STRIP FIN TUBING

We manufacture a complete line of high efficiency heat transfer surfaces in a wide range of standard sizes with individual square, round and oblong fins.

The use of individual fins permits using fins in groups at intervals along the straight length of any tube for the manufacture of U-bends, continuous return bend, and spirally wound coils. Where space permits, the use of square or oblong fins provides proportionately greater surface than a round fin of a diameter equal to one side of the square or oblong fin.

G&O Individual Fin Tubing is available for heating coils of every description, including unit heaters and blast coils, intercoolers, aftercoolers, baseboard radiation, and convector elements.

STRIP FIN TUBING

Continuous strip aluminum fin tubing is particularly suited for cooling and air conditioning coils. The lightweight all aluminum construction is of advantage in installation where weight is an important factor. Various fin spacing is available and sections can be stacked to provide any practical number of rows in direction of air flow within the limits of maximum height and width area dimensions.

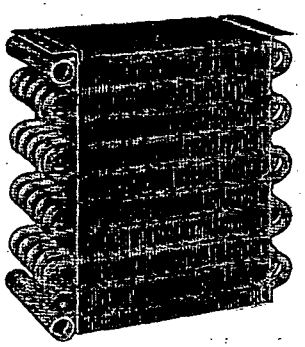
G & O continuous strip fin coils are produced in two standard sizes: ⅝ in. OD tubing with ⅞ in. wide continuous strip fins, or ⅝ in. OD tubing with $1\frac{1}{16}$ in. wide continuous strip fins.

INDIVIDUAL FIN TUBING

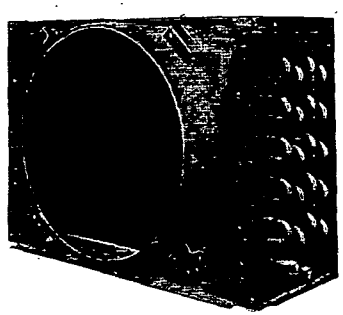
STANDARD SIZES

OD of Tube	Fin Size	Fin Spacing per Inch	Surface per Linear Foot
⅜"	⅞" sq	6	0.80 sq ft
⅝"	1⅜" r'd	6	0.87 sq ft
¾"	1¼" r'd	6	1.55 sq ft
¾"	1⅝" sq	6	2.40 sq ft
¾"	1⅝" sq	7	2.95 sq ft
⅞"	2⅛" sq	6	4.15 sq ft
⅞"	2⅛" sq	5	3.50 sq ft
1"	2⅛" sq	6	4.00 sq ft
1⅛"	2⅛" sq	6	3.82 sq ft

RADIATING ELEMENTS FOR ALL HEAT TRANSFER PURPOSES ARE AVAILABLE IN A WIDE RANGE OF SIZES



Continuous Strip Fin, All Aluminum Cooling Coil



Continuous Strip Fin All Aluminum Heating & Cooling Coil

The Patterson-Kelley Company, Inc.

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Representatives in Principal Cities

air conditioning and refrigeration equipment

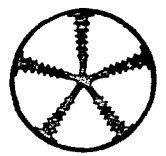


FIG. 1—Exclusive P-K aluminum fin is designed for optimum heat transfer and vaporization.

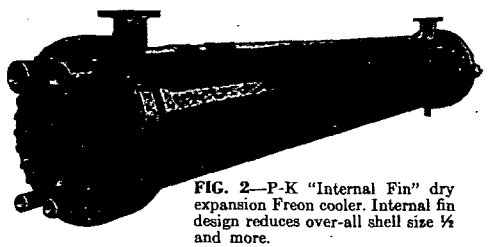


FIG. 2—P-K "Internal Fin" dry expansion Freon cooler. Internal fin design reduces over-all shell size ½ and more.

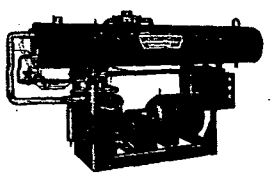


FIG. 3—P-K "Internal Fin" dry expansion Freon cooler installed on Worthington packaged water chiller. Package system designed for air conditioning.

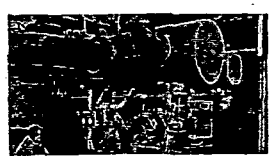


FIG. 4—P-K dry expansion Freon chiller installed on Worthington packaged air chiller.

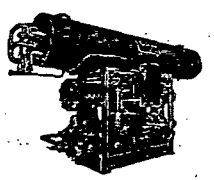


FIG. 5—P-K "Internal Fin" dry expansion Freon cooler installed on Chrysler Airtemp packaged whiskey chiller.

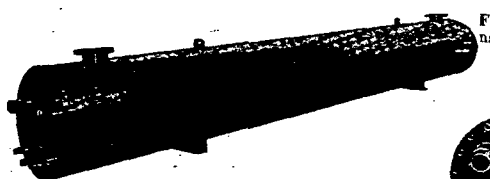


FIG. 6—Insulated 4 circuit "Internal Fin" Freon chiller.

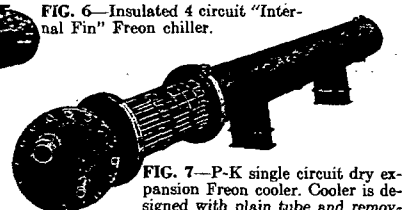


FIG. 7—P-K single circuit dry expansion Freon cooler. Cooler is designed with plain tube and removable "U" tube bundle.

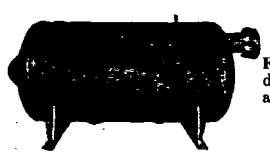


FIG. 8—P-K storage water cooler designed for process liquid cooling and chilled water services.

The Patterson-Kelley Co., Inc. • process heat exchanger division

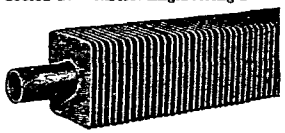
Rittling THE RITTLING CORPORATION

103 Kentucky St. Buffalo 5, N. Y.

REPRESENTATIVES IN PRINCIPAL CITIES

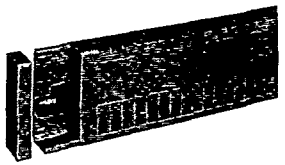
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RITTLING Heating and Ventilating equipment includes a complete line of finned tube units for residential and non-residential application; Wall and Baseboard Radiation, Convector Radiators, Unit Heaters, Back Draft Dampers, Air Cooled Condensers. Engineering and capacity data is included in catalogs mailed on request.



RITTLING FINNED TUBING

Steel tubing in 1 in., 1¼ in., and 2 in. IPS sizes. Steel tubing is supplied with steel fins. Copper tubing is available with 1 in. and 1¼ in. nominal sizes with aluminum fins. *Catalog FT55.*



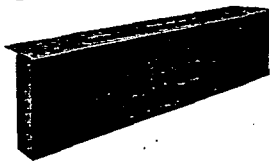
RITTLING 1200 HEAVY DUTY BASEBOARD

Heavy sheet steel cabinets with louvered inlet and outlet grilles. Designed for semi-recessing. Available with 1 or 2 heating elements for high output in short runs. *Catalog BB56A.*



RITTLING SIL-VECTOR

Attractive enclosures with rounded top, bottom, and ends. Outlet louvers located at top and front radius. Die formed louvered inlet panel at bottom provides strength, rigidity, and integral support for front panel throughout its entire length. *Catalog FT55.*



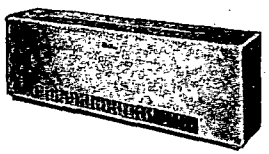
RITTLING SIL-N-WALL

Top of cabinet takes place of window sill. Front takes place of plaster finish under windows where unit is installed. Sizes fit standard window designs. Variety of heating elements in single, double, and triple height to provide needed output. Length and height as required. *Catalog FT55.*



RITTLING 485 BASEBOARD

RITTLING 485 Baseboard is only 6 1/16 in. low and projects 2 5/16 in. into the room—only 1½ in. when recessed to the depth of the plaster. Ideal for installation under picture windows. One-piece back plate has integral top and bottom cover. Covers are finished in neutral prime. Adjustable damper can be easily installed at any time. *Catalog BB-485.*



RITTLING SCHOOLMASTER UNIT VENTILATORS

Are complete heating and ventilating units with heating elements, circulating fans, air filters, and dampers to control the fresh air volume. Steel cabinets, available in five sizes, have baked-on enamel finish in a choice of colors. *Catalog UV-1157.*

MATCHING STORAGE CABINETS. Heavy steel cabinets match Schoolmaster Unit Ventilators to form an unbroken line when joined. Available with and without concealed heating element. Also furnished with locking doors.

ASK FOR YOUR COPY OF CATALOGS.

RITTLING BASEBOARD RADIATION. Rittling offers a complete line of fabricated steel baseboard radiation in five basic types to meet every requirement of residential and non-residential specifications. Wide choice of heating elements and enclosures provides sizes and designs to fit every installation condition. *Catalog BB-56.*

OTHER RITTLING PRODUCTS, NOT SHOWN, INCLUDE:

WALL RADIATION
CABINET UNIT HEATERS
UNIT HEATERS
CABINET CONVECTORS
SPACEMASTER HEATING AND VENTILATING UNITS
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Curtis Manufacturing Company

Refrigeration Division

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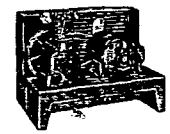
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OVER 100 YEARS OF SUCCESSFUL MANUFACTURING EXPERIENCE

REFRIGERATION • AIR CONDITIONING

Complete Line of Units from ¼ to 100 hp. Unit Coolers, Air Handling Units, Evaporative Condensers and Cooling Towers.

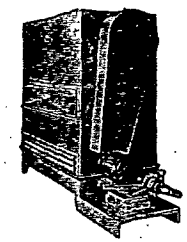
Air Conditioning Equipment, Packaged and Central Types. Complete Refrigerating Equipment for Dairies, Cold Storage Locker Systems, Walk-in Coolers, Drinking Water Systems, Commercial and Low Temperature Cooling and Processing.



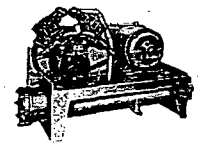
Combination air and water cooled Condensing Units. ⅓ through 3 hp.

COMMERCIAL REFRIGERATION

Air cooled condensing units from ¼ to 3 hp, inclusive, combination air and water cooled units from ⅓ to 3 hp, inclusive, and water cooled units from ½ to 100 hp, inclusive. All models charged with Freon 12 refrigerant. Mechanical advantages include Timken Bearings, Positive Pressure lubrication.



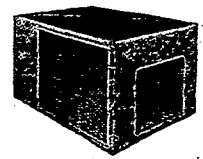
Evaporative Condensers, Cooling Towers, Air Handling Units. 3 to 100 tons.



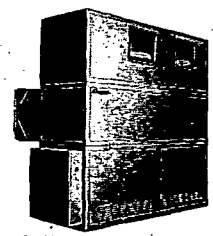
15 hp Cleanable Shell and Tube Condensing Unit. Other sizes from 7½ to 100 hp.

AIR CONDITIONING

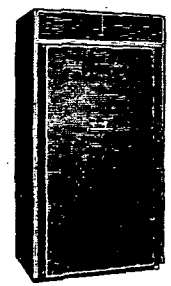
Curtis offers complete packaged air conditioning units. Air cooled condensing units with various types of matching coils can be added to present heating system. Water cooled units can be supplied with air cooled condenser. Water cooled units adaptable for heating.



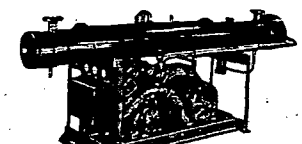
Air Cooled Type 3-5-7½ Tons



25-30-40-50 Ton Packaged Air Conditioners



3-5-7½-10-15 and 20 Ton Packaged Type Air Conditioner



7½ to 100 ton Packaged Liquid Chillers

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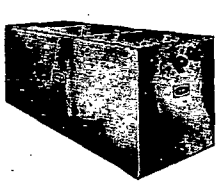
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Phone: RIchmond 7-5121

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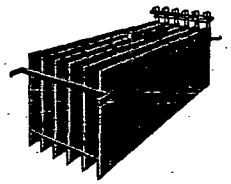
CANADIAN BRANCH
Dole Refrigerating Products Limited
29 Brock Street, North
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Ray F. Kempf, Manager
Gordon L. Timbers, Sales Representative

MANUFACTURERS OF AIR CONDITIONING AND REFRIGERATION EQUIPMENT

ICE-CEL ACCUMULATOR UNITS
LATENT HEAT ICE STORAGE SYSTEMS FOR AIR CONDITIONING



EXTERIOR ICE-CEL TANK



ICE-CEL INTERIOR X-UNIT

APPLICATION

DOLE ICE-CELS are used wherever the air-conditioning load is high for a relatively short period of time, such as in churches, mortuaries, auditoriums, lodge halls, cafeterias, offices, theaters, motels, retail stores.

OPERATION

The ICE-CEL consists of an assembly of DOLE Vacuum Plates submerged in a tank of water. A condensing unit is connected to and refrigerates the plates, causing ice to build up on their surface. A control is available to stop the condensing unit when the ice has reached any desired thickness. When cooling is to begin, chilled water from the ICE-CEL is pumped through a water coil in the air stream. In passing over this coil, the air gives up its heat to the water which is then returned to the ICE-CEL for re-chilling and re-circulation.

COMPRESSOR REQUIREMENTS

Since the compressor capacity can be spread over the whole day, if necessary, rather than just for the duration of the actual cooling period, a much smaller condensing unit, together with a correspondingly smaller cooling tower or evaporative condenser, can be used. This eliminates the need for water economizers. Also a compressor used for other purposes and not needed full time can have its idle periods devoted to building up ice on the ICE-CEL to be used while the compressor is doing its regular job. As a result, operating costs are kept at a minimum, especially in areas where demand power rates are in effect. The existing power service to the building is usually adequate.

HOLDOVER CAPACITIES AND BUILD-UP TIME FOR STANDARD ICE-CEL UNITS

STANDARD ICE-CEL UNITS		LHS-12 or LHS 12x		LHS-21 or LHS 21x		LHS-31 or LHS 31x		LHS-51 or LHS 51x	
Holdover capacity at given inlet water temp.	45°	18 T.H.		33 T.H.		48 T.H.		81 T.H.	
	40°	17 T.H.		31 T.H.		47 T.H.		77 T.H.	
	34°	13 T.H.		24 T.H.		36 T.H.		60 T.H.	
Comp. HP		Hrs.	S.T.	Hrs.	S.T.	Hrs.	S.T.	Hrs.	S.T.
¾ HP		26	19°						
1 HP		21	17°						
1½ HP		16	12°	25	19°				
2 HP		13	9°	20	15°	27	20°		
3 HP		11	8°	15	11°	20	15°	29	21°
5 HP				11	4°	14	10°	20	16°
7½ HP				10	1°	11	4°	15	11°
10 HP						10	0°	13	7°
15 HP								10	3°
20 HP								9	2°

NOTE: Add 10 percent if starting water over 50 deg
Add 5 percent if starting water between 40 deg-50 deg. S.T. indicates approximate suction temperature at end of ice buildup. Condensing units smaller or larger than shown not recommended.

ENGINEERING SERVICE

Complete Engineering Catalog BAE and further technical information available on request. Our staff of engineers will help you with your refrigeration problems at no charge or obligation. A DOLE application engineer is located near you to give you prompt service. Write, wire, or phone.

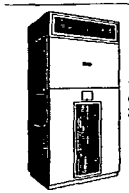
Frick Company

Waynesboro, Penna.

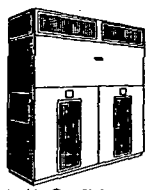
ATLANTA 3, GA. · BOSTON 15, MASS. · BUFFALO, N. Y. · CHARLOTTE 2, N. C. · CHICAGO 6, Ill. · CINCINNATI 2, OHIO · DALLAS 1, TEXAS · KANSAS CITY 8, MO. · LOS ANGELES 14, CALIF. · MEMPHIS 3, TENN. · NEW ORLEANS 12, LA. · NEW YORK CITY 17 · ORLANDO, FLA. · PALO ALTO, CALIF. · SAN FRANCISCO 4, CALIF. (sub-branch) · SEATTLE 1, WASH. (sub-branch) · PHILADELPHIA 4, PENNA. · PITTSBURGH 22, PENNA. · ST. LOUIS 3, MO. · WASHINGTON 5, D. C.

Distributors in Principal Cities

Air Conditioning, Refrigerating, Ice Making and Food Freezing Equipment



Unit Air Conditioners, 3, 5, & 7½ hp.



Unit Air Conditioners, 10 & 15 hp.

AIR CONDITIONING

Complete Frick systems, including original designs, in all sizes: also refrigeration for use with equipment supplied by others. Thousands of installations, the world over, attest the value of Frick air conditioning, which has 50 years experience behind it. See Frick *Bulletin 502* on hospital work; *505* on installations and systems; *522* on unit conditioners; and *146* on Frick air conditioned storages, which are revolutionising the cold storage industry. Estimates cheerfully furnished.

LOW-PRESSURE REFRIGERATION

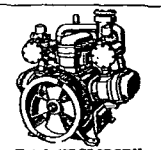
Small commercial units, "ECLIPSE" multi-cylinder compressors, and larger heavy-duty enclosed compressors, handling refrigerants 12 or 22, provide a complete and efficient line to meet your exact needs. We build unit air conditioners, central systems, plain and finned coils, water coolers, condensers, controls, valves and fittings: no job too special for Frick Engineers.

AMMONIA REFRIGERATION

Heavy-duty vertical compressors, with 2 or 4 cylinders, in sizes from 2½ to 1000 tons of refrigeration: handle refrigerant 22 equally well. Useful life of these dependable machines averages 40 years. Highly economical for cooling water on air conditioning, and for heat-pump service, in addition to regular refrigeration duty. Booster systems, developed by Frick Engineers 30 years ago, provide power savings at low temperatures. "ECLIPSE" compressors, with 3, 6, or 9 cylinders, also supplied for ammonia work.

SERVICE

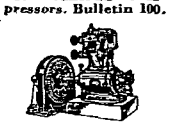
We supply the most complete service, including recommendations, estimates, design, manufacture, installation, test and maintenance. One responsibility, one superior standard. Quotations cheerfully furnished through Branch Offices and worldwide Distributor organizations.



Frick "ECLIPSE" Compressors, 2, 3, 4, 6 or 9 cylinders. Bulletin 100.



Combined Units with "ECLIPSE" Compressors. Bulletin 100.



Heavy-duty Industrial Compressors, 3 in. by 3 in. to 17¾ in. by 12 in. Bulletins 112, 516 & 651.

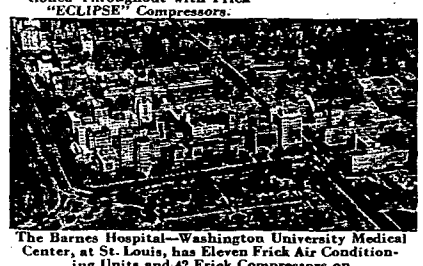


Low Pressure Refrigerating Units, Open Drive or Accessible-Hermetic Types. Bulletin 97.

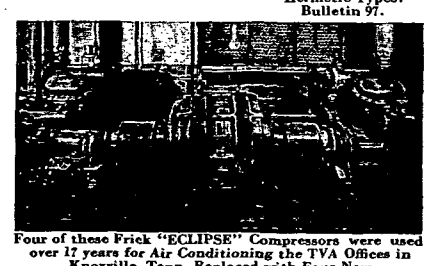


The Two Office Buildings of the American National Insurance Co. at Galveston, Texas, are Air Conditioned Throughout with Frick "ECLIPSE" Compressors.

DEPENDABLE REFRIGERATION SINCE 1882
FRICK CO.
WAYNESBORO, PENNA., U. S. A.



The Barnes Hospital—Washington University Medical Center, at St. Louis, has Eleven Frick Air Conditioning Units and 42 Frick Compressors on various Cooling Services.

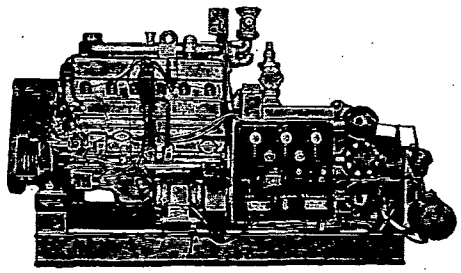


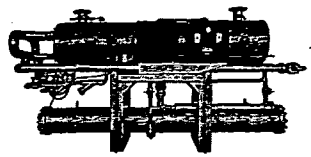
Four of these Frick "ECLIPSE" Compressors were used over 17 years for Air Conditioning the TVA Offices in Knoxville, Tenn. Replaced with Four New "ECLIPSE" Machines.

THE READY POWER CO.

DETROIT 14, MICHIGAN

NATURAL GAS PACKAGED AIR CONDITIONERS





GENERAL DESCRIPTION—Ready-Power Air Conditioning Units combine modern, Freon compressors with medium-speed industrial-type gas engines. These packaged gas air conditioning units are performance-tested under actual operating conditions to insure long, trouble-free, service.

These units operate on clean natural gas. Engine carbon deposits are nearly non-existent. These natural gas-carbureted engines last 3 to 5 times longer than similar engines operated on gasoline.

OUTSTANDING ECONOMY—Less than 13 cu ft of low-cost natural gas is required per ton hour. Fuel consumption is in direct proportion to the actual load throughout over 80 percent of the capacity range. A power take-off pulley is provided on the engine shaft to provid for belt-driven tower pumps. The economy combination of low fuel requirement, engine-powered tower circulation, and "matched to the load" fuel consumption makes Ready-Power outstanding in net refrigeration costs.

AUTOMATIC OPERATION—All Ready-Power Units are available with either Thermostat Start or Push-button Start Controls. All models are equipped with a-c electric starting motors and automatic LOAD-MATCH capacity controls.

Thermostat Start Units start and stop automatically in response to the setting of your thermostat. Push-button start units must be started and stopped manually.

AUTOMATIC CAPACITY CONTROL—Automatic Capacity Control is achieved by varying the operating speed and by unloading compressor cylinders after the unit has dropped to the minimum operating speed. This means that the pumping rate of the compressor is exactly matched with the cooling load. In this manner, maximum operating economy is realized.

This "LOADMATCH" capacity control system permits almost continuous compressor operation at a constant evaporator coil temperature, the result of which is extremely close control of humidity and temperature without the need for large re-heat systems.

COMPRESSORS—Modern precision balanced Freon 12 and 22 compressors assure a minimum of vibration. The Ready-Power control system automatically unloads compressor and permits start-up at idle speed with all cylinders unloaded. Approximately 2 to 3 minutes are required to fully load the compressor after start-up. This greatly decreases start-up impact loading on valves, bearings, etc., and substantially increases compressor life over similar equipment operating at synchronous constant speeds of 1750 rpm or higher.

COMPACT AND READY TO RUN—Complete "packaging" of auxiliary controls and accessory items with the direct driven compressor unit permits quick, easy installation in a space measuring only 3 ft × 7 ft.

MATCHED WATER CHILLERS—Ready-Power offers completely balanced and tested refrigerant systems for water-chilling applications. Matched chiller-condenser units are designed to take full advantage of the variable capacity feature of the engine compressor units. The chiller and condenser are mounted on a heavy structural steel frame assembly, completely piped and wired. A control cabinet houses the necessary related solenoids relays and switches and other controls.

LONG SERVICE LIFE—Powered by medium-speed industrial engines, these systems have achieved an outstanding reputation for dependability and long service. The compressors represent the latest developments in design, engineering, and manufacture, and are long-wearing and smooth-running. Units are equipped with automatic engine oil level controls and closed engine water cooling circuit.

PHYSICAL DATA ENGINE COMPRESSOR PACKAGED UNITS

ENGINE-COMPRESSOR No.	R220C5812	R220C5822	R220C9712	R308C13612	R308C9723	R450C13622
Compressor cylinders	3	3	5	7	5	7
Nominal capacity	22 tons	35 tons	57 tons	51 tons	60 tons	76 tons
Refrigerant	F-12	F-22	F-12	F-12	F-22	F-22
Length—Overall	6'10"	6'10"	7'2"	7'2"	7'2"	7'9"
Width—Overall	2'9"	2'9"	2'9"	2'9"	2'9"	2'9"
Height—Overall	4'1"	4'1"	4'1"	4'3"	4'3"	4'4"
Length of Base	5'10"	5'10"	5'10"	5'10"	5'10"	6'1"
Width of Base	2'6"	2'6"	2'8"	2'0"	2'0"	2'6"
Shipping Weight (approx.)	2450 lb	2450 lb	2800 lb	3000 lb	2900 lb	3300 lb

PHYSICAL DATA MATCHED CHILLER CONDENSER UNITS

Chiller Model No.	RCP20	RCP25	RCP30	RCP40	RCP50	RCP60	RCP65	RCP80	RCP100	RCP120	RCP130	RCP140
Nominal Capacity	20 T.	25 T.	30 T.	40 T.	50 T.	60 T.	65 T.	80 T.	100 T.	120 T.	130 T.	140 T.
Length	11'	12'	12'	12'	14'	12'6"	12'	14'	17'	15'	14'	14'
Width	2'	2'	2'	2'	2'	2'	2'	4'	4'	4'	4'	4'
Height	5'	5'	5'	5'	5'	5'6"	6'	6'	6'	6'	6'	7'
Shipping wt. Approx.	1860	2000	2340	2050	3440	3740	4040	4960	5730	5730	6450	10960

This equipment data and information is included for convenience in rough analysis, layout, and application. It is not guaranteed accurate for final detailed design. For current design information consult the nearest manufacturer sales office or your local Gas Utility Company.

Aerovent Fan Company, Inc.

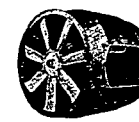
Ash and Blade Sts. Piqua, Ohio

Efficiency-engineered industrial fans and air handling equipment



DIRECT CONNECTED

Heavy-duty units for industrial exhaust or air supply. Steel ring or panel orifice. Size 9 in. to 96 in. with 2, 4 or 6 blades. For capacities to 145,000 cfm. Also available in Extended Shaft and Pulley types. Write for *Bulletin 100.*

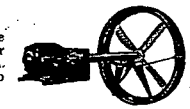


SEMI-PRESSURE FANS

Standard units with new 7-blade propellers move more air with less power at resistances to 4 in. S.P. Sizes 14 in. to 60 in. Five types. Shown is "Bi-Flo" Duct Fan. *Bulletin 400.*

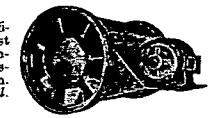
EXTENDED SHAFT

For high temperature, moisture or corrosive conditions. Ring or panel orifices. Sizes 12 in. to 84 in. with 2, 4 or 6 blades. Capacities to 120,000 cfm. *Bulletin 100.*



VANEAXIAL

Designed for maximum efficiency and economy against resistances to 6¼ in. S.P. High-velocity, straight-line air discharge. Sizes 12 in. to 48 in. Capacities to 77,000 cfm. *Bull. 450.*





NEW "LS" FANS

Belted and Direct Driven Ring, Panel or Duct types with new 4-blade "Macheta" Airfoil Propellers for peak performance at resistances to 1½ in. S.P. Sizes 24 in. to 48 in. for capacities to 46,000 cfm. Ask for *Bulletin 750 and 775.*



AIR MAKE-UP UNITS

Tempers and filters outside air entering working areas. Steam, hot water and gas-fired types. Sizes 24 in. to 54 in. 2, 4 or 6 blades. Capacities to 45,000 cfm. *Bulletin 850–875.*

PORTABLE UTILITY

For personnel cooling, product drying and processing and fume removal. Adjustable for 180 degree discharge. Sizes 16 in. to 36 in. for capacities to 20,450 cfm. *Bulletin 250.*





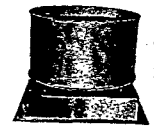
CENTRIFUGAL BLOWERS

For materials-handling, dust-collecting or exhaust systems. Reversible rotation. Adjustable discharge. 19 in., 26 in., 33 in., 40 in. and 51 in. wheel diameters. *Bulletin 700.*



MANCOOLERS

For spot cooling of men and machines, exhaust and general air circulation. Cradle, pedestal and 2 or 4-wheeled types. Portable, adjustable for 180 discharge. Sizes 24 in. to 48 in. with 2, 4 or 6 blades. Capacities to 40,000 cfm. *Bulletin 250.*



ROOF VENTILATORS

Three models, featuring rugged all-weather construction with new 'low-level' design. Sizes 12 in. to 72 in. 2, 4 or 6 blades. Capacities to 93,500 cfm. *Bulletin 600.*

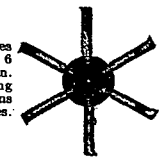
DIRECT-DRIVEN DUCT

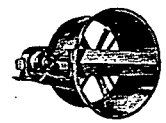
Preferred by industry for high-velocity air circulation and exhaust of non-hazardous atmospheres. Sizes 12 in. to 96 in. 2, 4 or 6 blades. Capacities to 145,000 cfm. *Bulletin 150.*



LARGE DIAMETER PROPELLERS

Adjustable-pitch aluminum alloy blades with cast aluminum hubs in 4 and 6 blade types and sizes 54 in. to 144 in. New AL-Metal propellers for cooling tower and heat exchanger applications in sizes 14 in. to 18 in. with 4 or 6 blades. *Bulletins 500 and 510.*





BELT-DRIVEN DUCT

Recommended for handling dusts, fumes, vapors, high temperatures and hazardous atmospheres. Sizes 12 in. to 96 in. 2, 4 or 6 blades. Capacities to 145,000 cfm. *Bulletin 160.*



PENTHOUSES

Weather-tight enclosure for exterior mounted fans. Eliminates downdrafts; provides safe exit for fumes and vapors. Sizes 16 in to 72 in. Shutters optional. *Bulletin 650.*

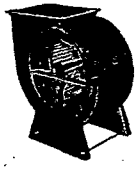
MEMBER *AIR MOVING AND CONDITIONING ASSOCIATION (AMCA)*

Air deliveries of Aerovent equipment are determined in accordance with established and accepted codes and each unit is guaranteed by the manufacturer to deliver rated performance.

CERTIFIED PERFORMANCE

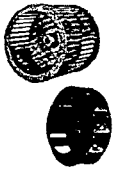
AIRFAN ENGINEERING COMPANY

7401 Telegraph Rd.
Los Angeles 22, Calif., U. S. A.
(Telephone RAymond 3-3354)



←HEAVY DUTY BLOWERS

Forward and backward curved wheels. Single inlet single width and double inlet double width. Wheel diameters range in size from 12 in. to 60 in. Any rotation, discharge or arrangement.



BLOWERS AND SCROLLS→

Made for original equipment manufacturers of air handling units and evaporative condensers. Heavy duty riveted construction, center or end suspension wheels. 12 sizes; 10 in. diameter to 30 in. diameter, standard or hot dipped galvanized.



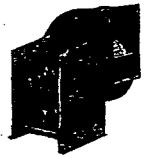
←STANDARD BLOWERS

Medium duty blower assembly made in single or double width, sizes 10 in. to 30 in. wheel diameter, and features universal discharge arrangement. Motor mounting in any location.



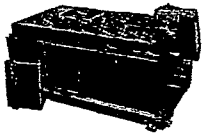
AIR WASHER EVAPORATIVE COOLER →

With slinger spray type water distribution system as part of packaged unit, assembled ready to use. Special features of design insure high cooling efficiency and lower maintenance costs. Features such as fiber glass evaporative and eliminator filters washed continuously by efficient slinger spray.



←UTILITY FANS

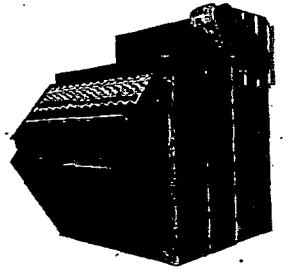
For general ventilation, complete with weather-proof covers. Self contained package units ready to run, easy installation. 300 cfm to 20,000 cfm. Wheel diameters 10 in. to 30 in. Forward and backward curved wheels.



AIR HANDLING COMBINATION UNITS→

Offering a complete line of air handling and comfort-conditioning units, utilizing direct expansion, water and steam coils for cooling and heating. Wide selection, both in vertical and horizontal arrangements. Capacity range from 1600 cfm to 35,400 cfm with special emphasis on quiet operation and thermal-acoustical insulation.

MULTIZONE AIR CONDITIONERS→



With the **Heat Supplied by AGA Approved Gas Fired Furnaces** from 200M-Btu to 1,200M Btu Input. Available in horizontal and vertical arrangements—Twelve sizes with capacity from 2500 to 35,000 cfm. Cooling coils from 6 to 60 square feet. Supplies air mixtures blended for simultaneous unequal heat load demands up to 25 individual zones from single unit.

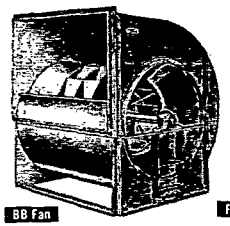
ALADDIN Heating Corporation

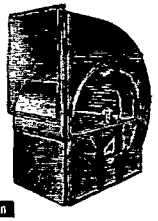
Aladdin AIR HANDLING EQUIPMENT

1111 West Avenue 137, San Leandro, California

Manufacturers of Centrifugal Blowers, Heating and Ventilating Equipment

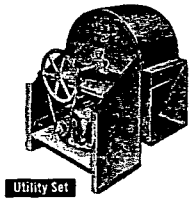
BB FANS—Backwardly inclined blade fan with non-overloading horsepower characteristic. Manufactured in 22 sizes, single and double width, all arrangements, 2 classes of construction. Request *Bulletin 605-A, 605-B.*

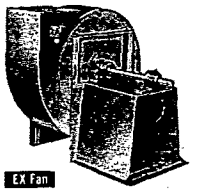

BB Fan


FC Fan

FC FANS—Forward curved fan, slow speed for quiet operation. Manufactured in 22 sizes, single and double width, all arrangements, 2 classes of construction. Request *Bulletin 605-A, 605-C.*

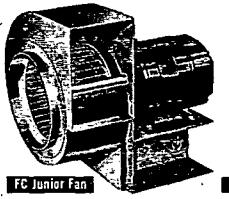
UTILITY SETS—Self contained units for a wide variety of ventilating applications. Manufactured in 14 sizes with forward curved rotors, 12 sizes with backwardly inclined blade rotors, single width only. Request *Bulletins 610-A, 611-B.*

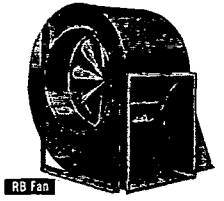

Utility Set


EX Fan

EX FANS—Radial blade fan used chiefly for conveying materials, fume exhaust, etc. Manufactured in 15 sizes, single width only, all arrangements, 5 classes of construction, 3 wheel types. Request *Bulletin 461.*

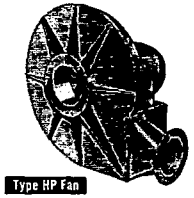
FC JUNIOR FANS—*Self contained direct connected fans with* forward curved rotors. Manufactured in 5 sizes with 13 motor speeds, single width only. Request *Bulletin 422.*

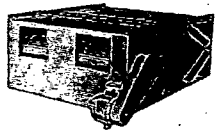

FC Junior Fan


RB Fan

RB FANS—Radial blade fan well suited for handling grease or other sticky materials. Manufactured in 13 sizes, single width only, all arrangements, 2 classes of construction. Request *Bulletin 450-C.*

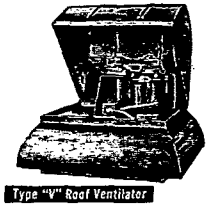
HP FANS—Designed for applications requiring low air volumes at high pressures. Manufactured in a variety of combination sizes. Request *Bulletin 470.*

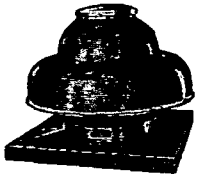

Type HP Fan


Type "M" Fan

TYPE "M" FAN UNITS—Self contained units for a wide variety of commercial and industrial requirements. Manufactured in 7 standard sizes, units are available with accessory face and by-pass dampers, filter boxes and mixing boxes. Request *Bulletin 510-A.*

TYPE "V" ROOF VENTILATORS—Backwardly inclined blade centrifugal type. Manufactured in 13 belt driven and 12 direct driven sizes with steel, aluminum, copper or stainless steel housings. Request *Bulletin 410-B.*


Type "V" Roof Ventilator


Type "S" Roof Ventilator

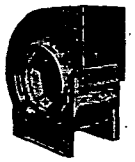
TYPE "S" ROOF VENTILATORS—Backward inclined blade centrifugal type. All units fabricated from spun aluminum. Available in both direct connected and belt driven models. Request *Bulletin 411.*

BAYLEY BLOWER COMPANY

Engineered Air-Handling Equipment

1821 S. Sixty-Sixth St. Milwaukee 14, Wis.

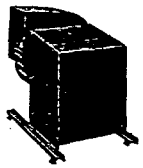
Representatives In Principal Cities



Type AP Ventilating Fan

GENERAL VENTILATING & AIR-CONDITIONING FANS

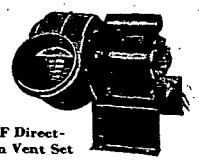
Bayley Type AP Fans have backward-inclined blades, non-overloading power characteristic. They are made in Classes I, II, III & IV for air-conditioning systems, plant and building ventilation, process air-handling, mechanical draft. Sizes from 12¼-in. to 132½-in. wheel diameter, capacities from 1000 to 700,000 cfm, all standard drive arrangements, single and double width. Complete line of modifications and accessories. Sizes up to 365 (36½-in. wheel diameter) have reversible housings, facilitating prompt delivery of any discharge position in popular drive arrangements. All standard housings are made with scroll seams inside-welded for strong, tight construction, smooth exterior.



Belted Vent Set

BELTED VENTILATING SETS

Compact unitary sets, complete with drive and motor, serve a wide variety of applications, such as kitchens, gymnasiums, lavatories, garages, restuarants, hospitals, laboratories, industrial plant areas. Sets may be used for supply or exhaust ventilation. They are available in wheel diameters from 9-in. to 36½-in., for capacities from 500 to 15,000 cfm. Smaller sizes have Type F forward-curved wheels, larger sizes have Type AP backward-inclined wheels, as required for most suitable drive selections. Wheels and housing are of same design and construction as regular Class I fans. Protective drive covers and automatic outlet louvres are available for outdoor installation.



Type F Direct-Driven Vent Set

DIRECT-DRIVEN VENTILATING SETS

Sets with forward-curve wheel mounted on motor shaft are particularly suited for light-duty low-capacity requirements. Wheel sizes from 4½-in. to 10½-in., capacities from 100 to 1500 cfm. Larger sizes made to order as regular arrangement 4 fans. Typical applications include small rooms, booths, cabinets, compartments, equipment cooling. Housings and wheels are designed and built for permanency under hard usage. Inlet screens and filters are available. Also can be equipped with motor cover for outdoor installation. These sets are particularly recommended for incorporation as ventilating or cooling fans in factory-built apparatus.



Type EX Industrial Exhaust Fan

INDUSTRIAL EXHAUST FANS

Type EX Fans are of the radial-blade medium speed design best suited to the wide diversity of industrial air-handling requirements. Sizes range from 5-in. to 60-in. inlet diameter, 8-¾-in. to 104¼-in. wheel diameter. 200 to 55,000 cfm capacity. Style GP open, back-plate wheels are recommended for the majority of materials handling applications. Style AH closed wheels are designed for maximum efficiency when handling clean air or gas. Many modifications and special construction features can be incorporated for unusually severe conditions of abrasion or corrosion. Temperatures up to 1450° F can be handled with appropriate bearing selections and heat flinger arrangements.



Type H Pressure Fan

PRESSURE FANS

Type H Fans are proportioned to suit high pressure/volume ratio characteristically required in certain phases of pneumatic conveying, cupola blast, burner draft, suction or blast cleaning, and the like. Usual pressure range is from 12-in. WG to 55-in. WG (2 psi). Four designs are offered: No. 1, to 21-in. SP; No. 2, over 21-in. to 28-in. SP; No. 3, over 28-in. SP to 35-in. SP; No. 4, over 35-in. SP. While basic sizes and proportions are established, the necessary flexibility of selection for the wide range of pressure volume combinations is provided by varying widths of fans as required. Wheel diameters are also variable in sufficient degree to achieve proper operation at standard direct-connected motor speeds.

BROOKSIDE CORPORATION

McCordsville, Indiana

Manufacturers of: Fans • Blowers • Ventilating Equipment

Brookside Corporation is large enough to serve you well—small enough for frequent contact between management and production supervisory personnel to insure top quality and coordination of all customer requirements. Brookside offers custom engineering free of charge—and production at competitive prices. A company-owned airplane is maintained to offer the fastest possible service and engineering assistance when it is necessary.



4-Blade with Ring

M-PEL-AIR 6-BLADE ONE-PIECE FANS; One piece fans in a wide range of sizes from 6 in. diameter 4-Blade to 14 in. diameter 6-Blade. Some models with slinger rings for room air conditioner application. Manufactured of aluminum or steel.



4-Blade without ring



6-Blade without Ring

M-PEL-AIR AUTOMOTIVE TYPE: Heavy-duty automotive type fans, both single and double spider types, in both steel and aluminum blades. A wide range of sizes from 10 in. diameter through 85 in. diameter. Available in 4, 5, 6, and 8 blade designs. Suitable for use on gasoline or diesel engines and cooling tower applications.



Offset Type

M-PEL-AIR ROOF EXHAUSTERS: Available in a wide range of models and types ranging from 250 cfm to more than 11,000 cfm. Spun aluminum construction with aerodynamic design for greater efficiency. Entire line features extremely low silhouette. Backward inclined, non-overloading centrifugal fan wheels assure quiet operation.



Slinger Type

M-PEL-AIR GRAVITY ROOF EXHAUSTERS: Engineered and designed for fresh air intake or relief of pressure within building. Lightweight, spun aluminum housing construction. 8 Models. Motor operated or spring loaded back-draft eliminators, cylindrical bird guards, bug screens optional on all models.



Direct Drive Exhauster

M-PEL-AIR BELT DRIVE ROOF EXHAUSTERS: Power unit is resilient-mounted for quiet operation. Fan wheel design, unrestricted air flow make for quiet operation. Rigid spun aluminum housing. All belts, pulleys, bearings and motors standard sizes. Low silhouette blends well with contemporary architecture.



4-5-6 & 8-Blade Automotive Type

M-PEL-AIR FAN BLADES: Slinger-type for air conditioning or offset spider and conventional propeller type fans for ventilating and other applications. All engineered exactly to your specifications free of charge. Slinger rings are all one-piece die drawn. Research and laboratory facilities available to solve air moving problems on your product.



Gravity Roof Exhausters



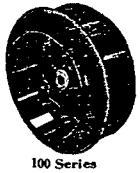
6-Blade One Piece

M-PEL-AIR SIX BLADE: One-piece or fabricated construction in a wide range of diameters, any pitch, in aluminum or steel. Gives efficient air movement against higher static pressures. Solid or resilent hub. Also available in three and four blade.



Belt-Drive Exhausters

M-PEL-AIR OFFSET TYPE: Gives maximum space-saving features by bringing blade around the motor. Brookside's advanced design blades give higher efficiency and may be made reversible.



100 Series

M-PEL-AIR SLINGER BLADE: Solid die-drawn slinger rings, with the tabs a part of the blade, operate at low sound level and high efficiency. Designed especially for air conditioning and refrigeration industry.



200 Series

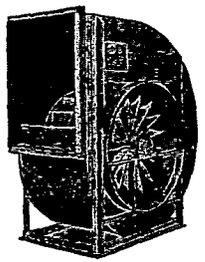
M-PEL-AIR 100 & 200 SERIES BLOWERS: 100 series specifically designed for higher pressure applications. Rugged construction with conical shaped inlet ring welded to blades. Available from 14 to 60 inch diameter. Series 200 designed for lower pressures. Fabricated of aluminum, stainless, or steel, riveted throughout. Both series have backward inclined blades for low noise and non-overloading characteristics and are statically and dynamically balanced.

BUFFALO FORGE COMPANY

450 Broadway, Buffalo, N. Y.

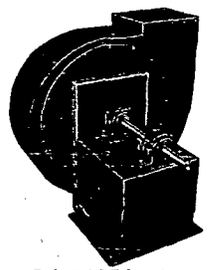
Canadian Blower & Forge Co., Ltd., Kitchener, Ont.

Ventilating Fans (centrifugal, axial flow and propeller), Mechanical Draft Fans, Blowers, Exhausters, Air Washers, Drying Equipment, Air Conditioning and/or Cleaning Equipment, Unit Heaters, Air Preheaters, Packaged Power Roof Ventilators, Makeup Air Units, Spray Nozzles. "Buffalo" is ready to work with you on any air problem you may have. Call the "Buffalo" Engineering Representative in your nearest principal city, or write for *Bulletins* mentioned below.



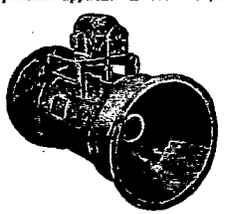
Type "BL" Fan

TYPES "BL" AND "BLH" *VENTILATING* FANS. Quiet, efficient, non-overloading fans with completely stable performance from free delivery to shut-off. Type "BL" Fans are designed for Classes I-II operation (moderate pressures). Type "BLH" Fans are designed for high pressure service covered by Classes III-IV. To provide minimum turbulence both fans have smoothly-curved inlet bell with directional guide vanes, wheel with full curvature shroud matching the inlet bell and streamlined, wheel-contoured housing. Type "BLH" Fans have divergent outlets, further minimizing turbulence. Capacities: 1,000 to 500,000 cfm. **Write for Bulletins F-104 and F-200.**



Industrial Exhauster

INDUSTRIAL EXHAUSTERS. All-welded steel plate construction is used in these heavy-duty fans. This provides smooth, rivet-free interior surfaces for minimum friction. Ease of maintenance is afforded by removable front and back plates. Available with interchangeable Air Wheels for airborne dusts, gases, fumes, or with Material Wheels for grit, sawdust grain, stringy materials. Corrosion-resistant construction can be specified. Available with heat slingers to handle gases up to 1000°F in induced draft and other high temperature applications. 600 to 37,000 cfm capacities. Write for **BULLETIN FI-110.**

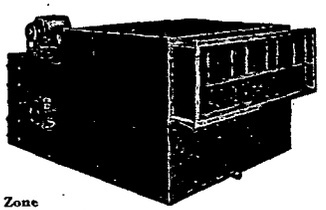


Axial Flow Fan

AXIAL FLOW VENTILATING FANS. These lightweight fans fit into straight duct runs, requiring no more room than a section of round duct for ventilation, exhausting, system boosting, draft and circulation applications. Extremely efficient, quiet and non-overloading. In Vaneaxial fans, guide vanes redirect swirling air at fan outlet, converting it into increased forward thrust. Also available in Tube axial models (without directional vanes). Drives include direct connected motor-in-housing, or V-belt drive with motor out of air stream. Models for high or low temperatures, non-sparking and/or corrosion-resistant construction. Capacities: 2,000 to 65,000 cfm. **Write for BULLETIN 3720-A.**

AIR CLEANING EQUIPMENT. Air Washers for cleaning, humidity control, heating and cooling are only one of the many types of units built by "Buffalo." Also available are highly specialized units for abating nuisance effluents in the chemical and process industries. More than a half-century of experience in this field is at your service, plus our complete manufacturing facilities. Write for Bulletins on industry-proved "Buffalo" equipment to solve any air cleaning or processing problem you may have.

UNIT HEATERS. Widely-used for versatile, economical, efficient "spot-heating". One-piece, seamless copper coil is continuous from inlet to outlet discharge ground joint. This makes "Buffalo" Breezo-Fin Heaters desirable for high pressure steam systems because it eliminates all possibility of leakage inside the unit. Safety is insured for steam pressures up to 250 lbs. These units also operate on as little as 2 lbs steam. Adjustable louvers aim heat where needed; fins are spaced for maximum radiation; high-efficiency fans give quiet operation. Write for **BULLETIN 3137-E.** Also, a wide selection of heating and ventilating units, with or without filtered intake and by-pass dampers, suspended and floor models for large areas.



Zone Control Cabinet

ZONE CONTROL CABINETS. These quiet, compact, efficient units are designed for installations where it is most feasible to use a single unit to provide varying degrees of conditioned air for several zones. Temperature is controlled by mixing cooled and heated air in correct proportions to suit each conditioned space. No need for separate reheat coils and their controls! Sectionalized for easy handling and installation. Convenient to service, with handy access doors, removable panels and outside-located bearings. Space-saving design with floor or ceiling clips. Can be adapted to any combination of heating, cooling and/or dehumidifying. **Write for BULLETIN AC-220.**

CENTURY FAN & VENTILATOR COMPANY, Inc.

45 Cedar St., Stamford, Conn.

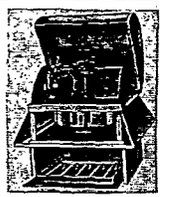
Plant #2 Torrington—Burrville, Conn.

Member of Air Moving Conditioning Association

Over 25 years experience and engineering development behind its time-proven performance

CERTIFIED RATINGS — ASTORCERTIFIED BY AMCA

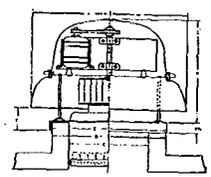
CENTURY CYCLONE ROOF FAN



A heavy duty centrifugal roof fan—compact, *symmetrical* and weatherproof design. High efficiency, large volume capacity, slow operating speed and quiet performance *makes the cyclone roof* fan excellent for schools and institutional type buildings.
Housings supplied in galvanized steel, aluminum, copper or stainless steel. Basic frame work constructed of heavy steel that guarantees long life and vibrationless operation.
Motors belt or direct driven in separate compartment—out of air stream.
Fan wheels are statically, dynamically and electronically balanced. Heavy plate construction, backwardly curved non-overloading type also furnished in non-sparking aluminum construction.
Motor and fan assembly suspended on rubber vibration eliminators.

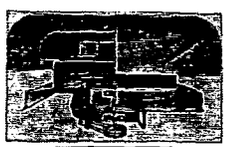
CENTURY SPUNAIR ROOF FAN





Designed to meet the modern architectural detail of low, one story buildings—streamlined gleaming appearance, low contour and simplicity of design.
Housing is of heavy gage spun aluminum—completely weatherproof. Designed to permit straight flow exhaust with venturi *inlet to minimize* turbulence.
The fan wheel is of heavy gage rugged construction—backwardly inclined, non-overloading type, statically and dynamically balanced, and quiet in operation.
Motor and fan are resiliently mounted out of air stream in ventilated compartment.
Spunair fans are made in direct and belt drive. Also furnished less motor and fan for intake hood.
Self-acting or electric operated damper.

TYPE MVF ROOF FAN



Meets the requirements of extra large exhaust volume, incorporating low contour design and rugged construction.
Housing made of galvanized steel, aluminum, copper or stainless steel.
Heavy duty propeller—four blade—for low static pressure up to ⅜ in., and eight blades for static pressure up to ¾ in. W.G.
Direct and belt drive—hinged hood for accessibility. Supplied with self-acting or motor operated damper.

CENTURY BLOWER



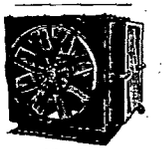
Utility Blowers FC and BC—forward curved and backward curved non-overloading type wheels made in sizes 6 in. to 36 in. wheel diameter.

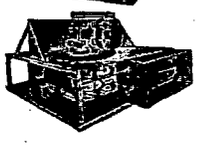
Sizes 12 in. to 90 in. wheel diameter

Class 1 Blowers	up to 3¾ in. S.P.
Class 2 Blowers	3¾ in. to 6¾ in. S.P.
Class 3 Blowers	6¾ in. to 9¾ in. S.P.

Heavy construction, manufactured in accordance with *AMCA* code of standardized arrangement, and testing procedure.
Newly designed high efficiency blower wheels are statically and dynamically balanced on the latest electronic balancing equipment.

CENTURY DRAFT INDUCERS





Induced Draft Fans provide the proper draft without the use of high chimneys.
TYPE MID Inducer is an axial flow, non-overloading type fan, which is easily installed on the boiler breaching.
Bearings and shaft are protected from heat by a double shell shaft tube with impeller blade to keep a steady flow of cool air over the shaft and bearings.
Adjustable barometric damper with self contained explosion relief valve.
Motor is standard single or two speed belt driven, and protected with ventilated aluminum heat shield.
THE AUTOMIC INDUCER is a backward curved, non-overloading centrifugal type blower for heavy duty service, and used when high static pressure is encountered, such as old flues of undetermined resistance.
Motor may be single or two speed, and with adjustable sheaves. When used with sequence controls, the draft will be constant regardless of conditions encountered.

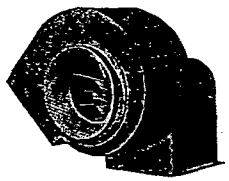
CHAMPION BLOWER & FORGE CO.

LANCASTER, PA.

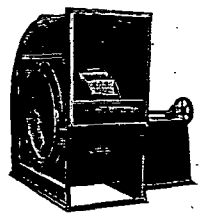
Address Correspondence to Div. 9

Manufacturers of Blowers, Ventilating Fans and Exhaust Fans for handling air, material and fumes

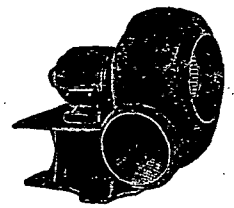
Representatives in Principal Cities



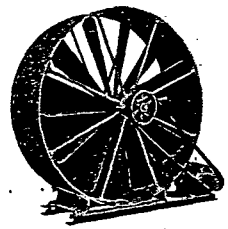
Champion Type D vent sets for general ventilation of all types equipped with backward curved non-overloading wheels up to 30 in. diameter, furnished complete with electric motor, adjustable drive and weather proof covers if required.



Champion Forward and Backward Curve Ventilating Fans and Exhaust Blowers manufactured in sizes up to 60 in. diameter. Fans also furnished in all standard arrangements with Ball, Babitted or Water Cooled Bearings as required.



Champion Cast Iron Housed Fans of various types for fume hood exhaust and for handling corrosive gases. Can be equipped with either plain steel or non-ferrous blast wheels, interior of housing and wheel can be covered with special corrosive resisting coating.



Champion Ventilating Fan for moving large volumes of air in either direction by reversing the rotation of the fan wheel. Develops pressures over 1½ in. w.g. Built in sizes from 18 in. to 84 in. diameter. Equipped with double Ball Bearings.

CHICAGO BLOWER CORP.

9871 Pacific Avenue, Franklin Park, Illinois

Blowers, Fans and Exhausters for Every Air Movement Requirement

AIRFOIL BLADED CENTRIFUGAL FANS.

for heating, ventilating, air conditioning systems

Aerodynamically-shaped blades move more air with less turbulence, thereby achieving higher efficiencies with reductions in noise and current consumption.

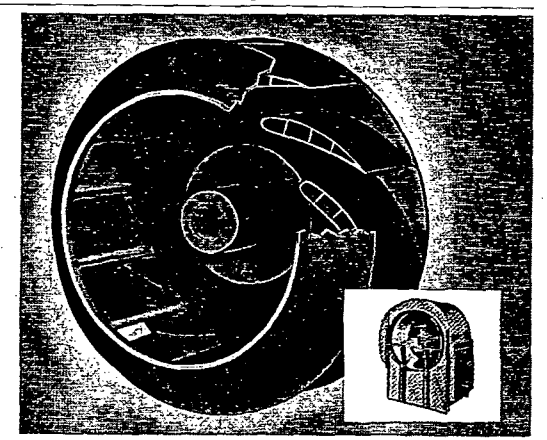
efficiencies up to 92 percent

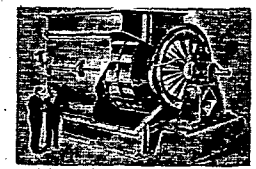
noise reduction up to 40 percent

horsepower savings up to 20 percent

Capacities: 440 to 561,680 cfm up to 32 in. S.P. **Sizes:** 12¼ to 80¾. Class I to V arrangements for heavy and standard service. Direct or belt drive. Single or double inlets and widths. Adjustable and fixed housings for all discharge positions. Specified since 1951 for:

- high pressure and couventional ventilation.
- air supply and exhaust.
- industrial processing.
- mechanical draft.

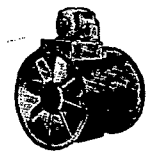




AIRFOIL MECHANICAL DRAFT

for power stations, heavy industry

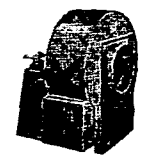
All performance advantages of Airfoil Fan. **Capacities:** up to 1,097,940 cfm for temperatures to 900F; ¼ to 32 in. S.P. **Sizes:** 24½ to 80¾. Single or double width, with or without inlet boxes. Welded heavy steel construction.



AXIAL AIRFOIL FANS

for exhausting fumes, hot gases; ventilation

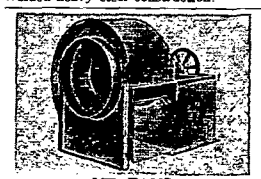
Welded airfoil type blades provide higher pressure characteristics, less noise and greater strength. **Capacities:** 1100 to 105,000 cfm up to 5 in. S.P. **Sizes:** 12 to 72. Orifice ring, duct, cradle, pedestal.



STEEL PLATE EXHAUSTERS

for material and dust handling

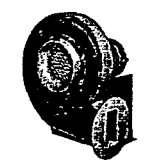
Separate wheels handle wood shavings, grinding dust, paper, fibrous and heavy materials. **Capacities:** 200 to 50,000 cfm up to 15 in. S.P. for temperatures to 1200° F. **Sizes:** 15 to 100. Arrangements for belt and direct drive. Adjustable.



MD FANS

for ventilating industrial, commercial, public buildings

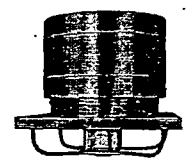
Large volume and low pressure types housed in cast iron or heavy steel. Welded backward or forward curved wheels. **Capacities:** 300 to 17,000 cfm up to 1½ in. S.P. **Sizes:** 62 to 300. Belt or direct drive, adjustable discharge.



TURBO PRESSURE BLOWERS

for combustion, air, forging, pneumatic systems

Supplies steady, high speed air pressure or suction. **Capacities:** 125 to 5250 cfm at pressures from 4 to 28 oz up to 750° F. Universal discharge adjustments. Direct or belt drive.



POWER ROOF VENTILATORS

for industrial, commercial ventilation

Packaged high velocity type. Welded axial airfoil wheel. Automatic butterfly damper—has emergency opener. **Capacities:** 6640 to 53,700 cfm up to ¼ in. S.P. **Sizes:** 24 to 60. Steel, aluminum or stainless housings—flat or pitched bases. Weatherproof motor. Others on request.

Technical Data Bulletins Available Upon Request

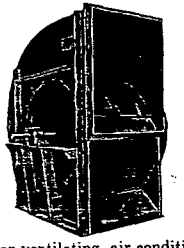
CLARAGE FAN COMPANY

Kalamazoo, Michigan

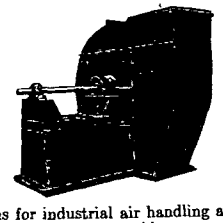
Air Handling and Conditioning Equipment

Sales Engineering Offices in All Principal Cities

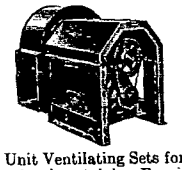
Since 1913 Clarage has been a leading manufacturer of air handling and conditioning equipment. By specializing—building a *complete* line of this equipment exclusively—Clarage has a concentrated "know how" that means important dividends to the user. (See "Air Conditioning—Central Systems" for other Clarage products.)



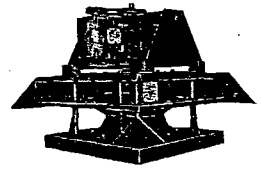
Clarage Fans for ventilating, air conditioning, and industrial applications—where clean air is handled at low pressures—are offered in many different designs, arrangements, and wheel types. Capacities range from 140 to 725,000 cfm.



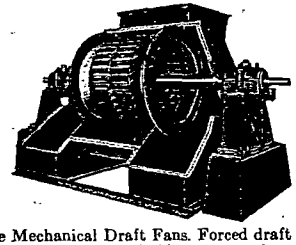
Clarage Fans for industrial air handling and material conveying are available in a wide range of types and sizes. This broad line includes cast iron fans, fans for pressures to 50 in. sp, and fans for temperatures to 1000 F.



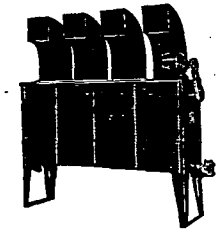
Clarage Ready Unit Ventilating Sets for smaller ventilating supply and exhaust jobs. Furnished complete, ready-to-run for both direct drive and V-belt drive. Capacities 150 to 14,000 cfm.



Clarage Centrilator Power Roof Ventilators (shown without cover) are V-belt driven. Capacities 1,500 to 26,500 cfm. Directilators are direct drive units. Capacities 100 to 3,500 cfm.



Clarage Mechanical Draft Fans. Forced draft and induced draft fans are available to meet virtually any capacity, pressure, and speed requirement. Unit shown has sections of housing and inlet boxes removed.



Clarage Unit Heaters. Unitherms (shown) are equipped with centrifugal multiblade fans. Clarcos, for smaller space heating assignments, have propeller fans.

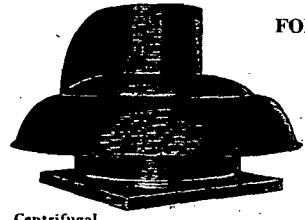
DeBOTHEZAT FANS DIVISION

American Machine and Metals, Inc. Main Office and Factory—East Moline, Illinois

Sales Engineering Offices In All Principal Cities

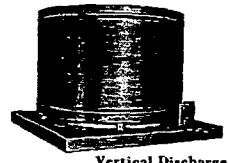
Consult Your Telephone Directory Under "Fans" or "Ventilating Equipment"

FOR CONTROLLED VENTILATION



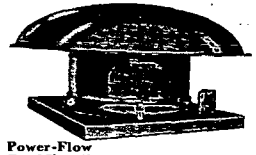
Centrifugal Roof Ventilator

Centrifugal Roof Ventilator—low-contoured, distinctively profiled—operates at very low noise levels. It is highly efficient against static pressures. Its large unit exhaust capacity is accomplished at low fan tip speeds by a backwardly curved non-overloading fan wheel. Capacities up to 32,980 cfm. *Write for Bulletin.*



Vertical Discharge Roof Ventilator

Vertical Discharge Roof Ventilators are designed for foundries, machine shops and other buildings where much oil or dust is in suspension in the air. This unit is economical in first cost and is convenient to install and maintain. Exhaust airstream disperses dust particles into atmosphere. Capacities up to 48,925 cfm. *Write for Bulletin.*

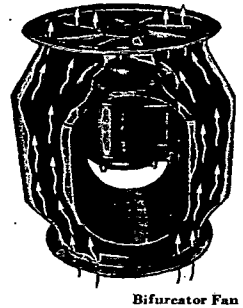


Power-Flow Roof Ventilator

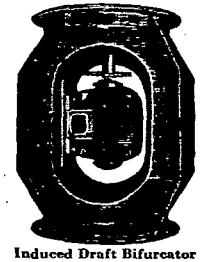
Power-Flow Roof Ventilators provide controlled ventilation at all times, regardless of weather. This unit is a motor-driven fan in weather-proof housing, designed to provide positive controlled ventilation. Works efficiently with or without a duct system. Capacities up to 43,070 cfm. All three DeBothezat roof ventilators are low in silhouette and distinctly profiled . . . blend well with modern architecture. *Write for Bulletin.*

FOR FUME REMOVAL

Bifurcator® Fans are excellent for exhausting air that is abnormally hot, corrosive, flammable or explosive. Motor is mounted in separate chamber and destructive fumes are by-passed (bifurcated) around it. Installs as a part of the duct horizontally or vertically. Fan wheels 12 in. through 48 in. in diameter, with capacities up to 54,200 cfm. *Write for Bulletin.*



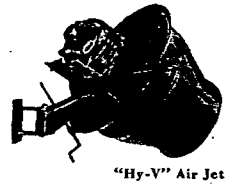
Bifurcator Fan



Induced Draft Bifurcator

Induced Draft Bifurcators provide controlled draft for boilers and furnaces, minimizing the need for large costly stacks. Built with fan wheels 12 in. through 36 in. diameter for high pressure boilers delivering up to 60,000 pounds of steam per hour—low pressure boilers delivering up to 190,000 EDR. *Write for Bulletin.*

"Hy-V"® Air Jet is used for air quenching and solving production cooling problems. Effective for reducing fatigue of workmen exposed to radiant heat from ovens, furnaces, forges or molten metal. Two styles—for stationary mounting or mounted on 2-wheel stand for easy mobility. Available with fan wheels 18 in. through 30 in. in diameter. *Write for Bulletin.* Additional bulletins are also offered on Axial-Flow Vent Sets and Panel Vent Sets.



"Hy-V" Air Jet

BONDED RATINGS Published DeBothezat ratings are backed by a $250,000 Performance Bond on deposit with New York Trust Co. Ratings are the output of the completely assembled unit, obtained under wind tunnel tests as per Standard Test Code for the particular product classification and conform to U. S. Department of Commerce Standard 178-51.

Garden City Fan Company

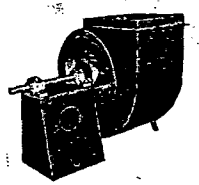
801 North 8th Street

Niles, Michigan

Representatives in principal industrial cities

Manufacturers of Design "B" Multiblade—Non-Overloading—Hi-Static and HSH Cycloidal—Radial Blade Industrial and C. I. Small Exhaust Fans. Wheels designed in forward curved, backward curved, and radial blade types for various commercial and industrial applications. Suitable for high temperature air circulation to 1850° F, air conditioning and ventilating, dust and fume removal, heavy materials conveying, and special applications requiring "Custom-built" features. High temperature fans are equipped with a special "air-cooled" shaft which eliminates bearing failures caused by excessive temperatures.

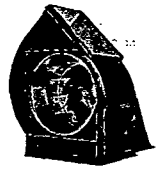
Design "B" Multiblade Fans—Compact, low speed air and gas handling fans with forward curved blades. Made for temperatures to 1650 F in uninsulated and insulated double housing construction. Well suited for systems with space limitations or low noise level requirements.



Design "B" Multiblade Fans

Non-Overloading Fans—High speed air and gas handling fans with backward curved blades. Non-overloading power characteristics. Made for temperatures to 725 F. Well suited for systems with fluctuating system resistance.

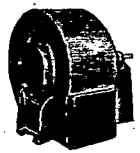
Radial Blade Industrial Fans—Heavy duty air, gas, and material handling fans. Made for temperatures to 1850 F in uninsulated and insulated double housing construction. Used for many material conveying applications, for all high temperature applications, and for exhausting against high pressures.



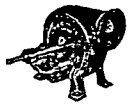
Non-Overloading Fans

C. I. Small Exhaust Fans—Durable air, gas, and material handling fans with heavy cast housing. Made for temperatures to 1650 F. Particularly suitable for applications requiring small volumes at moderate static pressures. Further information and literature available upon request.

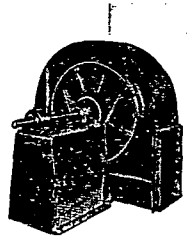
Hi-Static and HSH Cycloidal Fans—Rugged material handling and conveying fans. Two classes of construction designed to handle a wide range of materials. Specially designed "non-clogging" Cycloidal wheel reduces power requirements while offering efficient performance at low speeds.



Hi-Static and HSH Cycloidal Fans



C. I. Small Exhaust Fans

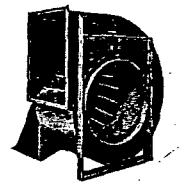


Radial Blade Industrial Fans

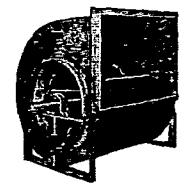
General Blower Company

8602 Ferris Ave. MORTON GROVE, ILLINOIS

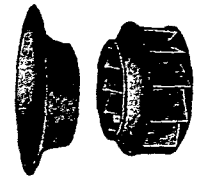
Sales Offices in Principal Cities



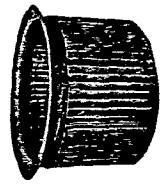
General Backward Blade Centrifugal Fans for heating, ventilating and air conditioning are non-overloading, quiet and efficient. Selection ranges through nineteen fan wheel diameters from 12¼ inch to 73 inch. **Bulletin BB-105** bears the *AMCA* seal of certification.



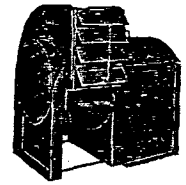
General Forward Curved Centrifugal Fans are offered in a complete range of sizes for low speed, quiet operation. **Bulletin FC-101R** bears the *AMCA* seal of certification.



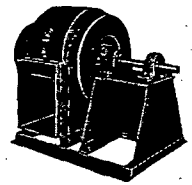
General Backward Blade wheel and inlet cone are assembled of spun parts for smooth, quiet air movement. Wheels can be manufactured of alloy materials, aluminum or hot-dipped galvanized. Ask for **Bulletin BB-105.**



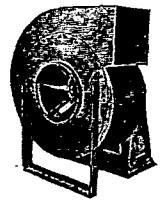
General Forward Curved fan wheel and inlet cone are assembled of spun parts for smooth quiet air movement. Wheel and cone can be manufactured of alloy materials, aluminum or hot-dipped galvanized. Ask for **Bulletin FC-101R.**



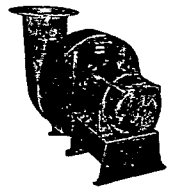
General Util-A-Vents with BB wheels are recommended for low cost air handling requirements. Belted and direct drives from 12 inch to 36 inch wheel diameters. **Bulletin UVS-104** bears the *AMCA* seal of certification.



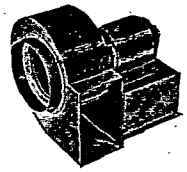
"DURA-TEMP" Heat Fans—Forward Curved and Radial Bladed Fans from 350°F to 1650°F. Heat resistant alloy and 8-in. insulated construction. Backward Blade Non-overloading Heat Fans to 350°F. **Bulletins HF-100 and HF-100A.**



General Industrial Fans for conveying dust and air borne materials. Four types of fan wheels available for air, fumes and materials handling. Modified construction for high temperature and corrosive conditions. **Bulletin IF-100.**



Turbo Blowers in two basic types: Type S single stage, and type M multi-stage. Both types used for gas and oil burners or gas boosters. 4 oz pressure—90 cfm, to 48 oz pressure—6000 cfm. **Bulletin TB-109.**

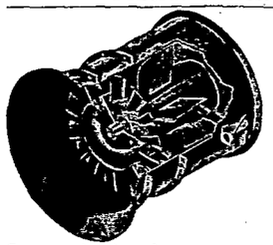


General Util-A-Vent Jr. with FC wheels are recommended for low cost air handling requirements. Belted and direct drives from 5 inch to 12 inch wheel diameters. **Bulletin FP-101.**

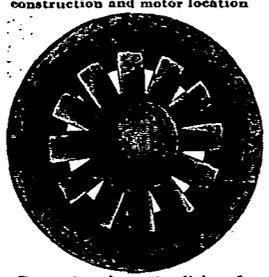
ALL GENERAL FANS TESTED ACCORDING TO AMCA STANDARD TEST CODE

JOY MANUFACTURING CO.

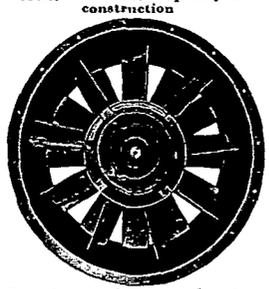
General Offices: Henry W. Oliver Building, Pittsburg 22, Pa.



Cut out drawing shows compact construction and motor location



Front view shows simplicity of construction



Rear view shows vanes and motor



Joy Axivane® FANS SERIES 1000

Joy AXIVANE® Fans, Series 1000, are available in 136 sizes to a volume capacity of 200,000 cmf and pressures up to 11 inches. Diameters range from 18 inches to 84 inches. Outstanding characteristics of Joy AXIVANE design are: straight air flow, blades adjustable on the job for volume or pressure change and simplified installation.

COMPACT

Joy AXIVANE Fans are built around the motor which permits the fan housing to become an integral part of the duct system. This inline installation frequently produces up to 70 percent more useable space than is available with conventional belt-driven centrifugal fans. Fan rooms can be eliminated in many instances.

EFFICIENT

Joy AXIVANE Fans utilize an aerodynamically efficient moving blade and stationary vane design that partially recovers the energy output of the rotor. Air is rotated by the blades, straightened by the vanes and passes through the duct in an axial flow with little pressure loss or turbulence. The natural result is increased efficiency.

EASY TO INSTALL

Because **Joy AXIVANE** Series 1000 Fans develop greater pressures and volumes per pound of unit, lighter units than equal centrifugal fans can be used. In addition, the induct installation eliminates complex duct offsets and elbows, drives, guards and foundations.

Joy AXIVANE Fans are shipped ready to install, and, in most cases, can be placed in service quickly and easily by relatively inexperienced or unskilled labor.

ADJUSTABLE BLADES

Joy AXIVANE Industrial Fans have the extra performance flexibility of adjustable blades. Adjustable blades are standard equipment on all Series 1000 fans. The factory blade setting can be quickly changed to provide either a wide pressure range for any particular volume or a change in a volume simply by loosening a lock nut with a wrench, setting the blades uniformly with the indicator, and retightening the lock nut. A permanent stop prevents setting blades in a position likely to overload the motor. Minimum blade settings are limited by the fan housing.

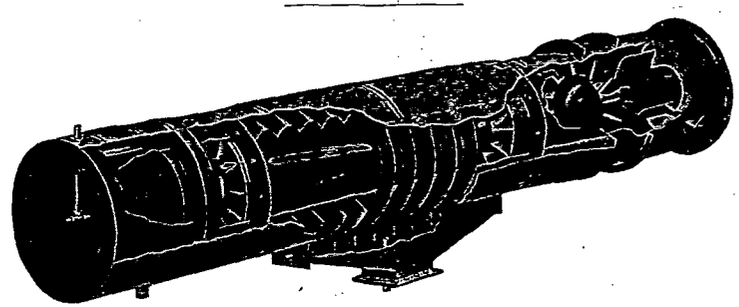
Blades are adjustable on the job by loosening each lock nut. Settings are permanently stamped on hub. This feature permits change of air pressure or volume.

Added advantage of the **Joy AXIVANE** design is that fan noise usually occurs at frequencies which are responsive to acoustical correction.

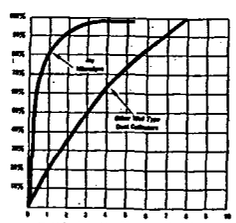
MATCHED ACCESSORIES

Inlet bells, screens, and fan supports are accessories designed to fit all AXIVANE fan housings. In ordering, it is only necessary to state the fan model number, with or without the accessories, as desired. If required with accessories, these will be furnished to fit the fan model ordered.

NOTE: An incorrect inlet bell will reduce fan efficiency by increasing intake turbulence which, in turn, increases noise level. When a fan takes its air directly from the weather, plenum, fan room, or duct system larger in circumference than the fan housing, a bell should be used.



The **Joy Microdyne Dust Collector** is a compact, wet inertial unit, 1/10 to 1/20 the size of any other collector. It collects over 99 percent of all dust particles 5 microns and larger, 92 percent of 2-micron dust, and substantial amounts of smaller dust. Cylindrically-shaped, it installs directly into the duct system—often at the point of use. It is available in ratings from 2500 to 64,000 cfm, constructed of mild steel, or Type 304 or 316 stainless steel for various corrosive and erosive conditions.



LITERATURE AVAILABLE

Industrial Fans—*Bulletin J-611*
Aircraft—Electronic Fans—*Bulletin J-614*
Joy Microdyne Dust Collector—*Bulletin J-616*
Write to Joy Manufacturing Company, Oliver Building, Pittsburgh 22, Pa. or offices listed below and specify Bulletin Number:

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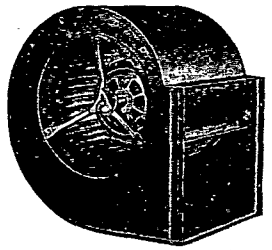
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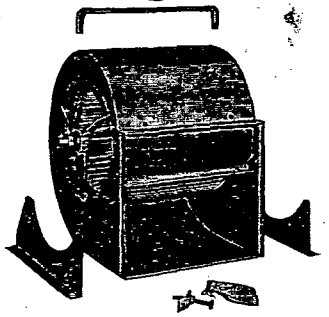
THE LAU BLOWER COMPANY

2027 Home Avenue, Dept. J, Dayton 7, Ohio

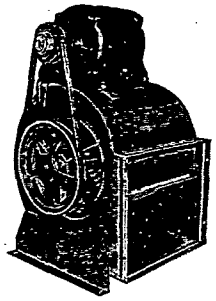
Other plants in Kitchener, Ontario, Canada and Irwindale, California



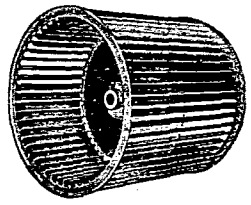
LAU all new *ELECTRO-WHEEL BLOWER: Two unobstructed air inlets on wheel sides plus compact motor design gives more air delivery with less power load. Stationary shaft with rubber mountings reduces vibration to a minimum. Rugged tripod mountings give extreme shipping strength without impeding air flow. Air deliveries from 500 to 1500 cfm, static pressure from 0.5 to 1.25 in. (water gage), ¼ and ⅓ hp motor, 9 in. and 10 in. wheels. Available in 115 and 230 volt a-c. Housing supports optional. Pre-punched holes in housing unit make any discharge angle possible.



LAU Econo-Pak Blowers cover most applications for residential air-moving equipment; effect savings up to 10 percent by eliminating special blowers for each application; reduce blower inventories by simplified order procedures, quick handling of model changes, lower handling costs, up to 26½ percent reduction in warehouse space. **Four Basic Sizes** each in 2 widths, with all features of Standard Series A Blowers. **Package Includes** blower wheel, housing with cut-off, housing supports, motor mount (not attached), all hardware; wheel, housing, bearings and shaft assembled. **Exclusive Lau-Pak Bearings** without oil cups require no lubrication. **Adjustable Discharge,** any one of 5 positions.



LAU Standard Blower Assemblies (Series A), heavy-gage steel housing assemblies, entire unit die-formed. Capacity range 350-22,000 cfm. **Motor Mounting** adjustable for any motor location. **Housing Base** heavy gage steel, rigid. **Discharge Outlet** improved design, construction. **Custom-built Belt** special composition for blower operation. **Shafting** 2-stage centerless ground, burnished. **Bearing Bracket** 3-point suspension; same bracket used for all discharge angles. **Lausteel Pulleys** employed; V.S. motor pulley allows speed variations up to 30 percent. **Center Suspended Wheel. Embossing of Scroll Sides** reinforces venturi inlet and discharge. Bearings highest grade self-aligning porous bronze sleeve type. **Plastic Thrust Washer** absorbs shocks.



LAU Preslok* Wheels, Center Suspended Riveted Wheels, S.I. and D.I. Weld Wheels, a complete wheel line. **Preslok Wheels,** sizes 9 in. through 18 in. center disc construction; baked enamel, Ruspruf, or hot-dip galvanized finish; standard steel or Power Lock hubs. **Riveted Center Suspended Wheels,** blades double-riveted to end rings, arc welded to center disc, diameters 22 in.-30 in.; standard steel or Power Lock hubs. **S.I. and D.I. Weld Wheels,** individual blades of correct length, width, pitch; baked enamel, Ruspruf, or plated; diameters 4½ in.-9 in.; widths 2¼ in.-9 in.

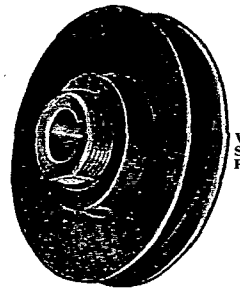
* T. M. Reg. U. S. Pat. Off.

THE LAU BLOWER COMPANY

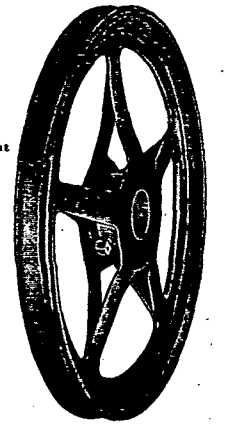
2027 Home Avenue, Dept. J, Dayton 7, Ohio

Other plants in Kitchener, Ontario, Canada and Irwindale, California

Lausteel Variable and Constant Speed Pulleys are in balance and true-running. **Lausteel Variable Speed Pulleys** available in three standard bores: ½, ⅝, and ¾ in. Diameter 3¼ in. OD. Allow speed variations up to 30 percent for many drive combinations. **Lausteel Constant Speed Pulleys** available in diameters of 6, 7, 8, 9 and 10 in., inclusive. Suitable for both "A" and "O" section belts. Lau also offers cast iron pulleys in diameters of 10 in. to 14 in.



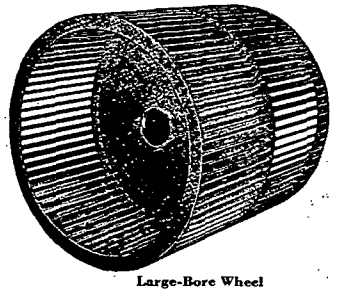
Variable Speed Pulley



Constant Speed Pulley

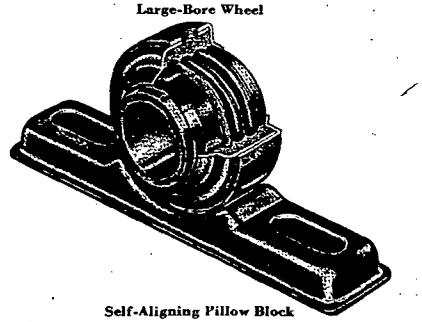
Write for LAU Blower Catalog LSO-463. Mention all Blower items of interest.

LAU Large-Bore Wheels, made to traditional LAU rigid performance standards, are now available. Bores for 3 in., 4 in. and 4½ in. shafting, with other bore diameters up to 5 in., are offered on Spun wheel sizes 9 in. through 18 in. and on Riveted wheel sizes 22 in. through 30 in. Variety of finishes (contact factory for details). NO set screws—no need for flats or keyways. New design eliminates excess weight of hub.



Large-Bore Wheel

LAU Self-Aligning Pillow Blocks are low-cost, light-weight, oil-tight steel housings with porous bronze bearings. Hold much more oil than cast iron types. Durex bushing feeds oil to shaft by capillary action. Spring keeps spherical surface of bearing and ring in close contact. Long bolt slots permit interchanging with other makes of pillow blocks. Bore sizes available are ⅝, ¾, 15/16, 1, and 1-3/16 in.



Self-Aligning Pillow Block

LAU Blower Catalog LSO-463 contains full information on Pillow Blocks and Pulleys.

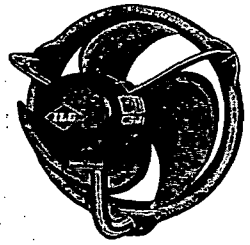
ILG ELECTRIC VENTILATING COMPANY

2880 North Pulaski Road, Chicago 41, Illinois

Offices in 57 Principal Cities • Member of *Air Moving and Conditioning Association, Inc. (AMCA)*

ILG PROPELLER FANS, CENTRIFUGAL FANS, ROOF VENTILATORS

Tested and rated by *AMCA, PFMA* and ASHVE code. Capacity guaranteed



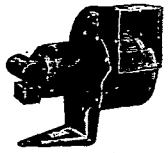
TYPE "LQ" DIRECT-CONNECTED, SELF-COOLED MOTOR PROPELLER FANS—Ilg's self-cooled motor features ventilated enclosure which keeps motor clean and cool. Permanently lubricated bearings and direct drive eliminate periodic maintenance attention. Non-ferrous Type "Q," dynamically balanced fan wheel. Exceptionally quiet operation. Wide range of sizes. Send for *Bulletin S-317.*



TYPE "XQ" Explosion-Proof Fan Motors carry UL certification of suitability for all Class 1, Group D hazardous applications. Send for *Bulletin S-317.*



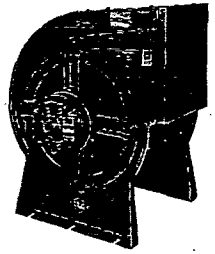
TYPE "TAQ" Tubeaxial Duct Fans for pressures up to 1.5 in. Easily installed duct section for vertical or horizontal use. Nine sizes, 2 types. Type TAQ is equipped with 3-bladed propeller; Type TA features the new 5-blade design. Capacity, range at free air 1040 to 21,300 cfm. Send for *Bulletin S-317.*



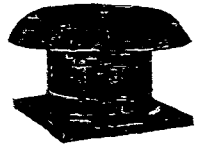
VOLUME BLOWER TYPE "B"—Small volume, low pressure, quiet running. Direct-connected motor. Universal discharge positions. 14 capacity models. Send for *Bulletin 257-28.*



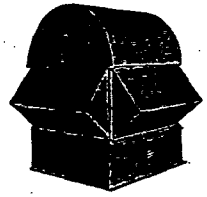
VOLUME BLOWER TYPE "P"—For exhausting dust, fumes, vapors. Cast iron housing and aluminum wheel. Direct-connected motor. 9 capacity models. Send for *Bulletin 258-26.*



NEW AIRFOIL TYPE BC DIRECT-CONNECTED CENTRIFUGAL FANS Non-overloading type. Airfoil blade design with high efficiency. Suitable for *AMCA Class I, II and III operation up* to 9-in. static pressure. Direct-drive design with motor partially recessed. 12 *sizes.* Single- and double-width models. Also belt-driven models. Send for *Catalog 257.*



ILG L-LSQ (*Exhaust*) and L-SRQ (*Supply*) **POWER ROOF VENTILATORS**—Direct-connected propeller fan type featuring low silhouette, high capacity. 11 sizes up to 21,100 cfm at free air and 13,800 at ⅛ in. static pressure. Send for *Bulletin 2302.*



ILG PRV POWER ROOF VENTILATORS—Centrifugal fan type. Direct-drive, self-cooled motor. Non-overloading, backward-curved wheel. 10 sizes. Send for *Bulletin 1907.*



ILG L-CRF CENTRIFUGAL POWER ROOF VENTILATORS—Suitable for all duct systems. Large capacity and pressure range—from 155 cfm up. Low silhouette. Sizes 67 to 182 are all aluminum. Sizes 200 and larger complete housings are constructed of spun steel. Backward-curved airfoil fan blades of patented design for quiet, high-capacity performance. Direct-connected, self-cooled motor. Send for *Bulletin 2701.*

THE NEW YORK BLOWER COMPANY

Sales Offices: 3171 South Shields Avenue, Chicago 16, Illinois

MAKE-UP AIR UNIT

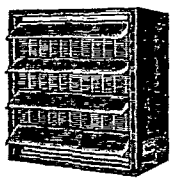


M-15 Unit 15,000 CFM

A unit that delivers warmed, filtered, outside air to industrial spaces *to replace* exhausted air and balance minus pressure. Corrects drafty conditions and uncontrolled infiltration. In 4 sizes from 5000 to 20,000 CFM. *Bulletin 568.*

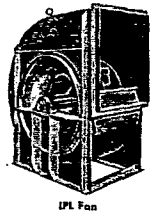
UNIT HEATERS

Ceiling suspension type with heavy duty, welded steel, fin-and-tube heating element. Suitable for continuous heating service on steam pressures up to 150 pounds or more. 10 basic sizes, capacities from 31 to 300 Mbh. *Bulletin 552.*



Unit Heater

STANDARD DUTY IPL FAN

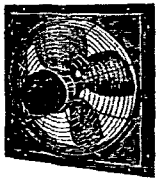


IPL Fan

A complete line of NAFM arrangements in both *forward* and backward curved wheels. Class I, II, and II construction. Capacities from 1,000 to *175,000* CFM. Quiet operating with non-overloading horsepower characteristics for all heating, ventilating and air conditioning applications. *Bulletins 561 and 572.*

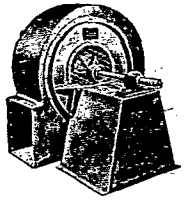
PROPELLER FANS

Propeller type ventilating fans deliver large volumes of air at low resistance and low current consumption. All wheels are machine balanced for smooth, vibrationless operation. Made in two types and ten basic sizes to cover a wide range of industrial applications. Square steel panels simplify installation. Wheel diameters from 10 inch to 48 inch. Direct or belted drive. Capacities from 610 to 24,200 CFM. *Bulletin 564.*



Propeller Fan

TYPE GI INDUSTRIAL AND HEAT FANS

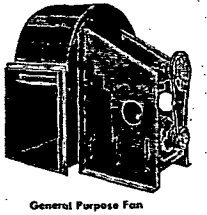


GI Heat Fan

For dust and gas *removal,* conveying of materials and the handling of hot gases in industrial processes. Housings, drives, both pulley and direct, and discharge arrangements to meet any requirement. With wheel diameters from 10 inch to 66 inch. Capacities ranging from 450 to 60,500 CFM. *Bulletin 585.*

GENERAL PURPOSE FANS

Portable, self-contained units for Class 1 industrial and ventilating applications. Recommended for their ease of installation, low maintenance and space saving features. Made in four types and twelve basic sizes to meet a wide range of industrial application. Both direct *connected* and belt driven. Capacities from 160 to 19,070 CFM. *Bulletin 563.*



General Purpose Fan

ROOF VENTILATORS



Bell-Ex (above) exhausts from hoods and duct work; VP-Ex (right) for lower resistances.



These "Twins" are completely weatherproof and corrosion-resistant, and of modern design. *Bulletin 582.*

THE NEW YORK BLOWER COMPANY

nyb

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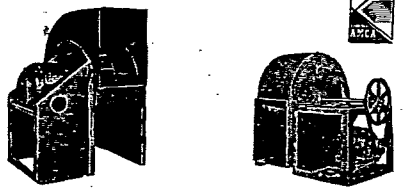
Member of
AIR MOVING and CONDITIONING ASSOCIATION

THE *Peerless-Electric*® COMPANY

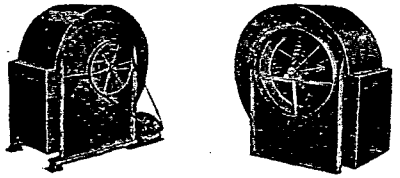
1402 W. Market Street · Warren Ohio

Representatives in Principal Cities

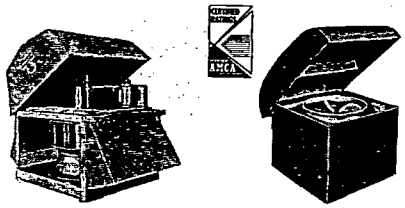
Industrial and Commercial Ventilating Equipment



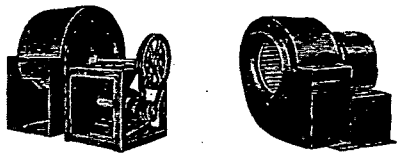
BACKWARD CURVE CLASS I BLOWERS—In sizes from 12-¼ in. to 36 in.; self-limiting hp characteristics; *AMCA* standards and ratings; dynamically balanced wheel; heavy hub; nothing in air stream; quiet entry; job-matched motor; convertible discharge; belt drive or direct drive.



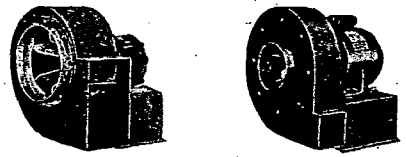
ARRANGEMENT 2 or 3, CLASS I or II—9 in. through 73-in. wheels; single width—single inlet or double width—double inlet; forward curve or backward curve type; all types of motors available; vibration bases, drives, inlet vane control and other optional features available; ball or sleeve bearing construction.



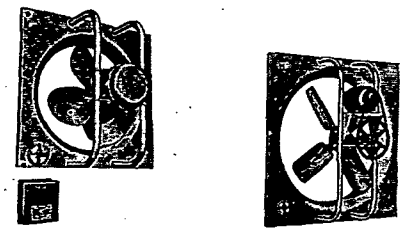
POWER ROOF VENTILATORS—CENTRIFUGAL OR PROPELLER TYPE—Direct or belt drive; 12 in. to 48 in.; 600 to 25,000 cfm; complete with disconnect switch; single phase or three phase motors; automatic shutters; inside insulation; 16 gage housing; baked enamel finish.



UTILITY BLOWERS—Belt or direct drive forward curve wheels with clockwise or counterclockwise rotation; ⅙ through 7-½ hp; motors in all current characteristics; convertible discharge; arc-welded housing; sleeve or ball-bearing construction; weatherproof drive covers optional.



RADIAL BLADE BLOWERS—⅛ to 5 hp; slow and high-speed units; self-cleaning wheel; ball-bearing motors; discharge as specified; base and housing of heavy-gage steel.



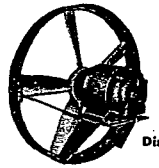
INDUSTRIAL EXHAUST FANS—Direct or belt drive; vertical or horizontal mounting; totally enclosed motors; adjustable motor base; continuous duty; ball-bearing construction; cfm rating from 470 to 29,350; shutters, inlet guards and other optional features available.

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ROBBINS & MYERS, INC. • 1359 STONE BLVD. • SPRINGFIELD, OHIO

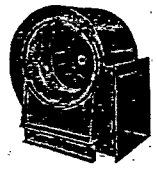
Industrial and Commercial Ventilating Equipment

Certified Ratings



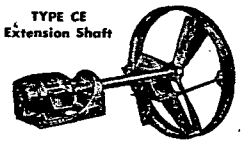
TYPE CD
Direct Connected

Type CD direct connected fans are designed for applications where the motor may operate within the airstream. They may be installed in any position in walls, roof ventilators, hoods or ducts. They are suitable for operation over a wide pressure range from free air to relatively high resistance. Available in sizes from 7¼ to 60 inches. Capacities range from 300 to 103,200 cfm. *Request Bulletin 690.*



TYPE IC
TYPE GU
Centrifugal

Designed for general ventilation, high temperatures, high pressures, fumes, gases and materials handling. Available with forward curved, backward inclined or radial blade wheels. All bearings are isolated from the fan housing. Available with clockwise and counterclockwise rotation in all standard positions. Sizes: 7½ to 33 inch wheel diameters. Capacities: 270 to 19,070 cfm. *Request Bulletins 750 and 760.*



TYPE CE
Extension Shaft

For use in hoods, ovens, spray booths or wherever the motor must be kept out of the airstream. Designed for use at relatively high static pressures. Oversize bearing assembly has sealed, prelubricated ball bearings which may be relubricated. Corrosion resistant coatings available. Nine shaft lengths from 16 to 66 inches. Sizes: 12 to 60 inches. Capacities: 1020 to 79,000 cfm. *Request Bulletin 640.*



TYPE QD
Quiet Operating

Designed for use in offices, schools, stores or wherever quiet operation is needed. Special cast aluminum-magnesium alloy airfoil propeller eliminates the sounding board effect of thin sheet metal blades. Steel venturi entrance ring prevents air recirculation at propeller tips, thus further reducing noise and increasing efficiency. Sizes: 12 to 36 inches. Capacities: 1235 to 12,000 cfm. *Request Bulletin 695.*



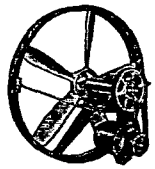
PROPELLERS

Propellair fans are equipped with airfoil propellers of a cast aluminum-magnesium alloy that provides strength, dimensional stability and balance, and greater corrosion resistance. Airfoil design assures maximum efficiency and horsepower savings impossible to attain with overloading sheet metal blades. Two-blade designs are for use at, or near, free air. Multi-blade types are for use against resistance pressure.



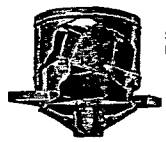
TYPE BT
Tubeaxial

For handling damaging or dangerous exhaust requiring isolation of the motor from the airstream. Integrally rolled mounting flanges eliminate fume or condensate leakage. Choice of totally-enclosed, ventilated or corrosion resistant bearing assembly. Factory-applied coatings are available for severe corrosive applications. Sizes: 16 to 60 inches. Capacities: 2755 to 85,000 cfm. *Request Bulletin 620.*



TYPE CB
Belt-Driven

Designed for large volume air delivery at relatively low noise levels and moderate pressures. For use in power plants, stores, auditoriums and factories, Type CB fans may be wall-mounted or used in the Sky-Blast power roof ventilator described below. Heavy duty bearing assembly is equipped with extra-wide prelubricated ball bearings. Sizes: 42 to 72 inches. Capacities: 14,200 to 58,100 cfm. *Request Bulletin 740.*



TYPE SB
SKY-BLAST® Power Roof Ventilator

A low-silhouette roof ventilator that blasts heat, moisture, dust and fumes high into the air away from roofs and windows. Free-swinging butterfly dampers open instantly at minimum air flow, close weathertight when fan is off. All-steel construction, hot-dip galvanized after fabrication. Rain shed and condensate drain assure weatherproof service. Sizes: 16 to 60 inches. Capacities to 78,800 cfm. *Request Bulletin 680.*



TYPE BV
Vaneaxial

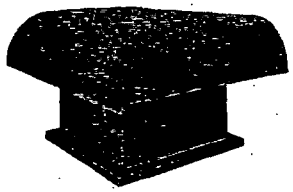
A highly efficient pressure fan that combines airfoil propeller and guide vanes. Fan bearings and motor are completely isolated from the airstream, thus making these fans suitable for handling dangerous or damaging exhaust. Rolled steel drum and belt tube are welded airtight for complete protection. Also available in direct drive. Sizes: 16 to 30 inches. Capacities: 1260 to 20,000 cfm. *Request Bulletin 795.*

Reed Unit-Fans, Inc.

501 N. St. Patrick St., New Orleans 19, La.

Manufacturers of Axial Flow Air Moving Equipment for Industrial, Commercial and Residential Use

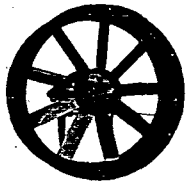
ROOF VENTILATORS



COMMERCIAL RATING: Blade Diam—12 in. to 36 in.—sheet metal blades. Air deliveries to 12,000 cfm, static pressure to ⅛ in.

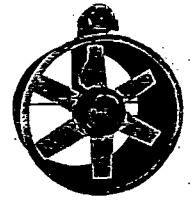
INDUSTRIAL RATING: Blade Diam—12 in. to 60 in.—cast aluminum blades. Air deliveries to 75,000 cfm, static pressures to 4½ in. wg.

Available direct and belt driven, heavy gage used throughout, fan section insulated from housing, hood easily raised from either end.



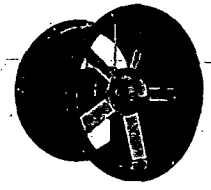
EXHAUST FANS

Blade Diam—12 in. to 60 in. Air Deliveries to 100,000 cfm. Static Pressures to 3 in. wg.
Sturdily constructed for heavy duty performance.



BELT DRIVE TUBEAXIAL FANS

Blade Diam—12 in. to 60 in. Air Deliveries to 75,000 cfm. Static Pressures to 4½ in. wg.
Cast aluminum alloy air foil blades, all welded construction, special finishes available.



DIRECT DRIVE TUBEAXIAL FANS

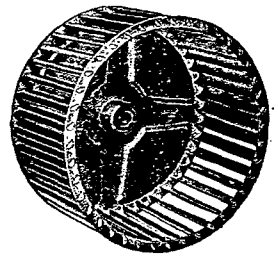
Blade Diam—12 in. to 60 in. Air Deliveries to 75,000 cfm. Static Pressures to 6 in. wg.
Cast aluminum alloy air foil blades, all welded construction, special finishes available.

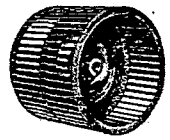
Fan Housings are of heavy gage steel, all welded construction. Impellers are of cast aluminum, Air Foil section of an unusual design which permits variation in both number of paddles and angle of attack. This unique feature makes it possible to tailor the fan to any exact air moving requirement within the ranges indicated. It also enables the manufacture of a truly reversible fan to give equal performance in either direction of air flow.

Revcor, Inc.

251 Edwards Street Carpentersville, Illinois Phone: HAzel 6-4819

ENGINEERS AND MANUFACTURERS OF SINGLE AND DOUBLE INLET BLOWER WHEELS, PROPELLER FANS, HOUSINGS AND INLET RINGS





Revcor Blower Wheels are available in a very large selection of sizes. 20 diameters of Single and Double Inlet Wheels from 3 in. to 12⅝ in. Single inlet widths from 1 in. to 6 in. Double inlet widths from 2 in. to 12 in. Individually attached, thin gage, full width blades eliminate air obstructions at the ends. Blades are attached to end rings by Revcor's time proven, tab locked method.

BLASTAIRE DESIGN

Engineered to provide outstanding performance. Ruggedly made for all appliance applications. Available in steel and aluminum. Unique *Blastaire* production methods result in lower costs while maintaining high standards of rigidity, strength, balance and performance!

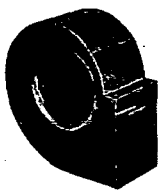
Individual Blades of correct curvature, number, depth and blade angle to deliver the greatest amount of air, efficiently and quietly.

Full Width Blades with a blade tip diameter substantially the same as the outside diameter of the wheel deliver the maximum air and pressure for the space dimensions required.

Blade tips are securely locked to the back plate and ring. Embossed back plate provides rigid support for blades and hub. Hubs can be mounted either inside or outside the wheel.

NEW DOUBLE INLET BLASTAIRE DESIGN

Features single center disk construction. Blades extend through slots in the center disk and are locked tight to the center disk by a unique patented method. Provides more air and pressure in less space.

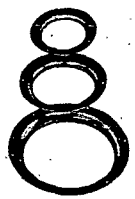


HOUSINGS

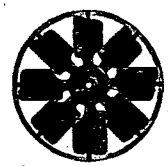
Sheet metal housings die formed to insure custom-made uniformity at mass production prices. Readily permit adaptation of larger size wheels when required by space limitations.

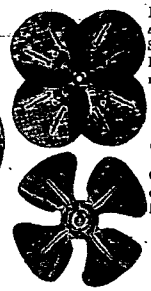
INLET RINGS

Scientific Bell Mouth Venturi design assures least resistance to air flow. Die formed for consistency. Sizes for all Revcor Blower Wheels. Also available in special sheet metal shapes.



PROPELLER FANS—SIZES RANGING FROM 6 IN. TO 48 IN.





Revcor's expanded fan line offers you a broad range of fan types for almost any air moving application. Fill your fan needs from the following types: Static Pressure Series, Price Saver (All Purpose) Series, Space Saver Series, Hurricane Series, Multi-Blade Pressure Series, Hi-Static Series, One Piece Series, Attic Fan Series. Slinger rings are available for most types.

CATALOGS AND DESCRIPTIVE SHEETS AVAILABLE

Containing technical details, diagrams, specifications, performance ratings, sizes, etc., of all Revcor Blower Wheels, Propeller Fans, Housings and Inlet Rings. Write Direct to:

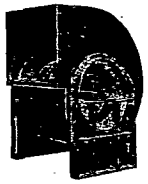
REVCOR, INC., Dept. R-1

251 Edwards Street • Carpentersville, Illinois

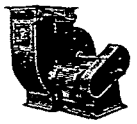
SHELDONS ENGINEERING LIMITED

GALT, ONTARIO, Montreal, Toronto, London, Ottawa, Hamilton

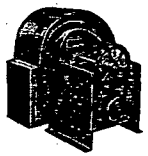
Representatives in all principal cities across Canada



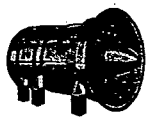
Backward Curved Fans



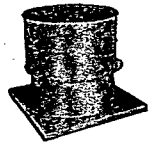
Industrial Fans



Utility Sets



Vaneaxial Fans

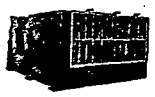


Roof Ventilators

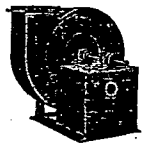
AIR HANDLING APPARATUS

No one Fan type can do all things. Sheldons since the year 1896 have designed and engineered the complete line of products to move and condition air. Sheldon products are accepted as "EQUAL" to any similar apparatus produced on the American Continent.

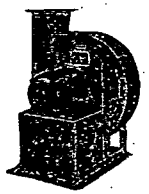
When planning your projects in Canada we invite your inquiries for specifications and complete information on SHELDON Air Handling Equipment. Consult our engineers. They are experienced and familiar with all requirements apt to be met in the Canadian field.



Multi-Zone Units



Mill Exhausters



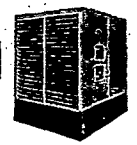
Pressure Blowers



Unit Heaters



Air Conditioners



Air Washers

Air Conditioning Units • Keith Fans • Silavent Fans • Utility Sets • Forced Draft Fans • Induced Draft Fans • Airscrew Fans • Vaneaxial Fans • Medium Blowers and Exhausters • Mill Exhausters •

Dust Separators and Collectors • Dust Filters • Centrifugal Blowers • Unit Heaters • Drying Equipment • Chemical Plant Fans • Mine Ventilating Fans •

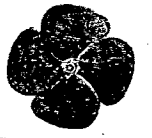
Complete engineering data and catalogs available upon request

THE TORRINGTON MANUFACTURING COMPANY

TORRINGTON, CONNECTICUT • VAN NUYS, CALIFORNIA • OAKVILLE, ONTARIO

TOTAL COVERAGE

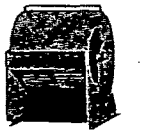
Torrington offers the most complete line of fans in the United States. Applications range from small precision fans for electronic cooling to the largest heavy-duty requirements of industrial and residential air moving—in diameters from 1½" to 52".



Torrington blower wheels are available in diameters from 1½" to 16". A full range of designs includes standard single and double inlet Airotors, in both continuous and individually-bladed construction.

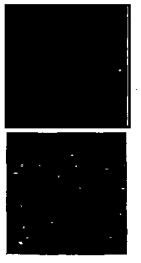


In complete blower units, Torrington offers proved-out performance ratings on three full lines, the Vari-Basic line of belt driven and direct drive units, plus the revolutionary new mixed flow Radiax blower.



The result is TOTAL COVERAGE—the best "problem insurance" you could have. For complete data:

Talk to Torrington!



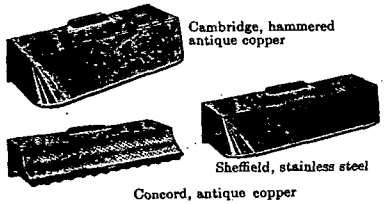
TRADE-WIND MOTORFANS, INC.

Division Of Robbins & Myers, Inc.

7755 Paramount Place, Pico Rivera, California

TRADE-WIND VENTILATORS
EXHAUST VENTILATION FOR HOME KITCHEN, LAUNDRY, BATH, DEN AND OTHER SMALL ROOMS

SPACE SAVER KITCHEN VENTILATING HOODS



Cambridge, hammered antique copper

Sheffield, stainless steel

Concord, antique copper

New high style hoods with built-in under hood ventilator. Takes up no cabinet space. Touch-Bar controls dual lights and 3-speed ventilator. Two oversize cleanable filters. Built-in back draft damper. 3¼ in. x 10 in. duct size. 30 in., 36 in., 42 in. and 48 in. hood lengths.

PRE-WIRED HOODS



Salem, antique copper
Montecito, brushed copper
Newport, stainless steel

Beautifully designed. For use with models 3501, 2501 or 1501 ventilators shown at right. Hoods are pre-wired with cut outs for ventilators at right, left or center. Touch-Bar controls dual lights and 3-speeds. 30 in., 36 in., 39 in., 42 in. and 48 in. lengths.

PATRICIAN VENTILATING HOOD



Coppertone
Satin Chrome

Complete pre-wired package with axial flow fan, light, switch and filter. 30 in., 36 in., 39 in., 42 in. and 48 in. lengths.

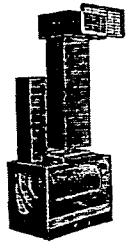
OVEN VENTILATING HOOD



Ventilator only available; also 3 hood styles—
Newport, stainless steel
Montecito, brushed copper
Salem, antique copper

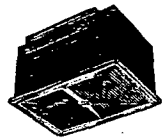
For built-in gas and electric ovens. Pre-wired with switch and automatic thermostat control. Snap-in filter; built-in back draft damper. 24 in., 27 in., 33 in. and 45 in. hood lengths. 3¼ in. x 10 in. duct size.

VENTILATORS



Model 3501 Ventilator

For cabinet installation only. Two inlets, one over stove, second at ceiling level, each with washable filters. Twin centrifugal blowers. Built-in back draft damper. Duct size 4 in. x 13 in.



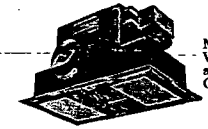
Model 2501/1501 Ventilators

For use with Trade-Wind hoods or for ceiling installation. Handsome new style grille. Built-in back draft damper. Model 2501 for rooms up to 2000 cu ft. Model 1501 for 1000 cu ft. Duct size 3¼ in. x 10 in. Drawer-type filter available



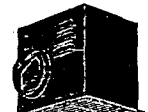
Model AF-7M Axial Flow Ventilator

Full 300 cfm. High-style grille. Installs in ceiling or cabinet. *Duct size 7 in. round.* 50 watt motor.



Model 1701 Ventilator—and Light Combination

For bathrooms, 2-75 watt lamps, 100 cfm ventilator. Built-in back draft damper. *Duct size 4 in. round.* 50 watt motor.



Model 1201

100 cfm, specially built for bathrooms. Duct size 4 in. round. 50 watt motor. Built-in back draft damper.

Viking® Air Products

5601 Walworth Ave. · Cleveland 2, Ohio

Viking Blowers Solve These Design Problems

Interchangeable Parts . . .

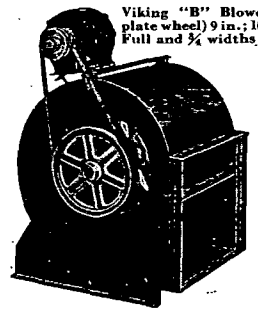
many of the important features of Viking Blower Assemblies available in these component parts: New Mark III Bearing is factory-packed but designed to permit the addition of oil; New Quadruped Bracket reduces noise possibilities and strengthens blower housing; New Center-Disc Non-Flexing "B" Wheel is engineered for strength; and Universal Lock-on Feet designed to require a *minimum* of *assembly time.*



- **Lighter loading, longer life**
- **Machined journal cuts shaft cost**
- **Interchangeable with conventional Bearings**
- **Factory-packed, but may be re-oiled**

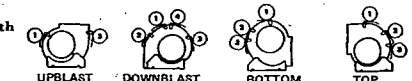
Interchangeable Blowers . . .

in performance, dimensions, belts, pulleys and air patterns. Viking's "B" Blowers meet *A.G.A.* requirements as acceptable substitutes for "A" Series Blowers without *A.G.A.* retesting and without design or specification changes by manufacturers. Also has Universal Blower quick-assembly features.



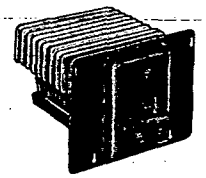
Viking "B" Blower (with center-plate wheel) 9 in.; 10 in.; 12 in.; 15 in. Full and ¾ widths.

Possible Positions with Lock-on Feet



Cost Reduction . . .

the Lock-on Feet and multiple motor positions of the Viking Universal Blower Assembly reduce inventory, eliminate dead blower stock, cut lead-time needed in ordering and reduce initial cost. Parts designed to permit quick assembly.

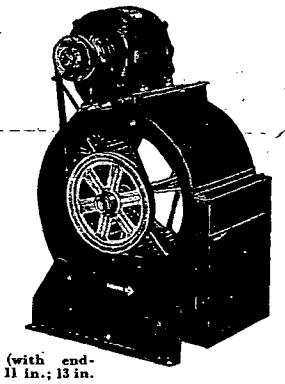


6000 Leak-Proof Humidifier . . .

New ovenware glass pan eliminates leaks, corrosion and rust common in old-fashioned humidifiers. Completely automatic unit.

Space Restrictions . . .

New Viking Direct Drive Blowers are available for application where *space is at a premium.*

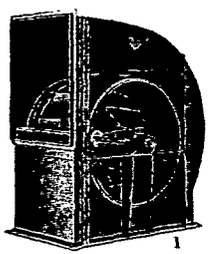


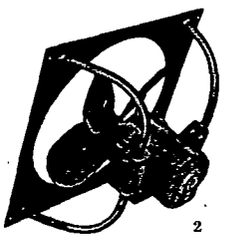
Viking "U" Blower (with end-supported wheel) 9 in.; 11 in.; 13 in. Full and ¾ widths

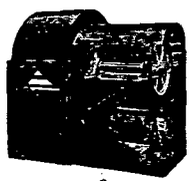
WESTERN BLOWER COMPANY

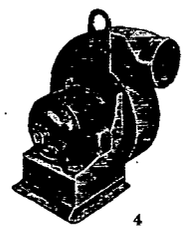
Main Office and Plant: 1800 Airport Way, Seattle 4, Washington

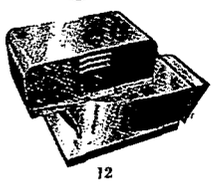
Sales Offices in the Principal Cities West of Rocky Mountains

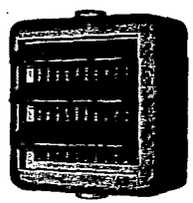

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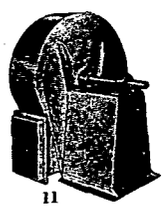

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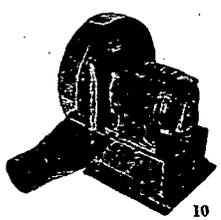

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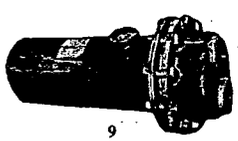

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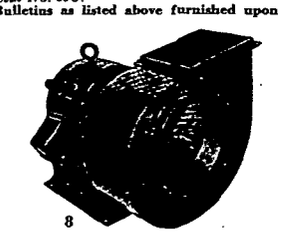

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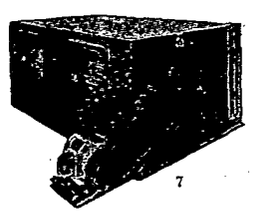

11


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10


9


8


7

1. **TURBINE MULTIBLADE FANS**—forward curved blade, Type TR, or backward curved blade, Type "S", fans for heating, ventilating, mechanical draft, etc. *Bulletins No. 30 & 31.*
2. **SPIROVANE PROPELLER FANS**—furnished either direct connected or V-belt driven for commercial or industrial ventilation. *Bulletin No. 60.*
3. **TURBINE UTILITY SETS**—V-belt driven, slow speed, quiet operating, for general utility duct ventilating systems. *Bulletin No. 31.*
   SERIES 59 UTILITY SETS—for general exhaust systems. Complete packaged units. *Bulletin No. 59B.*
4. **RB VOLUME & PRESSURE FANS**—radial blade type either direct connected or V-belt driven for ventilating and conveying applications. *Bulletin No. 59A.*
5. **WESTERN UNIT HEATERS**—vertical or horizontal, for general heating and drying applications. *Bulletin No. 55.*
6. **AIR WASHERS**—for cleaning, cooling, humidifying and dehumidifying. *Bulletin No. 50.*
7. **VOLUME HEATERS**—with one or more centrifugal fans for heating, ventilating and air conditioning. Available in vertical or horizontal cabinet units, or Multi-zone Units with zone dampers. *Bulletin No. 70.*
8. **TURBINE ELECTRIC VENTILATING UNITS**—direct connected for general ventilating systems. *Bulletins No. 30 & 31.*
9. **OLYMPIC HEAT EXCHANGERS**—Converters, Side Arm Heaters, Immersion Heaters, Oil Heaters, and Condensate Coolers.
10. **HIGH PRESSURE BLOWERS**—to furnish air at high pressure for gas and oil furnaces, flotation separators, blast drying, and cupolas.
11. **WESTERN SPECIAL & WESTERN SLOW SPEED PLANING MILL EXHAUSTERS**—single or double for general mill exhaust, and material handling. *Bulletin No. 52A.*
12. **SERIES 59 ROOF VENTILATORS**—Slow speed wheel. Forward curved blades. Motor, shaft & bearings out of air stream, all resilient mounted. Complete weather protection with automatic louvres on discharge of unit. *Bulletin No. 59C.*

Bulletins as listed above furnished upon request

Westinghouse Electric Corporation

Sturtevant Division

Heating, Ventilating, Cooling, Dehumidifying,
Electronic Air Cleaning and Mechanical Draft Equipment

Main Office—Boston 36, Mass. Offices in Principal Cities

AIR HANDLING PRODUCTS

Centrifugal Fans with Airfoil Blading—Standardized General Purpose and Heavy Duty Fans for both commercial and industrial applications. Capacities up to 700,000 cfm, all pressures up to 45 in. wg. *AMCA* pressure classes I, II, III, IV.

Axial Flow Fans—V-belt driven or direct connected: Designed to provide low cost fume and vapor exhaust and air supply with new high efficiencies. Easily installed in ductwork. Vaneaxial, Tube-axial and Spray-booth types available for wide range of applications in all industrial fields.

Ventilating Sets—Direct connected units for maximum efficiency at popular motor speeds. Manufactured in six sizes up to 3200 cfm. Compact V-Belt driven units for large capacity and quiet operation. Fourteen sizes up to 21,000 cfm.

Air Distributing Units—Air Conditioning, Sprayed Coil and Multizone; Heating and Ventilating, and Ventilating Fan Units. Factory built, unitary assemblies. Horizontal and Vertical. Coils available for Chilled Water and Refrigerant; steam and hot water. All units available with "In-Line" PRECIPITRON Electronic Air Cleaners. Capacities up to 48,000 cfm.

Cooling and Heating Coils—coils are available for direct expansion refrigerants chilled or hot water and steam. Water Coils in continuous tube or removable header type. Steam coils in standard, heavy duty, and steam-distributing types. Coils feature Air-Activating Plate Fins.

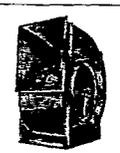
Surface Dehumidifiers—For central plant air conditioning. Sprayed coil units are available with chilled water or direct expansion refrigerant coils. Available with water spray and recirculating system, eliminators. Capacities to 250 tons of refrigeration and 62,500 cfm.

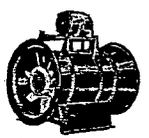
Industrial Heaters—For heating of manufacturing areas, warehouses, garages and similar industrial and commercial buildings. Also for continuous duty heating in industrial processes with heavy duty wrought iron coils. Capacities to 4,500,000 Btuh, 32,000 cfm.

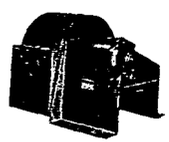
Cabinet and Propeller Fan Heaters—Available for steam, hot water. In horizontal and downblast models. Capacities range from 15,000 to 684,000 Btuh, 240 to 11,000 cfm.

Gas-Fired Unit Heaters—Propeller and blower type. Used with Natural, Mixed, Manufactured, L. P. Gases and L. P. Gas-Air mixtures. 25,000 to 250,000 Btuh. 400 to 3,600 cfm.

(J-20678)

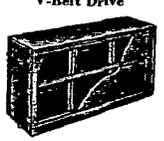

All Purpose Centrifugal Fan—Catalog 1121

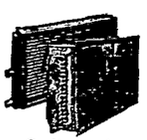
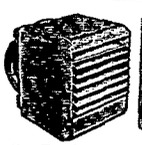
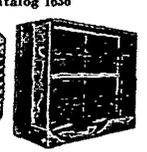

Axial Flow Fan Catalog 1110



Ventilating Sets—Catalog 1160
Direct-Connected V-Belt Drive

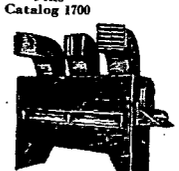

Air Distributing Unit Catalog 1600


Electronic Air Cleaner "In-Line" for Air Distributing Units—Catalog 1636




Cooling and Heating Coils, with Air Activating Plate Fins Catalog 1700

Gas Fired Unit Heater Catalog 1525

Surface Dehumidifier Catalog 1660-2

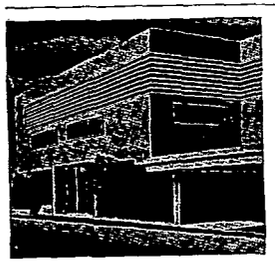

Industrial Heater Catalog 1510


Steam & Hot Water Unit Heater Catalog 1500

THE AMERICAN WARMING AND VENTILATING CO.

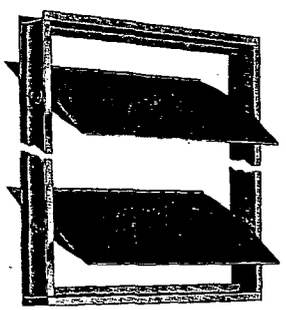
WAY 1017 Summit Street · Toledo 4, Ohio · CHerry 3-7151

"Quality Air Control Equipment"



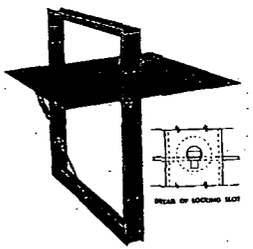
LOUVER

Architectural type louvers having an unbroken line regardless of the length of the run compliments today's contemporary design. A new principle of blade supports make the unit possible.



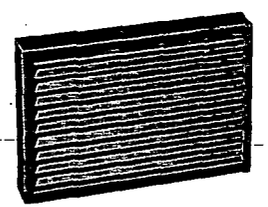
AIR CONTROL DAMPER

A new development, center pivot balanced linkage takes unnecessary torque from the operation of air control dampers. This new linkage is located out of the air stream thereby eliminating unnecessary turbulence.



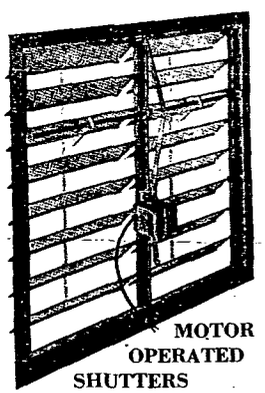
FIRE DAMPER

A new key slot locking device now insures tight and positive closing of fire dampers.



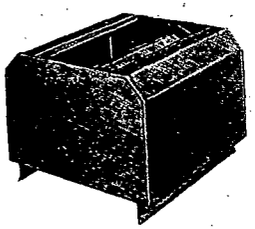
DOOR LOUVER

An entirely new engineering development makes possible door louvers of exceptional strength and blade alignment yet keeping cost to a minimum.



MOTOR OPERATED SHUTTERS

A recently perfected rotary leverage principle, for motor and manual operated, shutters makes possible free and easy opening yet when closed the shutter blades seal tightly against one another to repel weather.



LP-SERIES ROOF VENTILATOR

This new quality roof ventilator eliminates the necessity of many types, for its design encompasses universal application and functions either for air exhaust by gravity or power, air intake by power. One style unit now does every job.

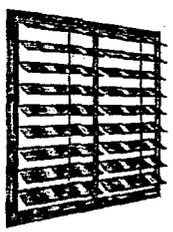
Elgo SHUTTER & MANUFACTURING COMPANY

2738 West Warren, Detroit 8, Michigan—Phone Tyler 8-0637

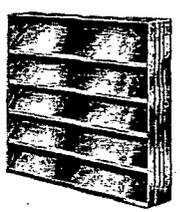
Elgo Ventilating Specialties

New! Extruded Aluminum Shutters with Completely Concealed Pivot Pins

Lightweight—Strong—Durable



AUTOMATIC WALL SHUTTERS



STATIONARY SHUTTERS



BACK-DRAFT DAMPERS

The new Elgo Extruded Aluminum Shutters give you shutters of modern design. They meet the demand of your customers who insist upon attractive, lightweight, strong and durable shutters.

The distinctive appearance of Elgo Aluminum Shutters blends with aluminum doors and windows. Your customers will also like the rust proof and non-corrosion qualities of Elgo Aluminum Shutters and, the fact that Elgo Shutters do not need paint to protect them from the elements.

The weather strip and extruded frame are one piece unit construction. All pivot pins are completely concealed. This gives you positive pivot pin action and prevents rusting, which means—no rust stains to mar the beauty of the frames. The full weather strip gives you ample protection from back-drafts, rain, snow, dust and other hazards of the elements.

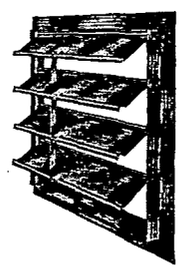
ELGO shutters give you all these advantages . . .

- Light Weight
- Full Weather Strip
- Low Freight Cost
- Easier Installation
- Concealed Pivot Pins
- Rust Proof and Corrosion Proof Aluminum
- Natural aluminum finish with fluted frames.

Types of Shutters—Steel or Aluminum Frame

Automatic Shutters
Automatic Back-Draft Dampers
Stationary Shutters
Hand-Operated Shutters
Unit Blower Shutters
Motor-Operated Shutters
Automatic Ceiling Shutters
Automatic Ceiling Dampers
Motorized Ceiling Shutters
Hand-Operated Ceiling Shutters

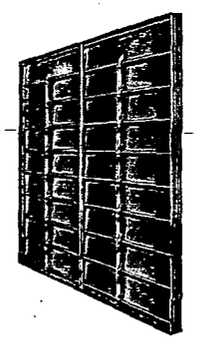
Write for complete specifications.



AUTOMATIC WALL SHUTTERS



CEILING SHUTTERS



BACK-DRAFT DAMPERS

Colt Ventilation of America, Inc.

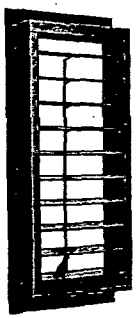
4652 Hollywood Boulevard, Los Angeles 27, California

Telephone: NOrmandy 2-1181

In England: COLT VENTILATION LTD • SURBITON • SURREY

Telephone: ELMBRIDGE 6511

Agents in: Australia, Belgian Congo, Belgium, Burma, Cyprus, India, Indonesia, Madagascar, Malaya, Mauritius, New Zealand, Pakistan, Portugal, Rhodesia and Nyasaland, South Africa and West Indies.



The Colt System of Customized Natural Ventilation

Colt Clear-Opening Ventilators for roof or wall—completely weather-proof

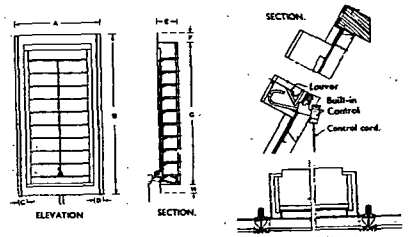
STANDARD AG/CO VENTILATOR manufactured in various sizes. Width can be varied to fit non-standard width of glazing. **COLT AG/CO/2046** comes complete with built-in control, endless control cord. Heavy gage anti-corrosive hardened aluminum, weight 23 lb. **COLT AG/CO/2446** same as 2046 except dimensions (see chart), weight 27 lb. **GIANT CLEAR-OPENING VENTILATORS (AG/CO/4060, 4460, 4080, 4480)** designed for use where large quantities of heat or smoke must be removed from a small "hot spot." Can be fitted into roof for extraction or at low level for air inlet. All ventilators can be operated individually or alternately in batteries from one control point.

TYPICAL DIMENSIONS OF COLT CLEAR-OPENING VENTILATORS

Type	A	B	C	D	E	F	G	H
AG/CO/2046	2'0"	4'6"	1"	1"	6¾"	3"	4'0"	3"
AG/CO/2446	2'4"	4'6"	3"	3"	6¾"	3"	4'0"	3"
AG/CO/4060*	4'0"	6'0"	1"	1"	8¾"	4½"	5'3¾"	4½"
AG/CO/4460*	4'4"	6'0"	3"	3"	8¾"	4½"	5'3¾"	4½"
AG/CO/4080*	4'0"	8'0"	1"	1"	8¾"	4½"	7'3¾"	4½"
AG/CO/4480*	4'4"	8'0"	3"	3"	8¾"	4½"	7'3¾"	4½"

* These Ventilators are designed on the same principle but have manual control bar only. Can be supplied with manual or power-operated gearing.

PATENT & APPLICATIONS: 650372, 19577/55, 10630/55, 114569, 16504/54. Other U.S.A., Canadian and Foreign Patents Pending.



Colt Inflow Units . . . Supply and Recirculation

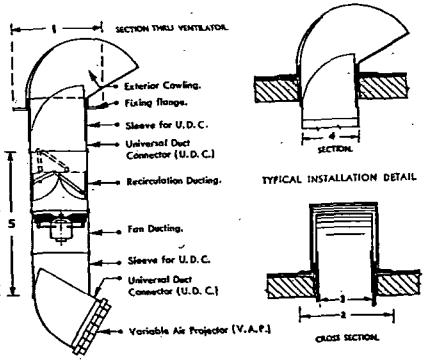
Powered ventilators for air intake. Fitted with Recirculating Dampers for recirculating heat within the premises. Can be fitted with filter unit to remove dust and dirt.

PRINCIPAL DIMENSIONS

Type	1	2	3	4	5
AG/RIU 15" Roof Inflow	2'2"	2'0"	1'4½"	1'4½"	3'4"
AG/RIU 18" Roof Inflow	2'6"	2'0"	1'8½"	1'8½"	4'0"

PATENTS: 718740, 845813, 861775, 506552, 2626556. Other U.S.A., Canadian and Foreign Patents Pending.

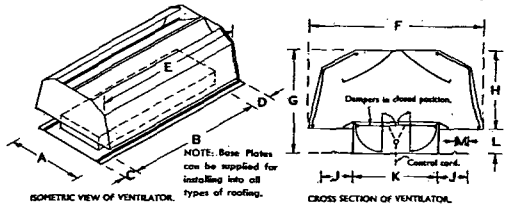
All ventilators available from stock



The Colt AG/SR Ventilators for Hot Industries

Positive, all year 'round, all-weather controllable ventilation, guaranteed regardless of wind conditions. No back draft from adverse winds provided inlet is balanced to extracts. With or without controls.

Designed on aero-dynamic principles embodying aerofoil curves which harness and control the free power of wind and roof eddies. Combined with power of convection currents within building, ventilators create a suction resulting in power of extraction equal to that of mechanical means without disadvantage of noise, wearing parts or operating or maintenance costs. Design effectively prevents rain from entering building.

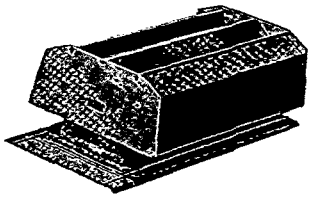


PRINCIPAL DIMENSIONS

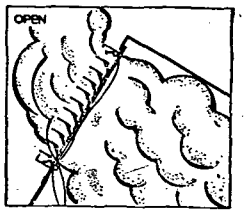
Type	A	B	C	D	E	F	G	H	J	K	L	M
AG/SRC 2046	2'0"	3'8"	4"	6"	3'11"	2'4½"	1'5½"	1'1½"	5"	1'2"	4½"	2"
AG/SRC 2060	2'0"	5'2"	4"	6"	5'5"	2'4½"	1'5½"	1'1½"	5"	1'2"	4½"	2"
AG/SRC 3080	3'0"	7'0"	6"	6"	7'2½"	4'0"	2'2"	1'9¼"	6"	2'0"	4½"	2"

PATENTS & APPLICATIONS: 567801, 610934, 727817, 166533, 19576/55, 10631/55, 114568. Other U.S.A., Canadian and Foreign Patents pending.

AIA File Numbers: 12-K, 30 D-1, 14-U



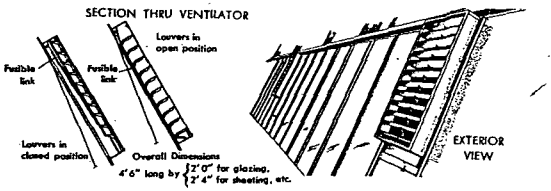
Colt Dual Purpose* Smoke and Heat Exhaust Ventilator



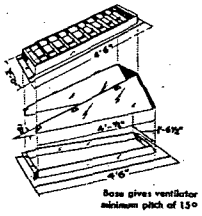
COLT AG/COF/2046 Ventilator in open position, for automatic escape of super-heated air and smoke

COLT AG/COF/2046, 4060, 4080. Each ventilator fitted with automatic release in event of fire. Immediately removes smoke and heat, conforming with requirements of International Fire Services who state that "proper fire venting is the key to the prevention of spread of fire once it has started in an industrial building." Fusible link can be designed to trigger ventilator open at any predetermined temperature.

* Also a day-to-day usable ventilator with manual controls as described under AG/CO Series on opposite page. Specifications same as AG/CO Series.



Typical arrangement of COLT AG/COF/2046 DUAL PURPOSE FIRE VENTILATORS





COLT AG/COF/2046 Dual Purpose Fire Ventilators for flat or shallow pitch roofs

Write for free Technical Manual No. AG5. Complete engineering data, sizes

Allen Cooler & Ventilator, Inc.

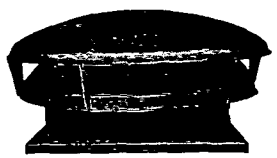
Rochester, Michigan

See *Sweet's Catalog* for further details

ALLEN ENGINEERED VENTILATION

Write to factory for catalog

Representatives in principal cities



"I-LINE" LOW SILHOUETTE ROOF FAN

Allen "I-LINE" Roof Fan is designed to meet the architect's demand for an attractive silhouette and the engineer's demand for high efficiency. This is accomplished by use of a circular spun steel hood, and venturi, with baffle so arranged in relation to the propeller as to provide maximum efficiency with minimum over-all height. Its appearance and quiet performance recommend it for new or old construction, such as schools and offices as well as industrial plants. It is easy to install (no extra base required) and it is easy to lubricate and clean. Available with *Direct Drive* (420 to 45,000 cfm) or with *Belt Drive* (4,000 to 31,000 cfm). Special units available to 70,000 cfm.



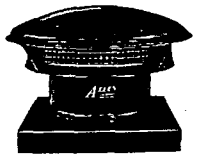
ALLEN STAXAUSTER for Conversions or New Construction

Allen Staxauster is a simple, compact, sturdy unit that can be used to convert all types of roof ventilators into powerful exhaust or air supply units. It is available in a wide range of sizes and capacities to solve almost any ventilating problem, with either *Direct Drive* or *Remote Drive*, as shown below.

REMOTE DRIVE STAXAUSTER for Corrosive Fumes



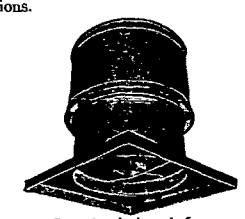
Allen Remote Drive Staxauster is specially designed for applications handling fumes, and/or high temperature air. Motor and belt drive are completely isolated from the air stream.



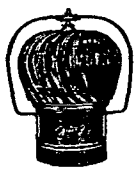
"T-TYPE" LOW VELOCITY ROOF FAN

Allen "T-TYPE" Ventilator is designed for low, one-story, flat-roof, modern buildings. It blends well with modern exteriors and its design and construction assure high efficiency, easy installation and low maintenance. Model T-9 has capacity of 420 cfm at 0 in. SP; Model T-16, a capacity of 600 cfm at 0 in. SP. Low height enables these units to be used for both roof and wall installations.

HIGH VELOCITY VERTICAL DISCHARGE FAN



Allen Type "VD" Roof Fan is designed for use where extremely high discharge velocity prevents the tendency of fumes to short circuit back through building openings. Fabricated of prime zinc-coated iron sheet, welded construction throughout. Automatic wing dampers open with air blast, close weathertight when not operating. Available with *Direct Drive*, with motor readily accessible, or *Remote Drive*, with motor in outside housing and enclosed V-belt drive. Capacities available to 90,000 cfm.



ALLEN TYPE C TURBINE VENTILATOR

Allen Type "C" Turbine Ventilator is wind driven and economically removes heated air, fumes and dust. Other Allen turbine-type ventilators include power-driven models for ventilating in calm air conditions.

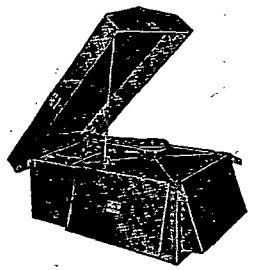
AirXpeler EXHAUSTERS

Ammerman Co., Inc.

Stillwater, Minnesota

Exhausters Powered or Gravity
Underfloor and Overhead Exhaust Systems

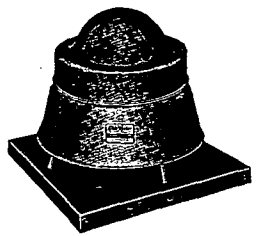
BELTED ROOF EXHAUSTER



"BCB" Lo-Boy Non-Overloading Centrifugal Wheel. Capacity table offers a selection of 16 wheel sizes—204 constant speed combinations with tip-speeds as low as 2000 fpm. Unit is exceptionally low in height. Standard construction consists of welded structural frame.

Write for special engineers bulletin

BACKWARD CURVE DIRECT CONNECTED



"BCD" Non-Overloading Roof or Wall Exhauster. Designed for optimum performance at lowest possible velocities, reduces noise level to an absolute minimum. Motor directly connected to dynamically balanced wheel eliminates costly maintenance.

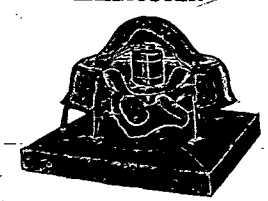
See Sweet's file for further details

ROOF RELIEF EXHAUSTERS



"RC" is regular Airxpeler design—but spun from one piece of metal. Throats may be round up to 42 in., square or rectangular up to 34 in. Constructed to accommodate any size roof opening. May be used as gravity vents. **Write for Bulletin.**

POWER ROOF EXHAUSTER



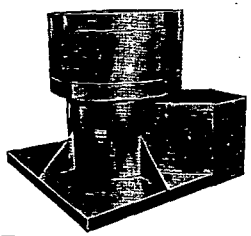
"PB" or "PFAS" Lo-Boy Wall or Roof Units. Efficient and quiet operation for low static pressures up to ½ inch. Rubber grommets are installed around all points where metal contacts metal. Powdered fresh air supply units on "PB" exhauster with reversed pressure blade to bring fresh air in under power.

DISAPPEARING EXHAUST SYSTEMS



Under the Floor and Overhead exhaust systems for all makes and models of cars and trucks. Will accommodate flexible metal hose up to six inches in diameter. Write for **Bulletin No. 650H.**

UPBLAST ROOF EXHAUSTERS



Designed for immediate heat, smoke and fume expulsion. Split disc damper that automatically opens when fan is turned on provides weather proofed units. Direct drive units equipped with totally enclosed ball bearing motors. Belt driven units furnished with motors out of line of airflow in separate weather proof compartment.

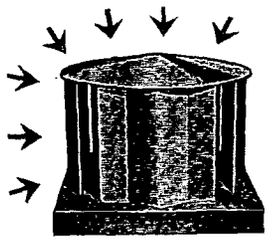
Write for Special Bulletin

THE G. C. BREIDERT CO.

P.O. BOX 1190, San Fernando, Calif.

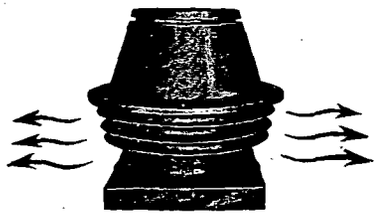
Representatives in All Principal Cities

POSITIVE VENTILATION GUARANTEED... NO MATTER WHICH WAY THE WIND BLOWS!



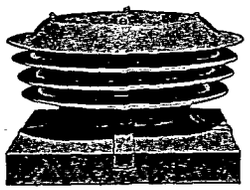
Breidert AIR-X-HAUSTER ... famous gravity-flow ventilator developed from principles of aerodynamics; proven by years of use in all types of application. Transforms winds from *straight-downward to 90 degrees upward* into positive suction—without a single moving part! Trouble-free, maintenance-free, weather-proof. Guaranteed backdraft-proof, barring internal negative pressures.

HORIZONTAL DISCHARGE ELIMINATES DOWN-DRAFTS!



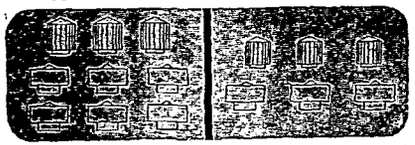
Breidert POWAIR-X-HAUSTER ... Unique power unit provides pleasing, unobtrusive appearance on top of any roof. Discharges air on a straight horizontal plane—preventing conventional back pressures and roof deterioration caused by *downward* discharge of ordinary power ventilators. Downdraft-proof against winds from any direction when fan is off (barring internal negative pressure). Operates as a gravity ventilator when power is off!

NEW! SPECIAL VENTILATORS FOR FORCED-AIR SYSTEMS!



Breidert "AIR-OUT" ... now available to meet the special requirements of forced-air duct systems. It's the same low-silhouette Powair-X-Hauster head (at left) without motor and fan assembly. Specially designed to provide an internal static resistance much lower than that of ordinary discharge heads. Like the Powair-X-Hauster, the new Air-Out discharges fumes horizontally—and boasts an architecturally-pleasing appearance to enhance any roof top!

MORE VENTILATION PER DOLLAR!



It takes fewer Breidert ventilators to do the job than conventional ventilators of the same size... **OR** *use the same number of smaller-size Breiderts to do the job of conventional ventilators.*

INSIST ON CERTIFIED CAPACITY RATINGS BASED ON VARIABLE WIND CONDITIONS—This is your assurance that the ventilators you specify will perform efficiently in actual use. Capacity ratings of Breidert ventilators are certified by *Pittsburgh Testing Laboratory*. Breidert POWAIR-X-HAUSTERS carry the Certified Rating Seal of the *Air Moving and Conditioning Association*. Breidert single-phase power roof ventilators also carry *Underwriters' Laboratory* approval.

WRITE FOR ENGINEERING DATA BOOK—44 pages of installation photos, certified capacity ratings, complete specifications.

Hirschman-Pohle Co., Inc.

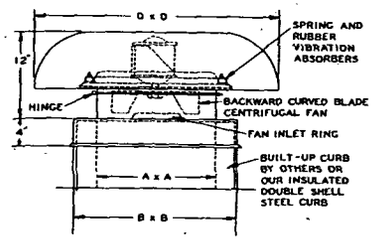
Since 1908

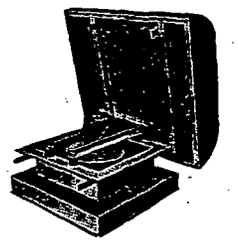
200 Lent Ave. LeRoy, N. Y.

Sales Representatives in Principal Cities

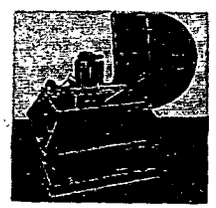
Manufactuers of all types of Power and Gravity Roof Ventilators

New Low Profile Electric Ventilator Type "CK"





The Type "CK" LOW PROFILE STATICK Power ventilator shown has been designed in response to the demand for the ABSOLUTE MINIMUM IN OVERALL HEIGHT for a ventilator using backward *curved blade centrifugal* fan, and is available in a wide range of capacities either direct or belt driven, with or without dampers or other extras as needed. The same LOW PROFILE type of housing is available using a propeller fan, or for gravity, pressure relief or intake. The latter is available in oblong shapes or sizes as may be needed, with or without our Insulated Double Shell steel curb of one of the several standard types. Any or all of these ventilators subject to modification to meet the need of any building project.



• Type LAC STATICK Power Roof Ventilator

This model features the damper housing as an integral part of the ventilator where the damper is accessible for cleaning. It shows the electric damper control mounted in the motor chamber where it is also readily accessible because of the hinged cowl.

Our Type "LAC" STATICK Power ventilator designed for efficient exhaust at either low or high static pressure, using non-overloading backward curved blade centrifugal fan, available in many sizes for almost any required capacity.

The Type "LC" STATICK Power ventilator (not shown) uses a similar construction minus the damper housing.



Capacity ratings for belt driven sizes of the Types "LC" and "LAC" STATICK Power ventilators have been formally approved by *Air Moving Conditioning Association*, and carry their CERTIFIED RATING SEAL.

See Sweet's Architectural, Engineering, or Industrial Construction Files.
Complete data on any or all types gladly furnished on request.

Jenn-Air Products Company, Inc.

1102 Stadium Drive, Indianapolis 7, Indiana

Warehouse stocks in all major cities

Roof Exhauster Equipment for Public, Commercial and Industrial Buildings

QUIET-TESTED! Jenn-Air Quiet-Tested Roof Exhausters are pre-proved under simulated field conditions. Each unit must pass rigid inspection by the critical *Vibronic Eye* . . . is "screened" in Jenn-Air's Sound-Elec Test Chamber to detect noise, vibration.



Direct Drive Power Roof Exhauster

Dependable U-Spring Suspension virtually eliminates noise caused by vibration. U-Springs suspend the power assembly, isolate it from the base. Motor and wheel are balanced together, eliminating vibration and noise. Heavy gage all aluminum construction.



Belt Drive Power Roof Exhauster

In this Quiet-Tested Belt Drive Power Roof Exhauster, Positive Weather-Proof Engineering is combined *with trim Low* Contour Design. Full Ball Bearing Drive and Motor are mounted out of the air stream. Heavy Gage All Aluminum Construction gives strength without weight. Exhauster can be quickly, easily, disassembled for re-assembly on the roof. Non-Overloading Blade, dynamically and statically balanced, also assures long life, smooth, quiet operation.



Hi-D Centrifugal and Axial Power Roof Exhauster

This exhauster fills the need for general area ventilation . . . has a clean engineering design that features compactness and serviceability. The wheel (centrifugal) or blade (axial) is located at the top, providing a high point of discharge which allows low curb construction. Housing is made of heavy gage aluminum which provides strength without weight. Bird guard of heavy gage stainless steel construction comes as an integral part of every unit. Unit opens easily for ready access to the drive assembly. Special vibration absorbers reduce noise to a minimum. Totally enclosed, full ball bearing, permanently lubricated motor assures smooth, uninterrupted service. Tubular drive assembly has permanently sealed, pre-lubricated ball bearings. Motor and drive can be serviced from interior of building, if desired.



Axial Roof Exhauster

The Axial type Jenn-Air Roof Exhauster is recommended for general ventilation work and for duct-type applications where static pressures do not exceed one-half inch. Special Non-Overloading Design protects against motor failures. Dependable U-Spring Suspension eliminates noise caused by vibration. U-Springs suspend the power assembly; isolate it from the base.



Centrifugal Wall Exhauster

Jenn-Air Centrifugal Wall Exhausters eliminate most duct work . . . provide a means of ventilation through the wall when roof exhausters would prove unsuitable or more costly, as in multi-story construction. Constructed of heavy gage aluminum, exhausters are completely weather-proof, unusually quiet. Dependable U-Spring Suspension eliminates noise caused by vibration. U-Springs suspend the power assembly; isolate it from the wall.



Relief Vent.

Jenn-Air Relief Vents eliminate goose necks or hoods used for air outlets or intakes. Weather resistant spun aluminum construction.

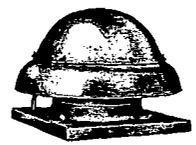
Member *Air Moving & Conditioning Association*

Jenn-Air Exhausters are tested and rated in accordance with standard test code *AMCA* Plate V-A (centrifugal) and VI-A (axial) as adopted by *AMCA* and *ASHAE*

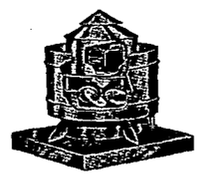
Muckle Manufacturing Co.

666 Belford Road Owatonna, Minn.

Sales Engineering Representatives in Principal Cities



MUCKLE SPUN ALUMINUM—Rating free air delivery starting at 460 cfm on larger units up to 1 in. static. Units may be had in backward curved wheels and propeller type blade. Heavy gage alloy aluminum .064 Motors are standard brands. Quiet, Sturdy, Low silhouette, Resilient mounted motors.



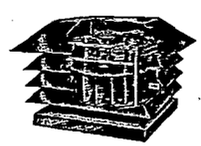
MUCKLE VENTS—are excellent for moisture laden applications or corrosive conditions. Motor is located out of the air stream, and housing is hot dipped galvanized after fabrication for lasting protection. Capacities 254 to 10280 cfm. Units very efficient at low static pressures. Constructed from heavy gage steel, electrically welded.



MUCKLE LO-SIL—designed for low silhouette appearance. Largest unit only 25½ in. high. Capacities from 225 to 17,600 cfm. Automatic back draft dampers built in on all models. Vibration separators are provided to eliminate vibration and hum. Totally enclosed, ball bearing motors are directly connected to propeller.



MUCKLE BC VENT—Centrifugal backward curved wheel is fine for operation where static pressures are encountered. Non-overloading, dynamically balanced wheel, directly connected to totally enclosed motor. No maintenance of belts, pulleys, or bearings. Motor is out of air stream for protection. Capacities from 280 to 7650 cfm. Heavy gage construction.



MUCKLE BELTED VENT—offers over 100 different capacities for any specific requirement. High exhaust capacity utilizes low fan tip speed to make unit very quiet in operation. Low in height, compact and symmetrical design, blends well with modern architecture. Heavy duty, ball bearing motors and non-overloading backwardly curved wheels used on all units.

MANUFACTURERS OF POWER ROOF VENTILATORS FOR OVER 19 YEARS
COMPLETE DATA FURNISHED UPON REQUEST

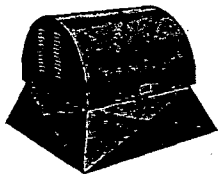
PENN VENTILATOR CO., INC.

Goodman above Allegheny Ave., Philadelphia 40, Pa.

Representatives and Distributors in Principal Cities



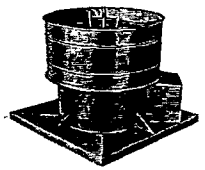
Domex



Penn Dynafan



Airette



Hi-Ex

Manufacturers of

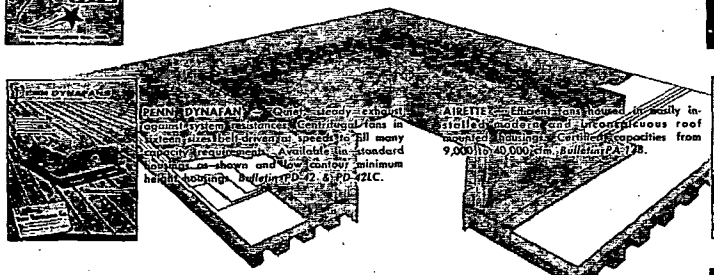
ROOF VENTILATORS AND ACCESSORY PRODUCTS
FOR PUBLIC, COMMERCIAL AND INDUSTRIAL BUILDINGS



DOMEX — V-belt and direct driven centrifugal fans in a weather-tight spun aluminum housing. Backwardly curved statically and dynamically balanced fan wheels with non-overloading characteristics. Over 60 capacity selections. *Bulletin DMXA-88.*

HI-EX — Straight through — high velocity vertical discharge air exhausters — streamlined housing. Certified capacities from 24,000 cfm to 60,000 cfm. *Bulletin PH-44H.*

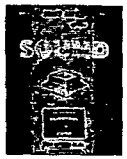




PENN DYNAFAN — Quiet, steady exhaust against system resistance. Centrifugal fans in fifteen sizes, belt driven at speeds to fill many capacity requirements. Available in standard contour as shown and low contour minimum height housings. *Bulletins PD-52, 61, PD-42LC.*

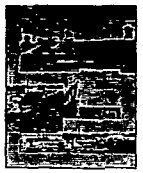
AIRETTE — Efficient fans housed in neatly installed, modern and unobtrusive roof mounted housings. Certified capacities from 9,000 to 40,000 cfm. *Bulletin PA-47B.*





SONOTROL CURB — Bulletin SC-88 contains data on Penn's roof curbs. Information pertaining to extruded aluminum curbs as well as fabricated types is outlined. In most instances where a gravity or powered ventilator is used, a roof curb is also required. Penn's acoustically and thermally insulated curbs are of superior quality and generally cost no more than field constructed curbs.

TURBINE — Ball bearing turbine ventilators are true rotary, air-driven suction turbines that work automatically and continuously without maintenance or operating expense. Ball bearing action enables the slightest motion of air to move the rotor. *Bulletin PT-24A.*











Write for any or all of the above illustrated catalogs

PENN VENTILATOR COMPANY, INC.

For More Than 25 Years, The Builders Roof-Top Line

Manufacturers of

PROPELLER FAN, CENTRIFUGAL, STATIONARY and ROTARY RIDGE VENTILATORS

Penn Ventilators are listed in Sweets Architectural and Engineering Files • Charter Member AMCA

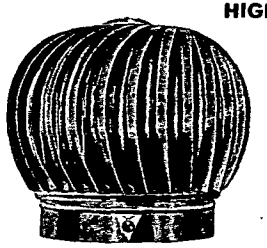
WESTERN ENGINEERING & MFG. CO.

ESTABLISHED 1921

4112 Glencoe Avenue, Venice, California

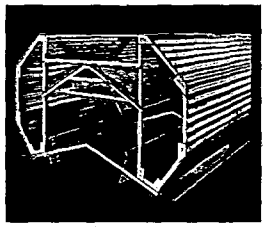
Representatives in Principal Cities

HIGH EFFICIENCY ROOF VENTILATORS



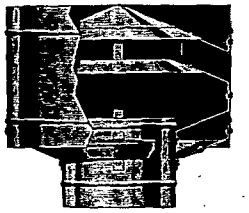
WESTERN ROTARY® VENTILATOR

Mechanical ventilation without power cost. Certified capacity ratings. Will not back-draft. Stormproof. Permanently sealed ball bearing assembly with lubricant reservoir. Rubber ISO-mounting. Baked enamel finish.



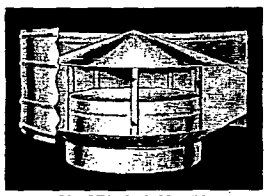
WESTERN CONTINUOUS RIDGE VENTILATOR

High volume exhaust. Architecturally desirable. Fabricated in any length in 12 throat sizes to fit any roof. Stormproof. Optional dampers (manually or electrically operated).



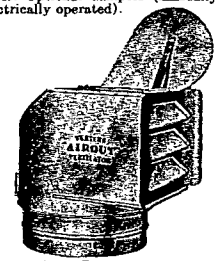
FORBES SYPHONAIRE VENTILATOR

A stationary ventilator providing high exhaust certified capacity regardless of wind direction. Rugged construction. Rainproof and stormproof. Low silhouette.



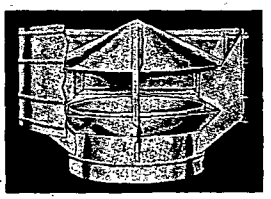
FORBES ECONOVENT VENTILATOR

Rugged design. Engineered construction at a low price. Certified capacity ratings. Low silhouette.



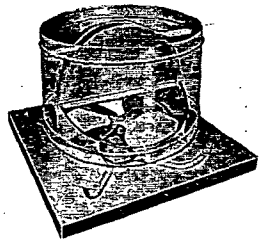
WESTERN AIROUT VENTILATOR

Directional type ventilator. Moves in the direction of airstream for maximum exhaust power. Certified capacity ratings. Balanced, rugged and stormproof.



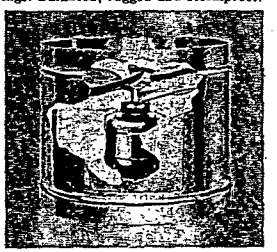
FORBES AMVENT VENTILATOR

Government specification ventilator. Meets Corps of Engineers Guide Specifications CE 220.09, October 5, 1951. Removable top plate. Optional bird or insect screen or damper.



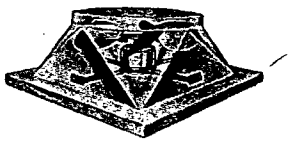
WESTERNAIRE® VERTICAL EXHAUST VENTILATOR

High capacity industrial "blow open" exhauster. Stormproof. Low silhouette. Prevents roof damage or re-entry of air in building. Fourteen gauge construction.



WESTERNAIRE® BOOSTER FAN

Duct mounted fan installs between ventilator and base. For new or existing construction. Permits ventilator exhaust when fan off.



WESTERNAIRE® CURB MOUNTED FAN

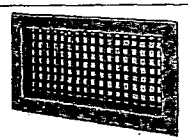
Provides fan diameter same size as throat diameter for maximum exhaust. Available in 40 models. Capacities based on NAFM ratings.

See our complete Catalog in SWEET'S *ARCHITECTURAL FILE, INDUSTRIAL CONSTRUCTION FILE* and *WESTERN STATES A-E-C CATALOG FILE*

AIR CONTROL PRODUCTS INC.

657 Center St. Coopersville, Michigan

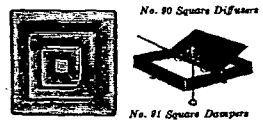
Air Conditioning Registers, Grilles, Diffusers—Ceiling and Baseboard Diffusers—Commercial Registers and Grilles



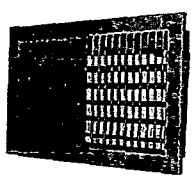
MULTI-TROL REGISTERS: Commercial type with special butterfly "Shallo-Valve", so shallow it can be used with standard 3¼ in. fittings. Fully-adjustable face bars, with rolled pivots that can't pull out of frame. Single or double deflection types. Beige prime coat finish. Full range of types and sizes. Matching grilles, all sizes.



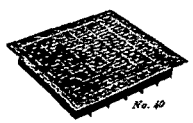
No. 76 BIG CAPACITY CEILING DIFFUSER: Up to 50 percent more capacity than ordinary round diffusers. Plenty of free area for use on ducts of same listed size. New "Air Flow" rings deliver air at lowest resistance. Contoured anti-smudge outer ring. Sponge rubber gasket. Flush, Step-Down and Adjustable types—6, 8, 10, 12 and 14 inch sizes. Beige prime coat.



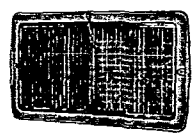
No. 90 BIG CAPACITY CEILING DIFFUSER: Up to 40 percent more capacity than ordinary square diffusers. Same engineering, performance and appearance advances as No. 76. In Step-Down style only—6, 8, 10, 12, 14 and 18 in. sizes.
NEW "STA-SET" DAMPERS in both round and square models have butterfly valve, nylon friction bearing and operator rod with Adjusto-Stop collar.



No. 120 SERIES MULTI-LOUVER REGISTERS: Commercial type with one-piece face, stamped fins ½ in. wide. Great strength, large free area. Fins adjust for horizontal control of air pattern. Multi-Louver valve for 45 deg downward deflection—any upward deflection. Removable operator handle. Adjusto-Stop. Beige prime coat finish. Matching grilles, all sizes.



No. 40 FLOOR REGISTER: Features the famous Rigid-Lock type construction with each fret locked to each crossing ing fret and to margin. Dial-operator valves run the short way of the face. Medium mesh (3/16 in.) between frets. Finished in Oak or Metalescent—all standard sizes. **No. 41 FLOOR RETURN AIR GRILLES** have same construction features.



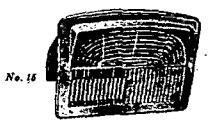
No. 20 SERIES AIR CONDITIONING REGISTERS: Positive control of air stream with vertical front fins, horizontal back fins—both readily adjustable from face. Damper opens to full 90 deg. Adjust-Stop for balancing system at face. Models for sidewall, baseboard and inverted installation. 8 × 6, 10 × 6, 12 × 6, 14 × 6 duct sizes. Matching grills, all sizes, plus 24 × 6 and 30 × 6. Two-tone beige finish.



No. 188 BASEBOARD PERIMETER DIFFUSER: With built-in rotary damper to insure uniform velocity at face at any setting. 2-ft and 4-ft lengths with Adjusto-Bottom base strips. Both diffusers and matching grilles can be installed before or after plastering, on subfloor or finish floor. Also **No. 170 BASEBOARD DIFFUSER** and matching grille for low-budget installations.



No. 42 PERIMETER FLOOR DIFFUSER: Blankets outside wall with warm air, prevents drafts. Has one-piece face, extra-wide margins, smoothly-curved vanes and wafer-thin valve. Set vanes for any deflection without shearing off at ends. Opening sizes 14 × 6, 12 × 6, 14 × 4, 12 × 4, 10 × 6, 10 × 4, 10 × 2¼, 14 × 2½ and 12 × 2¼. Also **No. 150 ECONOMY FLOOR DIFFUSER** for low-budget installations.



No. 15 SIDEWALL PERIMETER DIFFUSER: Rapidly diffuses warm air over outside wall without "scrubbing". Easily balanced at face. Sponge rubber gasket. Sizes 10 × 6, and 12 × 6. Beige prime coat or Metalescent finish. **"E" FRAME** for baseboard installation. Also *No. 16* **OUT-OF-WALL DIFFUSER** for use where stackhead and duct cannot be put in wall.

WRITE FOR AIR CONTROL CATALOG FOR COMPLETE DESCRIPTIONS, ENGINEERING DATA, SIZES

ADP AIR DISTRIBUTION PRODUCTS

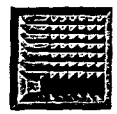
116 S. La Brea Avenue
Los Angeles 36, California



S-3 — Three-Way Throw True 3-way distribution, no spillage



S-4 — Four-Way Throw 360 deg distribution High Induction



SC-2 — Two-Way Corner Complete 90 deg distribution



S-2 — Two-Way Throw Side throw balanced to area served



S-1 — One-Way Throw Side throw balanced to area served

ADP diffusers are designed and engineered for better air distribution. Greater free areas permit greater efficiency. The unique but simple construction combined with high induction characteristics result in top performance.

The inner core assures a balanced air delivery over the collar and face of the diffuser. No blank-offs or baffles are required to meet special pattern or delivery requirements. *ADP diffusers* provide rapid temperature equalization with quiet, draftless efficiency.

The greater efficiency of ADP diffusers permit the use of smaller ducts and a material saving on installation costs.

ADP diffusers are made in a complete range of sizes and face patterns. The air to be delivered may be proportioned in any required direction. The Type S frame, as shown, is standard, however, *ADP diffusers* are available with flat frames, tile replacement frames and with removable cores.

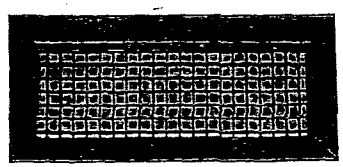
ADP diffusers are fabricated of cold rolled steel and welded throughout. Large diffusers are made of aluminum. The standard finish is baked on silver-tone enamel. Special finishes are available.

The TOB volume control has true-opposed blades. The blades are pivoted at edge and orifices remain constant. No contrasting velocities. Fine adjustment with fast action.

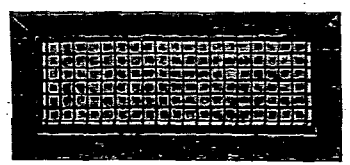
ADP diffusers and volume controls have been thoroughly field tested. Thousands of ADP diffusers are now performing with satisfaction in all types of industrial, commercial, residential and special applications.

All ADP diffusers have a smart trim appearance to blend with the decor of todays modern construction.

Sales engineering offices are located throughout the United States. They will welcome an opportunity to discuss your diffuser requirements.



K-310



K-130

ADP manufactures a complete line of adjustable bar grilles, registers and volume controls.

The *"K" series shown above is* available as a grille or register and in *single or double deflection with or without* attached dampers.

The new frame design gives an improved and attractive appearance and will add to any architectural treatment.

The "K" series is fabricated of extruded aluminum and the frame is of welded construction.

The solid section adjustable bars are of extruded aluminum and are spaced on ¾-in. centers and give top performance at a minimum noise level.

A complete line of volume controls is available.

Opposed blade dampers are adjustable through face of register with a removable key.

Dampers with individually adjusted blades are available.

Extractors are manufactured for permanent setting or with key operation.

We suggest you contact your ADP representative or write for catalog.

AIR DEVICES INC.

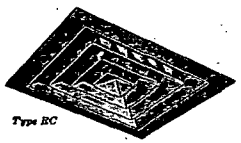
185 Madison Ave.
New York 16, N. Y.

THERMOTANK AGITAIR

registers and grilles • high velocity units • punkah louvers air diffusers • filters • exhausters

Representatives in Principal Cities

AGITAIR DIFFUSERS



Type EC

CUSTOM MADE DIFFUSERS

AGITAIR square and rectangular air diffusers incorporate built-in diffusing vanes, scientifically arranged in unlimited louver patterns to provide draftless, noiseless air distribution in 1, 2, 3 or 4 directions from any ceiling or side wall location.

These AGITAIR diffusers are custom made to suit each job condition and are available in three distinct types, "RC" with separate mounting frame and interchangeable diffuser cores; "R" and "RV" series, one piece units. *Catalog R-107* contains dependable, accurate performance data.

MR series matching return units (without diffusing vanes) are available in unlimited pattern designs, square and rectangular in shape to harmonize with AGITAIR supply diffusers. MR units are recommended only for return or exhaust applications.



CIRCULAR DIFFUSERS

AGITAIR circular air diffusers are made in two types—"O"—non-adjustable and—"OA"—adjustable. The graceful lines of the spinnings are identical for both types.

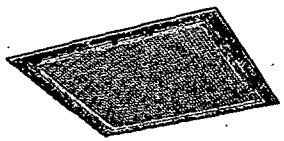
The "OA" adjustable units control the air direction at four different angles of discharge simultaneously. Air adjustment of these units is accomplished after installation and without changing the position of the spinnings. For normal ceiling distribution, Type "O" is recommended. *Catalog C-101* contains all dependable performance data.



STRIPLINE

STRIPLINE slot type diffusers are slenderly designed, inconspicuous, practical and versatile.

STRIPLINE combines the best features of both slots and efficient air diffusers to insure equalized air flow throughout the entire length of the diffuser, with . . . high rate of temperature equalization . . . gentle secondary air movement . . . even temperature in area served . . . quiet operation. STRIPLINE is available in various widths (some only 2 in. wide) and standard 4 ft lengths which can be butted together for unlimited runs. STRIPLINE can be located almost anywhere to suit interior design—in walls, ceilings, coves, moulds, window reveals or stools. *Catalog S-104* contains complete dependable engineering data.



PERFAIR
Perforated Air Diffusers

AGITAIR PERFAIR square and rectangular diffusers incorporate interchangeable cores available in unlimited air pattern arrangements for 1-, 2-, 3- or 4-way blows. These units also include built-in air controllers for positive adjustment of air deflection after unit is installed. The perforated face plate of PERFAIR blends in naturally with acoustical ceilings. Matching returns or exhaust units harmonize to enhance the architectural design. Separate mounting frames allow quick and easy installation of supply or return units. *Catalog P-200* contains dependable performance data.

THERMOTANK-AGITAIR PRODUCTS

The AGITAIR diffusers illustrated on this page and the THERMOTANK products shown on page 242 are sold exclusively by representatives for Air Devices Inc.

Complete engineering data, sizing and selection of these THERMOTANK-AGITAIR products are available from Air Devices Inc., 185 Madison Ave., New York 16, N. Y.

See THERMOTANK INC. ad on page 242.

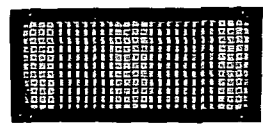
Air Devices Inc.

A complete line of Registers, Grilles and Scoops for all types of Industrial and Commercial air conditioning applications

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AGITAIR

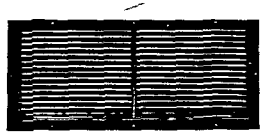
Agents in All Principal Cities.



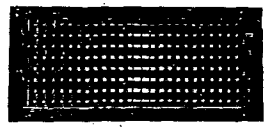
DDV A double deflection grille affording four way diffusion—individually adjustable streamlined fins ¾ in. C.-C. heavy gage metal and sturdy construction.



DDVO A double deflection grille with opposed action valves. The valve consists of a series of hollow, internally reinforced air-foil sections. Through a unique actuating mechanism each air-foil section turns in a 90 deg arc toward a similarly turning section.

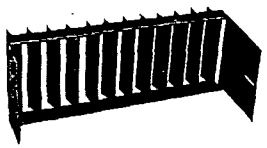


DG An exhaust grille with non-adjustable fins. Fins are normally factory set at 45 deg down angle to restrict vision. Also available with fins set at 0 deg vertical or horizontal.

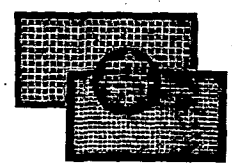


DVL A directional-type register employing 1 in. wide, parallel-acting valves. Face fins are individually adjustable right and left.

The 1 in. wide, parallel-acting valve mechanism is available with any/all STEWART face patterns.



SCOOPTROL A device designed to deflect and control the air from a supply duct into a collar, and installed at the junction of the duct and collar. Scooptrol is a double-blade functional product, easily adjusted, positioned and then fastened by sheet metal screws; it cannot move or fall.



PLAIN LATTICE GRILLES Available in satin bronze, brass and aluminum. Also plated finishes on steel.

FINISH DIRECTIONAL TYPE OUTLETS ARE PROTECTED BY A PRIME COAT AND A FINISHING COAT OF *SILVAIR* ENAMEL BAKED ON. *SILVAIR* FINISH IS A MODERN, ATTRACTIVE COATING CONTAINING ALUMINUM PIGMENT, AND IS SILVER-BLUE IN TONE. NO ADDITIONAL PAINTING IS REQUIRED.

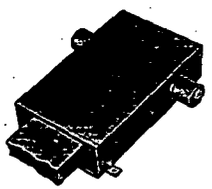
THERMOTANK INCORPORATED

11191 Lappin Ave., Detroit 34, Michigan

Toronto, Canada Glasgow, Scotland London, England Melbourne, Australia Hamburg, Germany Oslo, Norway Basrah, Iraq Johannesburg, S. Africa

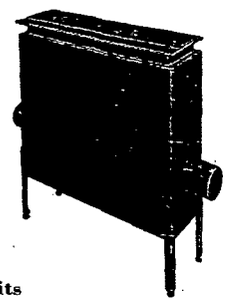
THERMO-REG UNITS

Thermo-Reg mixing and sound attenuating chambers for high-velocity air conditioning systems, single or dual duct, are available as (1) ceiling type or (2) under-window-type.



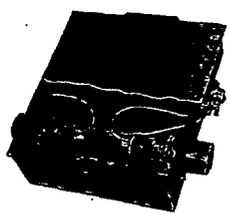
CEILING UNIT

Thermo-Reg ceiling units for rooms or corridors serve one direct connected AGITAIR diffuser or several remote diffusers. They offer considerable flexibility to accommodate room modifications or changes in internal loads after installation.



PERIMETER UNITS

Thermo-Reg perimeter units, operating at a low sound level, supply conditioned air through the AGITAIR Stripline diffuser in the top or front of each unit. In single duct systems, room conditioners are controlled by a zone thermostat. In dual duct systems, they are individually controlled by a thermostat in the space served. Ask for *Catalog TA-101*.



CON-VOL Equipped High Velocity Mixing Units

CON-VOL, a unique device now available in all Thermo-Reg Constant Volume perimeter and ceiling units assures precision control of air volume and temperature to meet all requirements regardless of inlet static pressure variations. *CON-VOL utilizes flow* regulators and standard pneumatic motors mounted on a sliding base. The entire system eliminates *the need of complicated linkages.*

DRUM PUNKAH LOUVRES



These units are specially designed for those heating or ventilating applications requiring extremely long throw. These units are suited for installations where large spaces must be served with a minimum of duct work. DRUM *PUNKAH LOUVRES* with directional air control and long throw are particularly adapted to large spaces in industrial plants, auditoriums and commercial establishments. Engineering data available.

ROUND PUNKAH LOUVRES



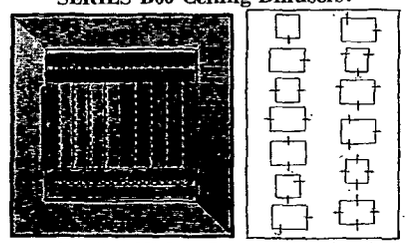
PUNKAH LOUVERS are nozzle-type outlets incorporating principles of high velocity air delivery with individual directional control *for spot* heating or cooling applications. These units are widely used in applications for air conditioning, heating and *ventilating* in ships, industrial plants, aircrafts, telephone exchanges and a variety of other applications. For complete data ask for *Catalog*.

THERMOTANK-AGITAIR Products illustrated on this page are sold exclusively by representatives for Air Devices Inc., 185 Madison Avenue, New York 16, N. Y. See Air Devices Inc. ad on pages 240-241.

AIR-FACTORS INC

1624 South Raymond Ave. **Monrovia, California**
Air Distribution **Air Control**

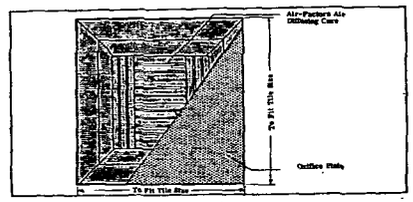
SERIES B60 Ceiling Diffusers:



MODEL LCF-RC (Removable Core)

Air-Factors' Series B60 Ceiling Diffusers for mounting on ceilings through patented construction provide proportionately balanced air distribution with controlled air motion through laminar counter-flow cores. Square or rectangular air diffusing outlets with full range sizes and capacities allow maximum mixing of primary and secondary air with distribution of supply air in direct ratio to floor area.

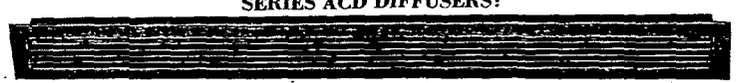
MODEL MO-MJ (Series T)



Patents applied for.

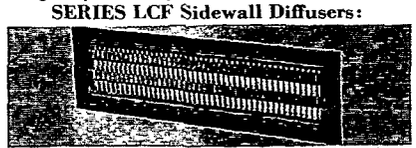
Air-Factors' Model MO-MJ recessed, square and rectangular induction type ceiling air diffusers are designed to comply with modern architectural tile ceilings and can be attached and fitted to any standard ceiling structure. The diffuser discharges supply air substantially parallel to the surface upon which it is mounted, inducing a large percentage of room air into the hooded section to mix with the supply air being discharged. On-the-job adjustment to the discharge angle is provided. One, two, three or four-way unbalanced or balanced air patterns are available.

SERIES ACD DIFFUSERS:



Air-Factors' Series ACD Air Curtain Diffusers are designed for ceiling installations where it is desirable to blanket a single or perimeter exposed wall with a curtain of air. It may be used to curtain any area bordering a conditioned space. Special butt ends are provided for continuous strip installations from 1 in. to 4 in. in width.

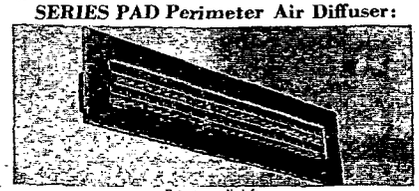
SERIES LCF Sidewall Diffusers:



MODEL LCF-RC (Removable Core)

Factory set outlets distributing air in laminar counter-flow layers mix supply with room air reaching design conditions in minimum time and space. LCF outlets can be equipped with secondary flexible horizontal vanes which may be reset on job, giving maximum, vertical control with varying length of blow.

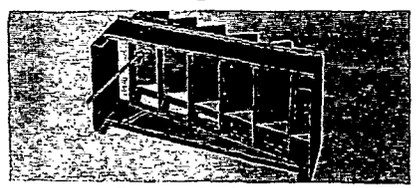
SERIES PAD Perimeter Air Diffuser:



Patents applied for.

Air-Factors' PAD Perimeter Air Diffuser provides for the perimeter system of air distribution. A register with built-in diffusing core. The core is factory set to distribute the air to reach comfort conditions in a minimum of time and space. The laminar counter-flow air deflecting vanes provide a maximum divergence angle to blanket a greater area of perimeter surfaces. The perimeter diffuser is provided with a dial operated volume control valve.

SERIES AVP-QO Adjustable Air Proportioning Vanes:



Balances air distribution for given supply outlet. Quick opening feature extracts air with a 2½ in. maximum entrance into main duct, reducing velocity evenly over outlet face. Full shutoff. Manufactured separately or attached to sidewall ceiling outlet or diffuser. Key or rod operated.

SERIES OBD-KO Air Volume Control:

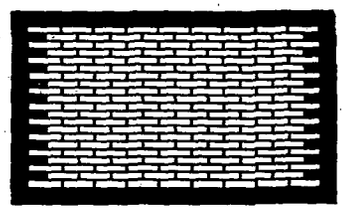


Opposed acting blades; center pivoted. Operate from full open to full closed position. Used directly attached to grille, diffuser or installed as separate unit. Key operated. Friction linkage maintains blades in position when set.

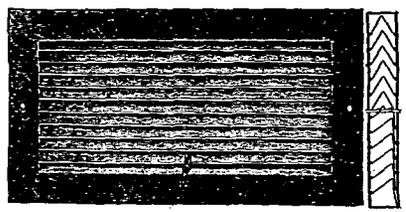
A-J Manufacturing Co.

3601 East 18th St. Dept. Y Kansas City 27, Missouri

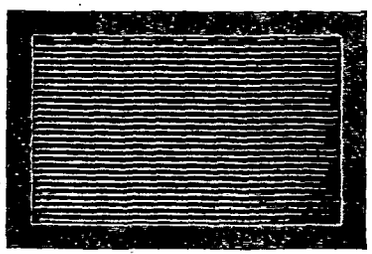
REGISTERS GRILLES DIFFUSERS



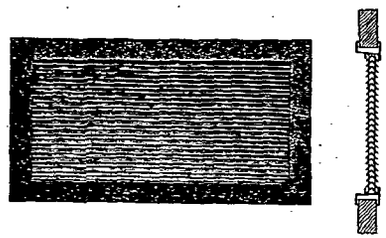
A-J Architectural Grilles are available in slotted designs B, C, and D, and square meshes of ½, ⅝, ¾, and ⅞ in. The grilles can be made to your exact order from steel, aluminum, bronze, monel or stainless. There is practically no size limitation. See A-J illustrated catalog for specifications.



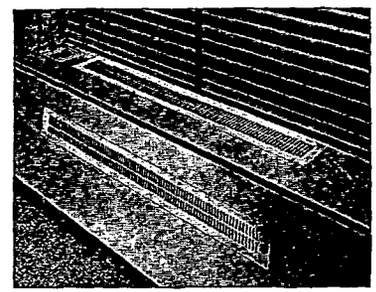
A-J A-76 No-Vision Grille, shut-off type. It has extruded aluminum V-bars and flange frame. Efficient in free area. Depth is 5/8 in. As in other styles of A-J no-vision grilles, the auxiliary frame telescopes over core from opposing side for fast, simple and neat installation. Spans 1¼ in. to 2⅛ in. depth. Cutaway view shows open and closed positions.



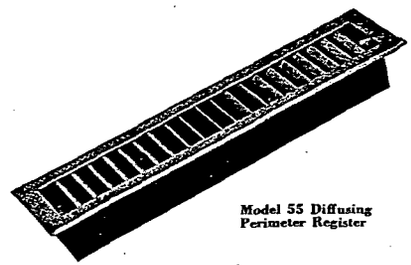
A-J A-7000-B Grille. This extruded all-aluminum no-vision grille blends attractively with contemporary interiors. Extruded aluminum V-bars-(3 to the in.) and flange frame. Resists corrosion, rust. Lightweight, approximately 70 percent free area. Mitered corners. Auxiliary frame telescopes over core from opposing side for neat, finished appearance in doors or partitions. Span from 1¼ in. to 2⅛ in. Grille 5/8 in. deep.



A-J No. 77 Series Thin Core Grilles are only ½ in. thick, yet are completely sight proof. Approximately 80 per cent free area. With auxiliary frame, the No. 77 series can be installed in panels or partitions from ¼ to ¾ in. thick. Illustration above shows grille and cut-away view of grille and auxiliary frame. No. 77-A is all steel with channel frame. Also available without framing—No. 77-C.

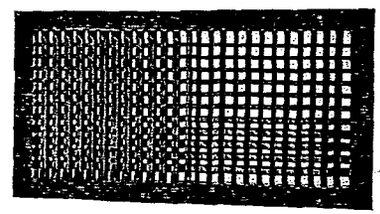


A-J Convector Grille and Register. These are custom-made to your specifications. Architectural grille of aluminum, munts, stainless or cold-rolled steel covers adjustable bar-type louvers, single or double bank, to direct air flow in desired pattern. Setting frame minimizes installation time. Construction eliminates tampering after installation.

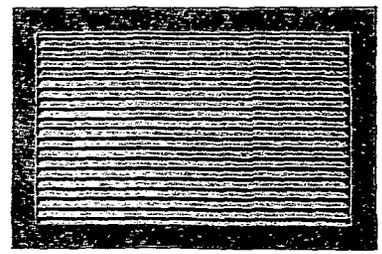


Model 55 Diffusing Perimeter Register

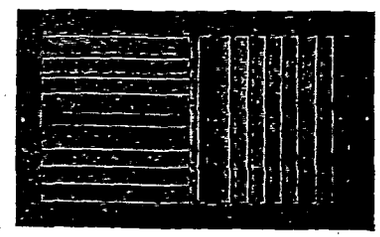
A-J No. 55 Diffusing Perimeter Floor Registers. Heavy gage curved vanes are factory set to deliver a fan-shaped air pattern. Ideal for installation in stair risers, under kitchen cabinets, in picture window sills, or as side wall outlets with high velocity systems. Valve louvers may be set with special lock screw after system is balanced.



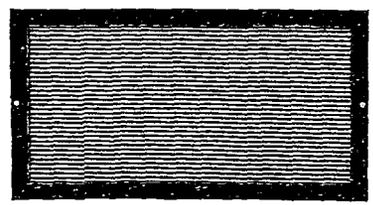
A-J No. 400 Series Air Conditioning Grilles and Registers. Extruded aluminum bar-type grille, steel frame. Single or double bank adjustable; in horizontal, vertical deflection or combination of either. No. 440 styles have opposed blade damper. The No. 430 register has single bank of bars with multiple valve body, removable handle. All-aluminum faces on styles A-400 and A-440.



A-J No. 42 Series Return Air Grilles. Available with either horizontal (42-H) or vertical (42-V) face bars on ½ in. centers. 45 degree deflection for minimized see-through. Overall, 1¾ in. beyond duct size; depth 5/8 in. All steel, gray prime finish. No. 432 has multiple valve body, removable handle; with opposed blade damper, No. 442.



A-J No. 200 Series Ceiling and Side Wall Diffuser. Available in a wide range of deflection patterns. Standard construction is steel frame with extruded aluminum, adjustable, curved face bars which are 1¼ in. wide on ¾ in. centers. Overall size is 1¾ in. greater than duct size. Approximately 75 percent free area. Aluminum enamel finish. No. 230 has register body, multiple valves, removable handle. No. 240 is the opposed blade damper style.



A-J No. 75 Grille. For dark room or other application which requires ventilation without light or sight. Steel construction throughout. Free area approximately 60 percent. With flange frame, No. 75-B; with channel frame, No. 75-A. Depth of No. 75-B grille, 1¾ in. Spans 3¼ in. with standard auxiliary frame. Non-specular black only.

FOR COMPLETE SPECIFICATIONS AND PRICES ON ALL A-J PRODUCTS, WRITE FOR ILLUSTRATED CATALOG

THE AUER REGISTER COMPANY

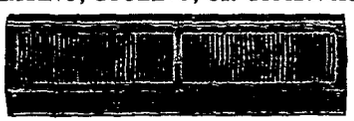
REGISTERS AND GRILLES FOR EVERY HEATING AND COOLING NEED

6610 CLEMENT AVENUE • CLEVELAND 5, OHIO

In Canada: Superway Products, Tilbury, Ontario

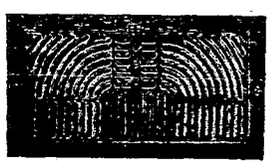
Auer features a complete line to meet every heating, cooling, or combination system requirement. The Auer line exemplifies correction of design and construction—decorator styling—easy installation.

PERIMETER HEATING, COOLING, OR COMBINATION SYSTEMS



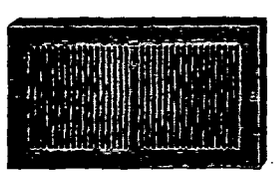
"PERFUSAIRE"

"PERFUSAIRE" . . . fits 2¼ in. x 12 in. and 2¼ in. x 14 in. openings. Only 18 in. long and 4¼ in. high, the "Perfusaire" has spread and capacity of 4 to 8 ft. units. "Limit-Lock" permits perfect balancing of system. No. 100.



"FANAIRE"

"FANAIRE" . . . for high or low sidewall installation. Efficient, low-resistant, perimeter air diffusion. Sizes 10 x 6, 12 x 4, 12 x 6 and 14 x 6.



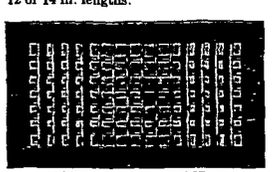
"AIR-FLEX" SERIES

"AIR-FLEX" SERIES . . . Wall register, Grille bars and multi-louvers adjustable to direct air flow in all directions. Grille to match. No. 4432.



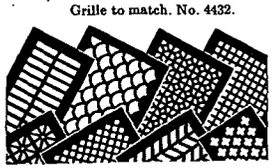
AUER "DRP" FLOOR PERIMETER REGISTER

AUER "DRP" FLOOR PERIMETER REGISTER . . . for intricate, "narrow-type" installations. Available with built-in Damper in 2¼, 4 and 6 in. widths; 10, 12 or 14 in. lengths.



"STREAMLINER" SERIES

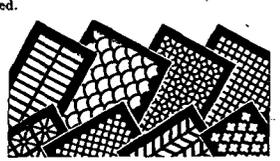
"STREAMLINER" SERIES . . . For high velocity air-conditioning systems. Obtain maximum diffusion, silent operation. SINGLE BAR MODEL: "Streamliner" Bars vertical in front, with horizontal multiple louvers in rear. Louvers operated by lever on face. DOUBLE BAR: No. 1205 VH-OB—double deflection grille with 2 banks of "Streamliner" bars. Front bars vertical, rear horizontal. Opposed blade damper louvres easily adjusted.



ORNAMENTAL GRILLES

ORNAMENTAL GRILLES . . . Complete line available in steel, stainless steel, brass, bronze or aluminum. Large variety of decorator designs and finishes.

WRITE FOR COMPLETE DESCRIPTIVE LITERATURE



BARBER-COLMAN COMPANY

1300 Rock Street, Rockford, Illinois

Field Offices in 95 Cities Throughout the United States and Canada—consult local phone book

A Complete Line of Engineered Air Distribution Products

Barber-Colman air distribution products are engineered "to make air behave," to eliminate complaints: "Too hot—too cold—too drafty." They provide uniform, properly controlled rate of flow and diffusion of air being introduced into the room. Their performance is guaranteed.

Barber-Colman offers you years of combined experience in both air distribution products and automatic controls and will help you develop the correct relationship between room temperature, velocity of air being introduced to the room, and temperature of moving air in relationship to average air temperature in the room to provide expected human comfort conditions.

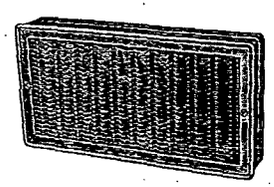
(See Barber-Colman page in Controls and Instruments Section.)

Complete descriptive literature and selection data available upon request.

Grilles & Registers



Model MA sidewall diffuser (has removable core)



Model FA sidewall diffuser (has removable core)



Model ST continuous line diffusing grille for under-window and sidewall installations.

Uni-Flo grilles and registers provide rapid diffusion, a high rate of aspiration with resultant high air entrainment ratio, low pressure drop, low noise level, simple adjustment in the air pattern, and a pleasing appearance. A variety of frame styles is available to meet sidewall, panel, or exposed duct installation requirements. The removable core feature makes it unnecessary to break the paint or plaster seal for cleaning or decorating purposes.

Barber-Colman also manufactures a complete line of return, exhaust, and transfer grilles, including the Sight-Tite core for use in doors to provide adequate ventilation without unsightly "see-through."

Accessories

Barber-Colman accessories include opposed-blade volume controls—Voladjusters, for remote manual adjustment of air volume—Airturns and Deflectrols, for efficient flow of air around turns and into take-offs.

Ceiling Diffusers



Venturi-flo ceiling diffusers

Venturi-flo ceiling diffusers feature quiet operation, rigid construction, wide range of sizes, efficient diffusion, adjustable deflection volume control, and attractive appearance. They are available for supply and return service in both recessed and flush mounting styles, as well as in combination with lighting units. A complete line of accessories includes volume controls, dampers, and Deflectrols.



Uni-Flo square and rectangular diffusers with perforated diffuser plate

Square and rectangular ceiling diffusers are available with attractively designed faceplates, both recessed and surface type. Air pattern modification and volume control are provided without altering the outward appearance of the diffusers.



Uni-Flo square and rectangular diffusers with louver face design

Square and rectangular diffusers are also available with a louver face design to meet specific architectural requirements. These are available with a variety of vane assemblies to provide 4-way, 3-way, 2-way, or 1-way air patterns.

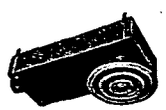


PARAFLO combination air distribution and lighting unit

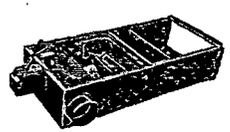
This new troffer provides precision-engineered air diffusion and high comfort illumination in a single attractive unit. Adjustment can be made from below. Air distribution assembly by Barber-Colman, lighting by Day-Brite Lighting, Inc.

High Velocity Units

High velocity air distribution, pioneered by Barber-Colman, requires technically designed units which will reduce the high velocity air to conventional velocities and correctly distribute it to the occupancy zones at minimum noise levels. Barber-Colman offers a complete line of single- and double-duct control equipment including constant volume control.

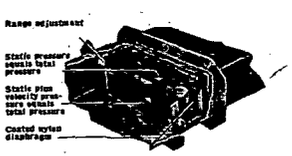


Barber-Colman high velocity control unit with Venturi-flo diffuser



Uni-Flo double-duct constant volume control

The new Uni-Flo double-duct control unit with self-contained constant volume control simplifies stabilization of high velocity systems. It mixes hot and cold air and reduces it to conventional velocity at minimum noise levels. It is self-operated, requiring no power source or motor operator. Cfm delivery and air mixture temperatures are held within very close limits.



Model SCR constant volume regulator

The Model SCR constant volume regulator illustrated above is applicable anywhere in a duct system where there are changes in static pressure. No motor operator or other power source is required. It provides extremely accurate constant volume control, reducing high velocities to conventional velocities for branch ducts.

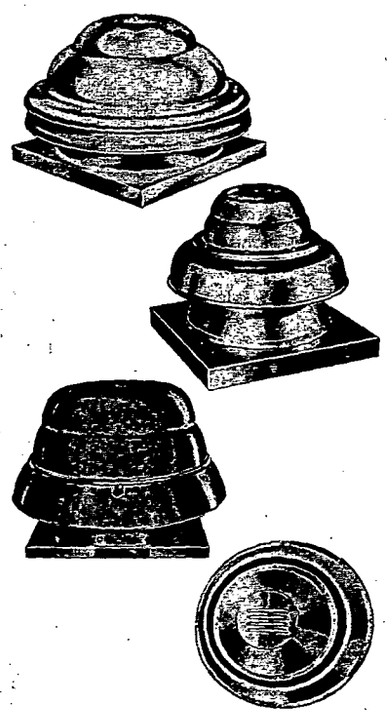
CARNES CORPORATION

VERONA, WISCONSIN

Roof and Sidewall Ventilators—Air Diffusers—Registers and Grilles—Baseboard

ONE OF THE MOST COMPLETE LINES OF VENTILATORS

The Carnes line of ventilators includes a wide range of models and a complete selection of sizes and capacities—to fulfill every requirement. Features include—quick, easy installation; aluminum housings and curb bases; heavy duty sealed ball bearing motors, installed out of air stream; motors and fan assemblies mounted on high quality isolators, and guaranteed performance.



MODEL DI

Extremely low silhouette, especially designed for low parapet buildings. Has a backwardly inclined, non-overloading fan wheel which is direct drive. Housing is heavy gage, corrosion resistant aluminum. Available in a wide range of sizes and capacities.

MODEL AX

Heavy gage aluminum housing and aluminum, non-sparking propeller fan direct motor drive with motor and fan assembly mounted on resilient mounts to eliminate vibration and noise. Square curb base is standard and is furnished attached when ordered with the ventilator. Several sizes in a wide range of capacities.

MODEL BE

Low contour for low parapet buildings. Housing is heavy gage, corrosion resistant aluminum. Backwardly-inclined fan wheels, statically and dynamically balanced to provide years of smooth, quiet operation. Large selection of sizes and capacities.

MODEL WD

Modern streamlined, centrifugal side-wall exhauster. Quickly and easily installed. Can be used with duct work, or as a straight exhaust. Each motor and fan unit is mounted on full-floating rubber vibration isolators to reduce vibration and noise to an absolute minimum. Available in capacities from 120 cfm to 2295 cfm.

CARNES CORPORATION

VERONA, WISCONSIN

Air Diffusers—Registers and Grilles—Roof and Sidewall Ventilators—Baseboard

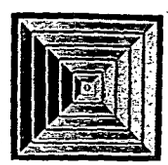
AIR DIFFUSERS

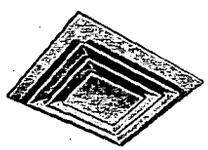




MODEL EJ is a completely adjustable diffuser for an infinite number of air patterns. Pattern adjustment is easily made by turning center cone. Ideal for installation where cooling, heating or ventilating uses the same duct.

MODEL EH is half-round diffuser—adjustable for use on ceiling or sidewall. Can be set for either throw at the factory, or easily changed after installation.

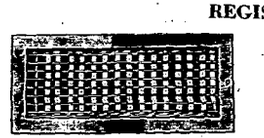




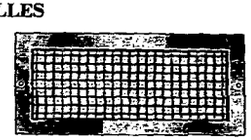
MODEL K available in one-way, two-way, three-way and four-way blow in square or rectangular models—for surface or recessed installations. Corner air scoops for an even 360 deg pattern. Inner vane removable for easy installation.

MODEL RC AND RCD Convertible Diffuser for heating and cooling. Simply adjusted without tools or disassembly for a two-way, three-way or four-way air distribution pattern. Available with opposed blade damper as an integral part of unit.

REGISTERS and GRILLES



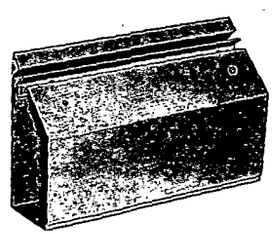
MODEL 100 Single Deflection Supply Register with opposed blade damper provides air deflection in two directions and positive volume control. Face bars individually adjustable. 18 gage steel frames. Also available with vertical front bars. 26 standard sizes.



MODEL 250 Double Deflection Supply Grille. Available with horizontal or vertical front bars. Adjustable rear blades. 18 gage steel frames –rounded edges—sponge rubber gaskets. In 26 standard sizes.

A complete line of registers and grilles available—write for catalog.

BASEBOARDS

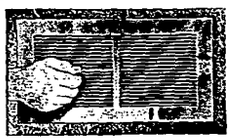


For continuous, even air diffusion, Carnes-Brandes is designed to meet the needs of architects and engineers for exacting requirements. There are standard models for all residential and commercial applications—and a complete selection of varied types. Write for *Catalog No. C-B108*

HART & COOLEY MANUFACTURING CO.,

HOLLAND, MICHIGAN

Air Conditioning Registers, Grilles & Diffusers—Warm Air Registers—Damper Regulators—Furnace Regulators—Pulleys—Chain



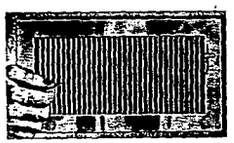
No. 74 DESIGN
ECONOMY TYPE AIR CONDITIONING REGISTER

Flexible-fin face permits any up or down deflection of air-flow. Positive valve with volume control for accurately balancing system. Gasket prevents streaking. Sidewall and Baseboard Registers in sizes 8 x 4 through 14 x 8. Grilles 8 x 4 through 30 x 8.



No. 76 DESIGN
IDEAL REGISTER FOR SHALLOW OR HORIZONTAL DUCTS

Multi-shutter valve (depth 1⅝ in. from wall) and flexible-fin face provide for all deflections desired. Any up or down deflection can be maintained with adjusting screw. Sealed to prevent streaking. Sidewall and Baseboard type Registers in sizes 6 x 4 through 30 x 8.



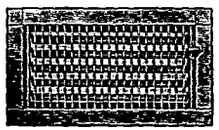
No. 88 DESIGN
SHALLOW DUCTS, LARGE INSTALLATIONS

Similar to No. 76 except that face bars are adjustable in 2½ in. sections (one moves all in section). Exceptionally good for large installations. Removable handle available to avoid tampering. Sidewall Registers or Grilles 6 x 4 through 30 x 24.



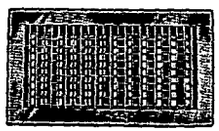
FIXT-AIRE® RETURN AIR GRILLES AND REGISTERS

Similar in construction and perfect complement to TRIPL-AIRE line. Same louver, but face bars (single bank) are fixed in either horizontal or vertical position. Grilles may be had with horizontal face bars set at 22 degree angle for up or down deflection.



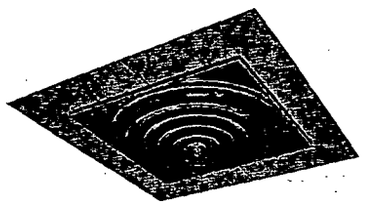
No. 821 DESIGN
COMMERCIAL-RESIDENTIAL A.C. REGISTER

Multiple deflection provided by parallel action louvers, adjustable to any upward angle and downward to 45 deg, and pivoted face bars. Vertical or horizontal louvers and face bars, snap-in valve positioning handle removable to prevent tampering with setting, shallow depth (1⅝ in. - valve open). Face bars pivoted at front to leave appearance of face uniform regardless of position. No. 831, not shown, has vertical louvers, horizontal face bars. Many standard sizes. Costs less than TRIPL-AIRE line. Ideal for year around air conditioning, particularly where low sidewall registers are presently installed.



TRIPL-AIRE® SIDEWALL REGISTERS
EXCELLENT FOR COMMERCIAL TYPE INSTALLATIONS

Provide any deflection or combination of deflections desired. Opposed louvers afford absolute volume control. 26 standard sizes up to 36 x 12 ... each available in 10 combinations of horizontal and vertical front and secondary face bars and louvers. All key adjusted from front.



No. 16 CEILING DIFFUSAIRE®
IN No. 18 ADAPTER SQUARE

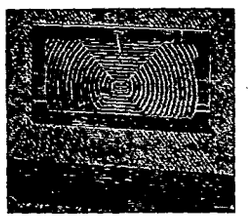
Both this diffuser and the No. 15 on the opposite page are designed to take full advantage of pipe sizes thus providing highly efficient cfm delivery at a minimum resistance. Butterfly damper is available with both. Adjustable from below it allows thorough balancing at the diffuser face. The No. 18 ADAPTER SQUARE, for use with either No. 15 or No. 16 diffuser, harmonizes nicely with acoustical ceilings solving the "round or square" problem with just one diffuser. Gasket prevents streaking. Designed for cooling as well as heating. 5 sizes.

GAS VENT PIPE AND FITTINGS

The H&C line of Type B gas vent pipe and fittings is complete. Provides everything necessary for venting gas water heaters and all types of gas-fired heating equipment.

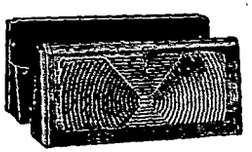
No. 401 DIFFUSAIRE®—For Perimeter Heating or Cooling.

Installed above the baseboard on outside walls, provides an excellent means of blanketing outside wall and window areas with a curtain of warm or cool air. This diffuser disperses air in full 180 degrees spread upward in addition to directing a small portion downward over floor. Adjustable stop on handle permits balancing system at the face. Sizes: 10 x 6, 12 x 6, 14 x 6.



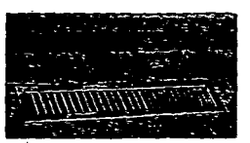
No. 405 DIFFUSAIRE®—For Perimeter Installations.

Particularly good for use in established homes. Out-of-wall Baseboard type. No stackhead required. Provides same air pattern as our No. 401 Sidewall DIFFUSAIRE.



No. 411 DIFFUSAIRE®—For Perimeter Heating or Cooling.

Floor type. Counteracts downdrafts from windows and cold walls. Opposed louver valves and set-screw adjustment permits accurate volume control for easy balancing of system. Design of louver blades provides even spread of air in fan shaped pattern with minimum resistance. Sizes 2¼ x 10, 12, 14; 4 x 10, 12, 14; 6 x 10, 12, 14.



No. 46 DIFFUSAIRE®—For Perimeter Systems.

Continuous baseboard type. Furnished in 2 ft and 4 ft units for installations of any length desired. Airfoil valve assures minimum resistance. This diffuser maintains air pattern regardless of volume, permits balancing system at diffuser face. Flexo-flange, furnished at no extra cost, provides a tight, quick connection between diffuser and boot. Matching RETURN AIR INTAKE is No. 47.



No. 15 CEILING DIFFUSAIRE®

This diffuser and its companion diffuser, No. 16, shown opposite, provide full 360 deg., draftless air distribution. The difference in the two is this: The rings of the No. 15 are flush, while in the No. 16 the rings step down at a 10 deg angle. Dependable for residential and small commercial installations. 5 popular sizes.



Hendrick Manufacturing Company

48 Dundaff Street, Carbondale, Pa.

Sales offices in principal cities—consult telephone directories

Hendrick Bulators; Hendrick Perforated Metal Grilles; Hendrick Mitco Open Steel Flooring, Armorgrids, Shur-Site Treads

HENDRICK BULATOR ....

THE *DUAL-UNIT* COMBINATION

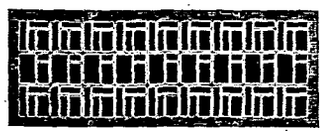
A Deflecting Vane Grille

+ An Ornamental Grille

Now you can secure in a single installation all the advantages of an adjustable vane grille—*to direct air flow*—and an ornamental grille—*to harmonize with the decorative scheme*—by specifying the new dual-unit Hendrick BULATOR*. It is a practicable combination that meets both the engineer's specifications for air throw and spread, and the decorative requirements of the architect.

In this dual-unit, the deflecting vanes are not noticeable, although they are mounted just behind the ornamental grille. The vanes are adjustable so that the air flow can be deflected to right or left, up or down, or in a combination of both directions.

For the imposed ornamental grille section of the dual-unit Bulator, Hendrick offers a wide selection of attractive designs, with the essential open area.



Photograph taken with deflecting vanes less than an inch behind grille, shows that vanes are not noticeable

Tested at Case Institute

To determine the efficiency of the dual-unit, tests were made at the Case Institute of Technology, Cleveland, under the direction of Professor G. L. Tuve, as a result of which, "it was found that at a given air volume, the presence or absence of the 'Mosaic' (design) grille made very little differences on either the air stream pattern or the throw."

A copy of the detailed report on these tests will be mailed on request.



Vertical deflecting vanes, showing how the vanes may be set to produce any desired air stream pattern

How to specify or order Hendrick Bulators

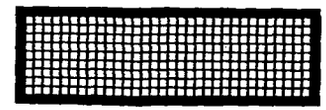
In specifying or requesting quotations on the Hendrick dual-unit Bulator, the following information is required:

1. The name or description of the desired ornamental grille, as given in the Hendrick Grilles catalog.
2. The metal, gage and finish of the ornamental grille, that is required.
3. The dimensions of the air duct opening.
4. The type of deflection desired in the deflecting vane section; whether right and left, up and down, or a combination of both.

* Beauty + Ventilator.

HENDRICK PERFORATED METAL GRILLES

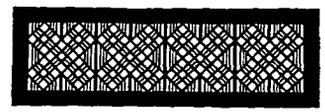
Hendrick decorative grilles are furnished in over a hundred patterns, and in a wide variety of overall dimensions, bar sizes, and number and size of perforations.



Argive—51 percent Open Area



Arglin—63 percent Open Area



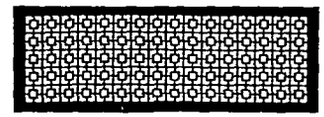
La Crosse—55 percent Open Area

Many exclusive Hendrick designs, originally produced to meet an architect's specifications for some particular project, are now available as standard numbers, and facilities for making special designs to specifications make the Hendrick service even more complete. The wide range of patterns permits the choice of a grille that will harmonize with any style of architectural design or period construction.

Grilles are fabricated in heavy-gage aluminum, bronze, copper, Monel, steel, stainless steel, and other commercially rolled metals. With ample open areas and accurate sizes, Hendrick grilles are characterized by clean-cut perforations, fine finish, and freedom from burrs and other imperfections.

They are easy to install, and always lie flat because of a special flattening operation in their manufacture.

If among the Hendrick special and standard designs you do not find something which meets your requirements, we shall be glad to submit suggestions or to work with you in the development of a special design.



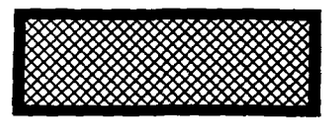
2" Link—52 percent Open Area

Air Conditioning Grilles and Registers

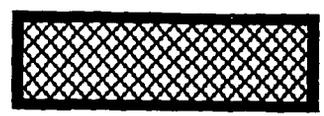
Hendrick air conditioning grilles and registers for directed air flow are made in various meshes, the standard having ½ in. or ⅝ in. opening between face bars. All types furnished with either horizontal or vertical directional bars either permanently set or so that they may be individually adjusted on the job to direct air flow to any desired degree. Grille and register faces are made regularly of steel with prime coat finish. They can be furnished in a lacquered finish to match any color sample.



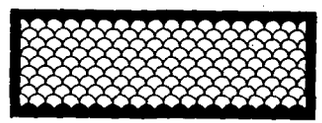
M-No. 2—57 percent Open Area



⅞" Diamond—60 percent Open Area



M-No. 9—67 percent Open Area



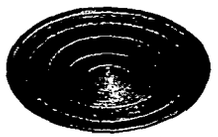
1¾" Shell—72 percent Open Area

Charles Demuth & Sons, Inc.

Mineola, N. Y.

Demuth Draftless Air Distributors

Representatives in Principal Cities



New-Type (D-3) available now

Designed and Patented by Demuth



Illustration showing curved equalizing deflecting air vanes and method of installation



Type (S)

The DEMUTH DRAFTLESS AIR DISTRIBUTOR

Consists of a series of curved vanes, mounted on a deflecting cone, to insure 360 deg distribution without the aid of secondary equalizing devices. A second hollow deflecting cone is arranged at a tangent to the primary cone, forming an injection nozzle.

The curved vanes discharge the supply air in turbulent streams, mixing room air with conditioned air, thereby providing constant air motion, uniform temperature and draftless distribution throughout the room.

Secondary air currents, which are created by the turbulence of the discharge air, are drawn into the diffuser by the bottom or secondary cone, mixed with the supply air, resulting in positive aspiration.

Type (SQ) square and rectangular Air Diffusers are made to any size with removable core assembly. Air distribution is from one, two, three or four sides of the diffuser. Made to any air pattern.

Accessories

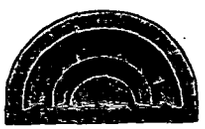
Combined lighting is available with the circular type. Volume and Splitter Dampers are available for all types.



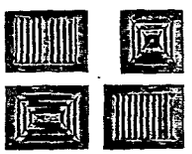
Type (F)



Type (M)



Type (SW)



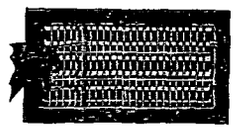
Type (SQ)

The Independent Register Company

3747 East 93rd Street • Cleveland 5, Ohio

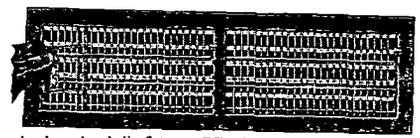
Independent "Fabrikated"® Registers and Grilles

For Sidewall

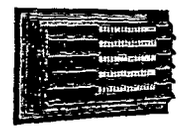


No. 321A-HMV

Every Grille Bar Adjustable Individually—Before or After Installation



No. 321A-HMV Tandem



Rear View Showing "HMV" Valves

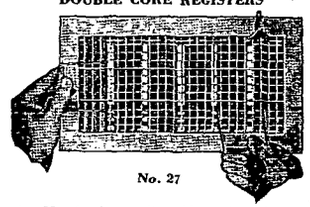
The valves swing from their edges and, when closed, lie flat against the flanges on the back of register face. With valves closed, register depth is 1¼ inches, except on the side, or end, of the operating lever, which is 1¾ inches. With valves open the complete depth is 2 inches.

If the valve length exceeds 24 inches, face and valves are made in tandem and valves are operated by two or more levers, as may be required.

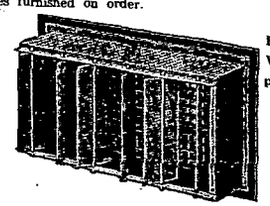
RIM WIDTHS: on sizes 12 inches vertical or less standard rim width is $1\frac{1}{16}$ inch all four sides beyond the stackhead or catalog sizes. Sizes over 12 inches vertical, $1\frac{5}{32}$ inches wide all four sides beyond wall opening sizes. Wider or narrower rims can be furnished at moderate additional cost.

These registers are available in 114 standard sizes to fit practically any opening from 8 in. x 6 in. to 48 in. x 48 in. Special sizes furnished on order.

INDEPENDENT ADJUSTABLE BAR DOUBLE CORE REGISTERS



No. 27



Rear View—Valves partly open

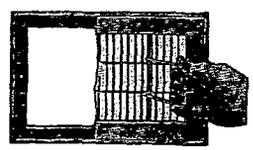
These units, No. 27, have adjustable vertical front bars and adjustable horizontal rear bars. Opposed blade, volume control dampers are key operated. Style No. 28 has adjustable horizontal front bars and adjustable vertical rear bars. Made to fit wall openings from 8 in. x 4 in. up to 48 in. x 24 in. in 90 different sizes.



No. 311D-WC—Bars Faced Down

These grilles are designed for commercial installations. Style No. 311-D-WG illustrated has grille bars set permanently at approximately 35 deg down. Other models available have grille bars set for straight flow.

"WT" WALL FRAME



Can be separately installed before plastering. Duct openings must be made approximately ⅛ in. oversize. Frames must be fitted and attached to grilles or registers at our factory. Made in sizes to accommodate sizes of No. 27 and No. 28 Double Core Registers.

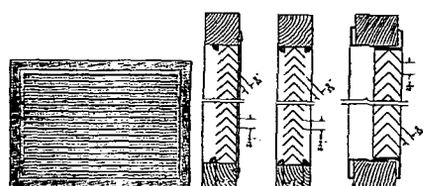
"Fabrikated" Floor Registers for Perimeter Heating



No. 60A

Faces are famous "Fabrikated" construction, excelling in rigidity, open area and fine appearance. Adjusting screw permits directed air flow toward outside wall when valves are set into proper position. Sizes comply with code requirements of *NWAH & ACA*. Fabrikated floor faces for return air are also available.

NO-VISION GRILLES, DOORS, WALLS AND PARTITIONS



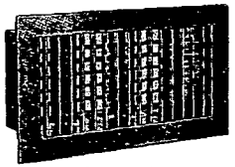
No. 1312R No. 1312R No. 1312C No. 1312RR

The grille bars are V-shaped; it is impossible to see through the grille from any viewpoint

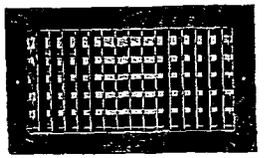
KRUEGER AIR CONDITIONING CORP.

19 East RILLITO **KRUEGER** TUCSON, ARIZONA

SUPPLY REGISTERS AND GRILLES



Series 880 O.B.D. Registers combine double deflection grilles and opposed blade damper units to give complete control of air in both planes plus equal distribution over entire face of register. Volume control valve is easily adjusted from face.



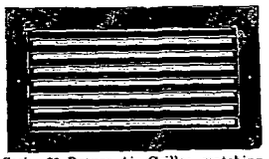
Series 800 Multi-Shutter Registers provide adjustable front fins for any desired deflection and rear louvers are gang operated. Models have either vertical or horizontal streamlined fins ⅝ in. deep set on ¾ in. centers.

MULTI-SHUTTER REGISTERS



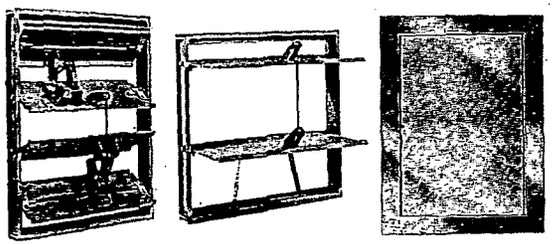
A fine low-cost register for combination heating and cooling. Vertical fins diffuse air in two directions. Adjustable multi-louver has positive shut-off control. Front fins are spaced ½ in. for greater free area.

RETURN AIR GRILLES



Series 80 Return Air Grilles, matching units for air supply registers, combine rugged construction and fine appearance. Fins spaced on ¾ in. centers are set at 35 deg angles. Available in horizontal and vertical models.

DAMPERS AND ACCESS DOORS



MIXING DAMPERS (left) Manufactured in Opposed-Blade, Multi-Louver, Face and By-pass, and Right Angle designs. Construction of heavy channel frame rigidly welded with brass pivots. Louver bearings are bronze sleeve type. Built for manual or motor operation.

FIRE DAMPERS (center) Manufactured to N.B.F.U. specifications. Construction of welded heavy channel with 12 gauge damper blades pivoted off-center to assure positive closing. Bronze sleeve bearings. Fusible Link furnished at 165 deg or specified temperature. Closing spring is standard part of damper assembly.

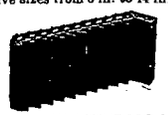
ACCESS DOORS (right) Manufactured to any desired size and offset edge or flat frame types. Construction of 16 or 18 gauge cold rolled. Continuous type hinge located on long dimension. Metal screw locks door to frame. Special locking devices available.



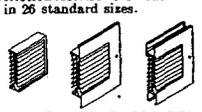
PERIMETER REGISTERS: Series 900 Baseboard Registers offer easy installation in both new and old construction. Register housing can be installed at time of "rough in." Grille facing and damper installed later. Features rubber-tipped lever operated damper and attractive appearance. 18, 24, 30 in. lengths.



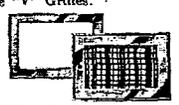
ROUND CEILING DIFFUSER has 360 deg air diffusion pattern. Friction controlled butterfly damper has full 90 deg control operated from center shaft. Step-down design does not require increase in duct size at neck for installation. Five sizes from 6 in. to 14 in. diam.



AIR & VOLUME CONTROLS designed to balance air volume and distribute flow equally over grille face. Curved blades, adjustable from full open to full closed, stabilize pressure and minimize friction loss. EX-8 Extractors available in 26 standard sizes.



NON-VISION DOOR GRILLES provide constant passage of air with complete obstruction to vision. "V" shaped fins are spaced on ½ in. centers for greater free area. Series 600 Non-Vision Grilles are available in three styles, 88 sizes. Also available are light-tight Double "V" Grilles.



PLASTER FRAMES provide a neater installation for registers and grilles by protecting plastered or panelled walls. **FILTER FRAMES** provide easy access to filter pads installed in Frame directly behind return air grilles.

THE PYLE-NATIONAL COMPANY

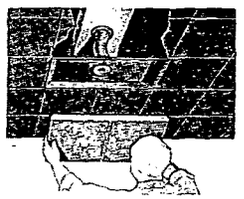
Multi-Vent Division
1363-78 North Kostner Avenue
Chicago 51, Illinois

Sales Engineers and Agents
In Principal Cities
U. S. and Canada

MULTI-VENT® AIR DIFFUSERS

Low velocity, draft free air diffusers that permit complete concealment in ceiling and complete freedom in placement regardless of the proximity of partitions (movable or stationary). Multi-Vent low velocity, air diffusing panels are available in various models, styles and sizes to suit all types of ceiling construction. The more popular, standard types are briefly described below.

In addition, Multi-Vent panels can be custom built in almost any shape or size, material or finish, to suit unusual needs. Your inquiries for additional information on the selection and application of Multi-Vent panels will be given every consideration by our experienced engineering staff. Our extensive laboratory testing facilities are also at your disposal.



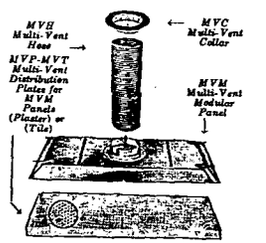
Modular Multi-Vent Panel in Standard Acoustical Metal Pan Ceiling.

TYPE MVM MODULAR MULTI-VENT PANELS

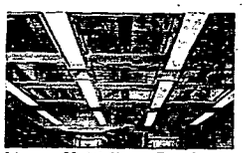
For acoustical metal pan ceilings.
For radiant pan ceilings.
For plaster and acoustical tile ceilings.
For exposed duct applications.

BASIC MULTI-VENT COMPONENTS

Modular Multi-Vent diffusers consist of four basic parts: Duct collar, tubing, panel, and perforated distribution plate. (The distribution plate is not required when standard perforated acoustical ceiling pans are used.)



Type MVM Multi-Vent Modular Panel



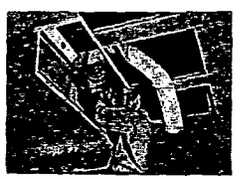
Linear Multi-Vent Panels in Acousti-Line Ceiling.

TYPE MVL MULTI-VENT PANELS ESPECIALLY SIZED AND ADAPTED FOR USE IN THE FOLLOWING CEILINGS:

Acousti-Line Mineral Tile Ceilings
Acousti-Luminous Ceilings
Alumi-Coustic Ceilings
Simplex Ceilings
Wakefield Ceilings



MVL Multi-Vent Linear Panel



Multi-Vent Troffer in acoustical Mineral Tile Ceiling.

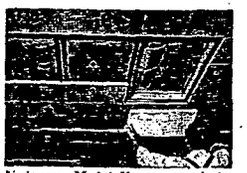
TYPE MVB MULTI-VENT TROFFER®

Patent Number 2,845,854 and 2,845,855

Combination Air Diffuser and Troffer Light.
Air diffuser by the Pyle-National Company, lighting by Benjamin Electric Manufacturing Company.
For use in any type of drop ceiling. Takes flexible hose and duct collar.



MVB Multi-Vent Troffer Valve

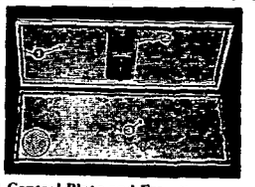


Unitary Multi-Vent panel in acoustical metal pan ceiling.

TYPE MVAC TYPE MVAR UNITARY MULTI-VENT PANELS*

For acoustical metal pan ceilings.
For plaster and acoustical tile ceilings.
For exposed duct applications.

* For special applications available with built-in secondary filters; built-in germicidal lamps; built-in radiant heating, radiant cooling coils.



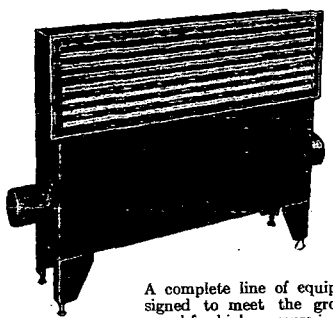
1-Control Plate and Frame.
2-Pressure Displacement Air Valve.
3-Perforated Plate (not required when standard acoustical ceiling pans are used).

TUTTLE & BAILEY

Engineered Products for Residential, Commercial and Institutional Air Conditioning, Heating, Ventilating

NEW BRITAIN, CONNECTICUT

HIGH PRESSURE AIR DISTRIBUTION UNITS



TYPE MPW

A complete line of equipment designed to meet the growing demand for high pressure installations in multi-story office buildings, schools, hospitals, hotels and institutions.

TYPE MPW

Medium Pressure Periphery Wall Mixing Plenum Unit for individual room control on heating or cooling cycle, or zone control. Handles static pressure in system up to 3.5 in. w.g.

TYPE MPD

Medium Pressure Ceiling Diffuser Double Duct Mixing Plenum Unit for individual unit or zone control. Single duct units also available. Handles static pressure in system up to 3.5 in. w.g.

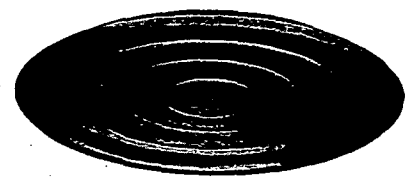
TYPE MP

Medium Pressure Sub-Central Station Unit for distribution of air to one or more diffusers by means of rigid or flexible ductwork. Handles static pressure in system up to 3.5 in. w.g.

TYPE PRV

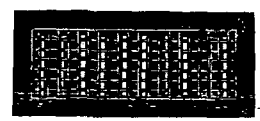
Pressure Reducing Valve for use in medium pressure systems, temperature mixing units and sound attenuation boxes. Manual or automatic operation.

AEROFUSE CEILING DIFFUSERS



A wide selection of diffuser types and sizes, styled and engineered to meet the most rigid requirements of appearance and performance at the point of air delivery.

TRI-FLEX SUPPLY AIR GRILLES AND REGISTERS



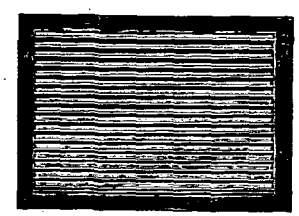
Available as single and double deflection grilles; single and double deflection registers with opposed blade dampers. Blades are regulated by means of a key operator and may be set in any position from fully open to fully closed. Face bars are individually adjustable. Stocked in 26 standard sizes.

26 STANDARD SIZES

8 x 4	14 x 4	20 x 6	30 x 6
10 x 4	14 x 5	20 x 8	30 x 8
10 x 6	14 x 6	24 x 5	30 x 10
12 x 4	16 x 5	24 x 6	30 x 12
12 x 5	16 x 6	24 x 8	36 x 8
12 x 6	20 x 5	24 x 10	36 x 10
		24 x 12	36 x 12

AEROVANE RETURN AIR GRILLES AND REGISTERS

Aerovane grilles are available with horizontal or vertical bars. Aerovane registers are furnished with the same opposed blade damper units as supplied with Tri-Flex registers. Stocked in 20 standard sizes.



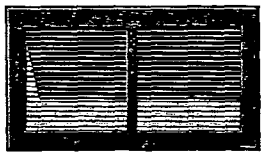
20 STANDARD SIZES

10 x 6	24 x 24
10 x 8	30 x 12
12 x 6	30 x 18
12 x 8	30 x 24
12 x 12	36 x 18
18 x 6	36 x 24
18 x 12	36 x 30
18 x 18	48 x 24
24 x 12	48 x 30
24 x 18	48 x 36

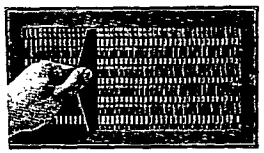
UNITED STATES REGISTER COMPANY

Battle Creek, Michigan, U.S.A.

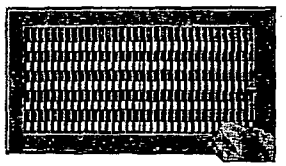
AIR CONDITIONING REGISTERS—GRILLES—DIFFUSERS—EXHAUST and BACK PRESSURE DAMPERS—ACCESSORIES



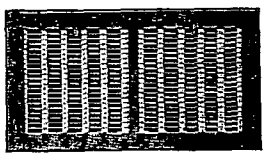
No. 153 U.S. Air Conditioning Single Valve Register, Lever-Operated. ¼ in. Bar Spacing 22 deg. affords Non-Vision. With Rubber Sealing Gaskets.



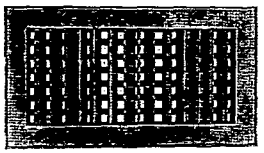
No. 256 U.S. Air Conditioning Multiple-Valve Register, Lever-Operated. ¼ deg. Bar Spacing 22 deg Right and 22 deg Left gives 4-Way Diffusion and Non-Vision.



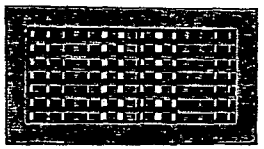
No. 249 U.S. Air Conditioning Multiple-Valve Register, Lever-Operated Valves. Vertical Grille Bars Completely Adjustable to 45 deg Right and Left Flow and Two-Way Flow.



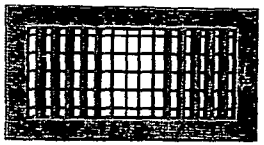
No. 153-VVL U.S. Air Conditioning Multiple-Valve Register. Horizontal ¼ in. Bar Design. Two-Group Lever Operated, Vertical Rear Valves which afford Right and Left Flow or Two-Way Flow.



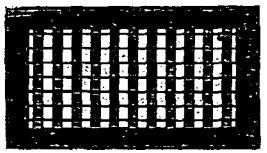
No. 190 U.S. Multi-Flex Register, Vertical Streamlined Front Grille Bars, Horizontal Rear Bars, Individually Adjustable, Screwdriver Operated Opposed Rear Valves.



No. 191 U.S. Multi-Flex Register, Horizontal Streamlined Front Grille Bars, Vertical Rear Bars Individually Adjustable. Screwdriver Operated Opposed Rear Valves.

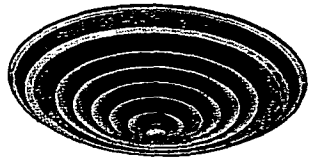


No. 192-L Multi-Flex Register, Vertical Streamlined Grille Bars Individually Adjustable. Lever-Operated Parallel Multiple Valves. No. 192 Same, but Screwdriver Operated Opposed Valves.



No. 201 Non-Flex Register, Return Air—Fixed Horizontal Bars. No. 200—Fixed Vertical Bars. For Multi-Flex Grilles and Non-Flex Return Air Fixed Bar Registers and Grilles, see Complete Catalog.

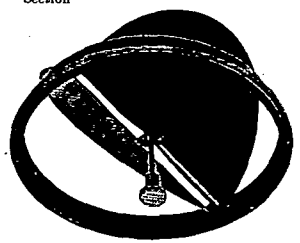
UNITED STATES REGISTER COMPANY—A-C REGISTERS—GRILLES—DIFFUSERS—DAMPERS—ACCESSORIES



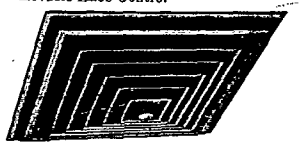
No. 1500 ROUND STEP-DOWN CEILING DIFFUSER. Also made in No. 1501 FLUSH-TYPE



No. 1500 Round Ceiling Diffuser Cross Section



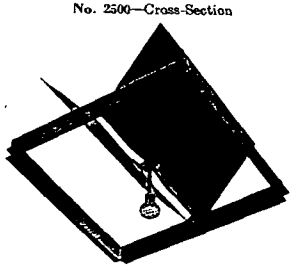
No. 1800 Round Ceiling Damper, Removable Knob Control



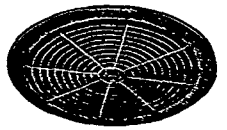
No. 2500 SQUARE STEP-DOWN CEILING DIFFUSER. Also made in No. 2501 FLUSH-TYPE



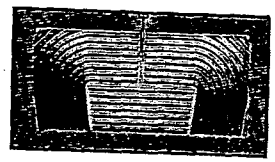
No. 2500—Cross-Section



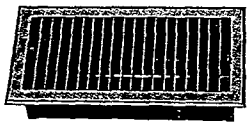
No. 2800 Square Ceiling Damper, Removable Knob Control



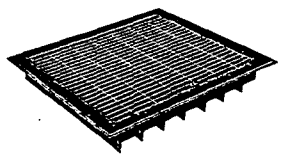
No. 500 SERIES—U.S. ROUND CEILING DIFFUSER will be continued. Fine for Sound Outlets—also Residential



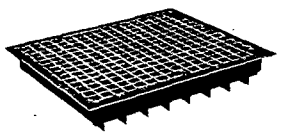
No. 105 U.S. DIFFUSER A-C REGISTER. 180 deg Perimeter Diffusion. Three Sizes



No. 410 DIFFUSER FLOOR REGISTER for Perimeter Systems. Made in Nine Sizes



No. 400 U.S. Trussteel Floor Register, Grid-Bar Face 2 x ½ in. O.C. spacing of Bars.



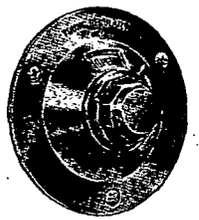
No. 600 U.S. SANI-STEEL REGISTER, $15\frac{5}{8}$ in. Square Mesh $\frac{5}{16}$ in. Bars. Valve and body construction same as No. 400; Lever Operation No. 600. Key Lock Operation specify No. 625. Key-Operation specify No. 620. Pole-Operation specify No. 650.

YOUNG REGULATOR COMPANY

20910 MILES AVENUE, CLEVELAND 28, OHIO

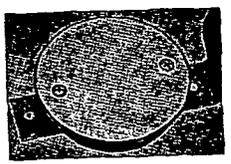
SURFACE - CONCEALED - SPLITTER - SQUEEZE - REMOTE CONTROL DAMPER REGULATORS

Young Regulator Company offers a complete line of manually controlled damper regulators for every type of heating, ventilating and air conditioning installation in office buildings, factories, hospitals, ships, schools, churches and residences. Young Regulators are used by many leading companies in nearly every industry and are specified by leading architects and engineers.



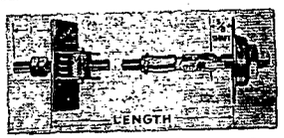
No. 1 Surface Mounted, Direct Connection Regulator

The original Young Damper Regulator, of which thousands have given satisfactory service for years, has been modernized and streamlined. It is used to regulate a damper, controlling the air flow through a duct. It is readily set and locked into position by a key.



No. 301 Concealed Damper Regulator

This regulator is installed in ceilings or walls. The box is embedded in the finished ceiling or wall material so that the flange is flush with the finished surface. Cover plate is 3¾ in. in diameter and covers the joint between the box and plaster. Operation is by key or a socket wrench that fits the end of the damper rod.



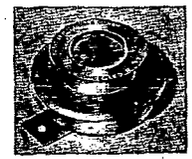
No. 900 Series Air-Split Damper Regulators

No. 900, illustrated above, is one of six Young Regulators used for operating splitter or squeese dampers. The operating heads may be placed in any accessible location. Operation is by crank.



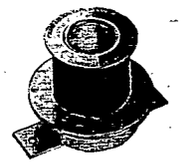
No. 700CP Remote Control Damper Regulator

This regulator is for manual operation of dampers and registers to give individual room temperature by remote control. With No. 704 Corner Pulleys, Remote Control Regulators operate Young Dampers at a distance of 250 feet or more. Two registers or dampers can be operated by a single unit.



No. 315 Adjustable Cover Concealed Regulator

This regulator is for use in acoustical ceilings. It is similar to the No. 301 Concealed Type, except the cover plate is adjustable for height by screwing down the cover plate in order to meet the various thicknesses of acoustical or plastered ceilings. The adjustment or locking of the regulator is by a key or socket wrench.



No. 330 Floor Line Adjustable Cover Regulator

This new Young Regulator is set flush with the floor in a branch duct near a grille and is used for perimeter heating and air conditioning installations. It gives accurate volume control of air in branch ducts and makes possible the use of grilles in place of expensive registers and eliminates the necessity of separate ducts running to the heating and air conditioning room.

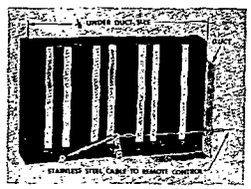
FOR COMPLETE DETAILS SEE OUR CATALOG IN SWEET'S ARCHITECTURAL FILE 30F/YO

YOUNG REGULATOR COMPANY

20910 MILES AVENUE, CLEVELAND 28, OHIO

DAMPERS - REGISTERS - GRILLES

Young Regulator Company offers a complete line of dampers, registers and grilles for air conditioning and heating installations.

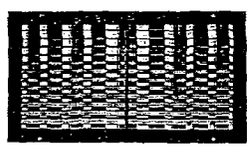


No. 830 Opposed Blade Damper

No. 830 Opposed Blade Damper distributes the air evenly over the entire face of the grille. It is easy to install in the duct behind the grille and is operated by a Remote Control Regulator placed in a convenient location. Tubing to the Remote Control Regulator is attached to the duct by a Compression Fitting.

No. 830 Damper is furnished as part of No. 840 Series Air Conditioning Register or separately.

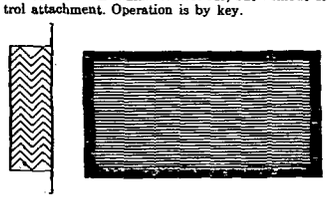
No. 831 is the same as No. 830, but without the Remote Control attachment. Operation is by key.



No. 840 Series Adjustable Bar Registers

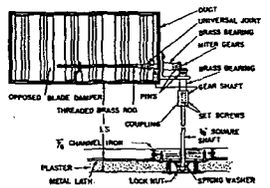
Young No. 840 Series Registers combine the various No. 800 Series Young Grilles and a vertically Opposed Blade Damper. These registers give individual room control when operated by No. 700 Remote Control.

No. 841 Series is similar to No. 840, but without Remote Control attachment. Operation is by key.



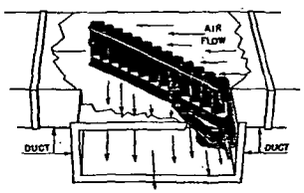
No. 875 Series, No Sight, No Light Grilles

These grilles are for dark room installations or where light must be excluded but where ventilation is required. Finish is of non-specular black, assuring no light reflection.



No. 817A Volume Control Damper

No. 817A is an Opposed Blade Damper similar to No. 830. It is installed in a branch duct to control the volume of air and is operated from a suspended ceiling by No. 301 Regulator. The unit is operated by a crank, and can be locked in position by key.



No. 890 Air Extractor

No. 890 Air Extractor has blades that operate in unison and are controlled by a key through the grille. The adjustable extractor scoops the required amount of air from the main duct into the branch duct.



No. 882 Perimeter Baseboard Diffuser

This Young Diffuser provides proper distribution of warm or cool air to all parts of the room and is engineered to induce continuous aspiration of air throughout the room. Using connectors furnished, the lengths can be combined to furnish continuous runs to fit any length of wall.

No. 800 Series Air Conditioning Grilles

Grille bars are made of extruded aluminum secured in a frame and dividing mullions. They must be adjusted with a hand tool. No. 800-VH has double banks of blades, one vertical, the other horizontal, both banks adjustable.

No. 845 Series Return Air Grilles

These grilles, for intake, have been designed to match the No. 800 Series for supply.

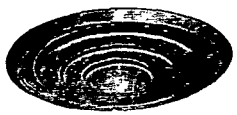
FOR COMPLETE DETAILS SEE OUR CATALOG IN SWEET'S ARCHITECTURAL FILE 30F/YO

UNIVERSAL DIFFUSER CORP.

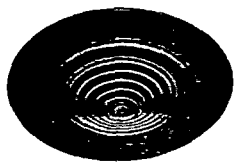
38 Marbledale Road, Tuckahoe, N. Y.
AIR DIFFUSERS · REGISTERS · GRILLES

ADJUSTABLE DIFFUSERS

Provide an infinitely variable means of controlling air volume. Control mechanism moves all blades up or down simultaneously.

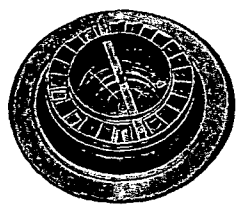


TYPE QP—A circular 4-cone adjustable pattern diffuser. Rotating lower blade moves core up or down, changing air delivery pattern from horizontal through any angle to vertically down; for cooling, ventilating or heating.



TYPE R—Circular constant pattern with volume control; air travels parallel to ceiling; for cooling purposes.

TYPE V—Circular variable diffusion pattern from horizontal to vertical with simple adjustment; for heating, ventilating, cooling or for high ceilings.



TYPE CH—Circular supply and return diffuser with both "R" and "V" characteristics. Eliminates return grilles.



TYPE H—Square diffusers with round necks, made in 3 models:

HAR—constant pattern with control of air volume through cone adjustment.

HAV—variable pattern by moving cones up and down.

H-F—fixed pattern with 4 different face patterns.



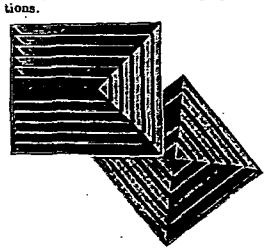
LINE-AIR—Long slot diffuser of any desired length for side wall and ceiling: Counter-current vanes provide means to deliver air at 90° to wall face without the use of auxiliary straightening vanes.

NON-ADJUSTABLE

Made with fixed blades; volume control is obtained by the use of face operated volume dampers.

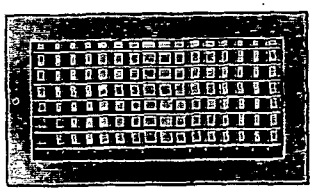


TYPE B—Circular, non-adjustable diffuser for cooling and heating; attractively styled for small commercial, industrial and residential ceiling applications.



TYPE MP—Rectangular and square, multi-pattern diffuser: 1-way, 2-way, 3-way and 4-way; made in 18 styles and 64 sizes for any type and shape of interior. Air diffusion pattern is determined by the style, without use of baffles.

UNIVERSAL GRILLES AND REGISTERS



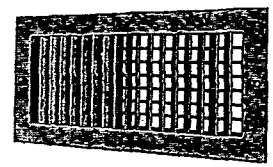
Available in steel and aluminum; stocked in 20 standard sizes

1—Double Deflection Grilles
2—Double Deflection Registers
3—Opposed Blade Dampers
4—Return Grilles and Registers
5—Door and Partition Grilles
6—Air Extractors
7—"Grilline" Long Slot Grilles
8—Volume Controllers
9—Plaster Frames

WATERLOO REGISTER COMPANY, INC.

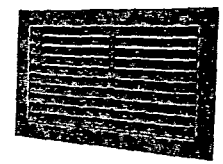
"Since 1902"
P. O. Box 72 Waterloo, Iowa

WATERLOO offers a complete line of Air Diffusion Equipment designed particularly for commercial and industrial heating, ventilating and air conditioning



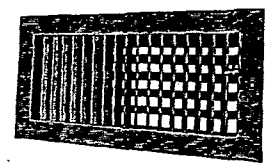
2V Supply Grille

Two sets of extruded aluminum louvers, each individually adjustable for four-way deflection at degree desired.



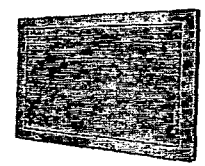
3H Return Air Grille

Fixed fins at 45 degrees of 0 degree deflection, in heavy gage frame, strengthened with adequately spaced support bars.



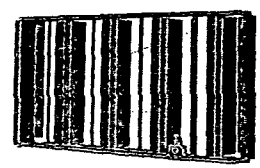
1VMH Supply Register

One set of adjustable louvers for easy control of air, right or left, with multi-shutter damper, horizontal blades. Ideally suited for residential installations because of narrow stack depth.



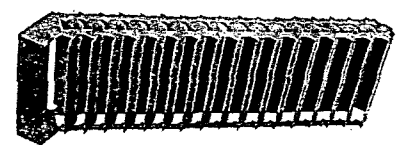
DV Door Ventilator

Inverted V-Shaped fins are permanently affixed to channel frame providing rigidity and complete sight proofing. Provides 72 percent free area. Available flange both sides, one side or channel frame only.



OB Damper

Opposed Blade Volume Controller may be purchased as a separate unit or can be incorporated as integral part of all Waterloo grilles.



DUCTROL Air Volume Extractor

Volume controller and extractor adjusts from full open to full closed, to provide even distribution to entire grille surface. Completely assembled at the factory—ready to install on job.



Airline Grille

This product of all extruded aluminum construction features solid ¼ in. face bars on ½ in. spacing. Available with or without dampers. Ideally suited for wall, sill or floor applications.

The complete Waterloo catalog is available upon request containing illustrations and engineering data on all products.

DURO DYNE Corporation

Route 110, Farmingdale, New York

Manufacturers of Specialties for the Sheet Metal, Heating, Air-Conditioning and Ventilating Fields

NEVA BIND DURO-BLADE KIT

Pat. No. 2,863,374

Multi-Blade Louver Damper Hardware

The Neva-Bind Duro-Blade Kit (multi-blade louver damper hardware) assures absolutely smooth damper operation, despite the fact that little care is required in assembly, with no alignment necessary after assembly.

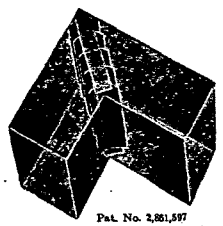
The kit's unique 4-way adjustability automatically compensates for slight errors in blade dimensions or formation; minor misplacement of bearing holes, or other possible inaccuracies which cause bending stress of connecting rod—the main cause of binding.

The Neva-Bind feature allows the rod to remain straight regardless of any of these inaccuracies of damper construction.

In addition, a damper made with the Neva-Bind Duro-Blade Kit can be fully closed without the bracket hitting the duct. The damper can be fully opened without interference from the connecting rod.



VANE RAIL* FOR MAKING AIR TURNING VANES



Pat. No. 2,861,597

The savings in installed costs of air turning vanes run from 50 percent to 70 percent when vanes are assembled with Duro-Dyne Vane-Rail.

Vanes are sheared from simple scrap. Tab cutting on vanes is completely eliminated.

Vanes are simply inserted into concaved notches of Vane Rail. To lock assembly, place chisel (supplied with each bundle of Duro Vane Rail) on portion of vane protruding through notch . . . one sharp blow on chisel securely locks vane to rail.

P-K* REGULATORS AND QUADRANTS



A—UNXLD* DAMPER QUADRANT SET—⅜ in. size used for dampers up to 20 in.; ½ in. size for dampers up to 30 in. Provides positive control. Sturdily constructed, easily installed and adjusted.

B—HEAVY DUTY QUADRANTS—Regulate larger size dampers up to 60 in. The ball bearing quadrants are fine manually operated heavy duty units. The plain bearing quadrant for larger regulators up to 52 in. has the same rugged construction except for the ball bearing feature. Size ⅜ in. to 1 in.

C—JIFFY* REGULATOR SET—Installs more quickly since it does not have to be attached to the duct.

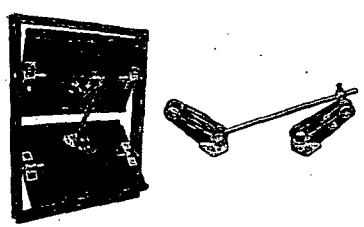
D—DIAL DAMPER REGULATION SET—¼ in. size for dampers up to 10 in.; ⅜ in. size for dampers up to 20 in. Special construction prevents air leakage and damper rattle. Made of heavy gage steel, cadmium plated.

"OPAX"* OPPOSED ACTION DAMPER LINKAGE SET

Opposed action dampers are preferred by engineers to the usual parallel assembly because turbulence is minimized and control of air more linear. "OPAX" Linkage Set includes two brackets and pre-assembled linkage rod which will provide efficient opposed action of damper blades 5 inches and wider.

The 4 way self adjustability automatically compensates for construction inaccuracies to assure smooth bind-free operation. Rugged construction permits use of the "OPAX" Linkage Set for the largest commercial and industrial damper installations.

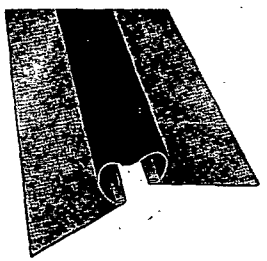
Duro-Dyne's "OPAX" Linkage Sets are boxed in complete units with installation instructions supplied.



* Trade Mark

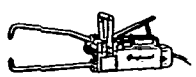
FLEXIBLE DUCT CONNECTOR PRODUCTS

Coils of Pre-assembled Metal-to-Fabric-To-Metal for Eliminating Vibration Travel

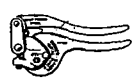


*DURO-METAL-FAB** ECON-O-FAB* and "JUNIOR"* are pre-assembled rolls of metal and fabric. They eliminate the annoying, expensive, time-consuming job of attaching metal to fabric in the shop. Featuring an unusual "Double-Loc Seam," the metal cannot pull away from the fabric when the finished connector vibrates; nor can the metal cut the fabric. For industrial applications or heavy gage duct work use DURO-METAL-FAB. Consists of 24 or 26 gage galvanized iron attached to *UL* Approved Canvas, Asbestos or Neoprened Fiberglas. Dimensions: Metal 4 in.; Fabric 3 in.; Metal 3 in. For residential and lighter gage duct work, use ECON-O-FAB. Identical to DURO-METAL-FAB except for lighter gage of metal. For lowest cost residential application, "JUNIOR" flexible duct connector is available in narrower dimensions: Metal 2 in., Fabric 1¾ in., Metal 2 in. All three sizes are packed in 50 ft or 100 ft coils in handy Dispens-o-Flat Cartons.

Special DUCT FABRIC* is available where fabric is required without metal. Packed in 100 ft rolls. Canvas comes in 6 in. and 10 in. widths; asbestos and neoprene 6 in. wide.



Duro-Dyne Portable Spot Welder



"OX" Metal Punch

DURO-DYNE INSULATION ADHESIVE IA-22*



A specific use adhesive designed for a quick, strong, permanent bond of insulation to sheet metal.

Created for bonding insulation to metal; insulation to insulation (at joints); sealing joints; "Buttering" insulation edges. Designed for glass-fiber, asbestos, cork insulation, etc.

Tack Time is only 10 seconds. Bond is immediate and positive. Long "open time" range permits bond one hour after application of adhesive. Permits rapid coating of duct surfaces and provides positive bond of heavy insulating materials without auxiliary holding devices.

Brake-forming of metal and insulation is possible immediately after bonding without separation of insulation from metal.

Coverage of Duro-Dyne Adhesive IA-22:
Brush Application: 230 + sq ft per gallon
Roller Application: 270 + sq ft per gallon
Temperature Range:
−30 F to 325 + F (Blanket)
−30 F to 250 + F (Board)

Duro-Dyne Adhesive IA-22 is waterproof, clear in color, solvent in petroleum-naphtha or white gas. Requires no thinning. Packing: 5 gal and 1 gal cans.

DURO-DYNE DUCT TAPE*



For use in sealing air duct installations and as vapor barrier sealer on insulation.

Two types available—cloth and reinforced fiber. Both have excellent characteristics. High strength, waterproof, temperature range. Aluminum color—popular widths available.

HIGH VELOCITY DAMPER REGULATOR AND QUADRANT SETS

A complete line of damper controls designed specifically to meet the requirements of increased velocities and pressures found in modern air distribution systems.

Damper rattle is virtually eliminated by the positive locking device between the damper shaft and the regulator or quadrant. A new close tolerance tail piece bushing is used to support the damper in smooth rattle-free operation.

Air leakage is overcome by the use of sealing washers clamped into position by locking nuts at control end as well as tail end of damper.

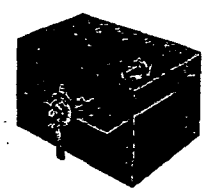
Hi-V Dial Regulator Sets are available in two sizes:
¼ in. for small dampers—HV-145
⅜ in. for larger dampers—HV-195

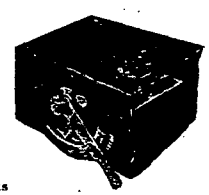
Hi-V Quadrant Sets are available in the ⅜ in. size:
⅜ in. for heavier applications—HV-385

More detailed information is available on request

* Trade Mark

Hi-V Dial Regulator





Hi-V Quadrant Sets

UNITED SHEET METAL CO., INC.

540 South Drexel Avenue **Columbus 9, Ohio**

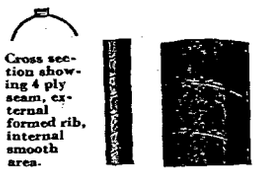
Representatives in Principal Cities of United States and Canada

Specialists in Low Loss Components For High Velocity Systems

Manufacturers of Prefabricated Metal Spiral "Lockseam" Duct, Die-Stamped, Two-Piece "Laminar-Flo" 90 deg Elbows, Matched Low Loss Fittings, and Sound Absorbers—For High Velocity Systems

SPIRAL "Lockseam" Duct

For High Velocity Systems. Furnished in any lengths up to 20 ft as continuous duct. 12 ft lengths are standard.



Cross section showing 4 ply seam, external formed rib, internal smooth area.

REDUCES FRICTION LOSS

At 4000 fpm the friction loss averages of spiral duct as compared with clean, round, galvanized, metal duct having approximately 40 joints per 100 feet are as follows:

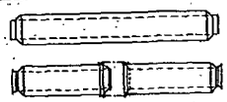
DIAM.	SPIRAL DUCT FRICTION LOSS AVERAGES
20"	5% less
12"	8% less
3"	15% less

Specifications of Spiral "Lockseam" Duct

DIAMETER NOMINAL SIZE	EXACT I.D.	WEIGHT LBS. PER FT.	METAL GAGE
3"	3.084	.532	26
4"	4.015	1.056	26
5"	4.993	1.353	26
6"	6.008	1.585	26
7"	6.983	1.534	24
8"	8.077	2.059	24
9"	9.052	2.059	24
10"	10.037	2.293	24
12"	12.038	2.533	24
14"	14.025	4.875	24
16"	16.023	5.251	24
18"	18.044	5.946	24
20"	20.073	6.563	24
24"	23.642	7.319	24

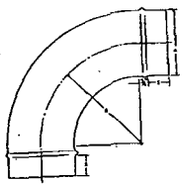
SOUND ABSORBERS

Sound absorbers for high velocity systems. Eliminate high and low frequency sounds.



Two-Piece "LAMINAR-FLO" 90 deg Elbow

Die-Stamped Type E-1



Specifications of "Laminar-Flo" 90 deg Elbow, Type E-1

NOMINAL SIZES "A"	ACTUAL O.D.	ACTUAL I.D.	"R"	"S"	WEIGHT POUNDS
3"	3.944	3.572	4½	2	1.25
4"	3.979	3.908	6	2	2.15
5"	4.954	4.833	7½	2	3.07
6"	5.948	5.876	9	2	3.90
7"	6.943	6.871	10½	2	5.00
8"	8.037	7.965	12	2	6.36

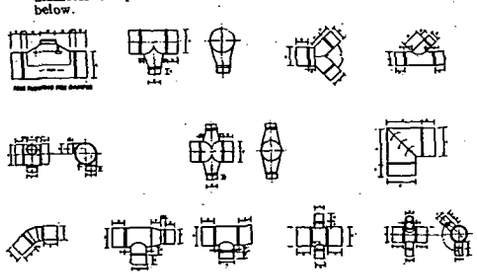
Available in diameters of 3", 4", 5", 6", 7", 8" 20 ga. galvanized steel, continuous weld, longitudinal seams. Available in aluminum and other specified alloys.

Static Pressure Loss of "Laminar-Flo" 90 deg Elbow, Type E-1
Inches of water at 0.075 lb/cu ft Density
"Zero Length" Loss. Add to duct friction calculated from intersection of duct centerlines

DUCT VELOCITY FT/MIN	NOMINAL DIAMETER 3"	4"	5"	6"	7"	8"	DUCT VELOCITY FT/MIN	NOMINAL DIAMETER 3"	4"	5"	6"	7"	8"
600	.009	.006	.005	.004	.004	.003	3400	.218	.157	.123	.100	.085	.075
800	.015	.011	.008	.007	.006	.005	3600	.238	.175	.137	.113	.093	.083
1000	.022	.016	.013	.011	.009	.008	3800	.253	.191	.150	.124	.106	.091
1200	.031	.023	.018	.015	.012	.011	4000	.290	.210	.165	.136	.116	.100
1400	.041	.030	.024	.020	.017	.015	4200	.326	.231	.180	.150	.127	.110
1600	.053	.039	.030	.025	.021	.019	4400	.343	.252	.198	.163	.138	.120
1800	.066	.048	.038	.031	.027	.023	4600	.374	.273	.213	.177	.150	.131
2000	.081	.058	.046	.038	.033	.028	4800	.400	.295	.231	.190	.161	.141
2200	.098	.070	.055	.045	.038	.034	5000	.434	.320	.250	.205	.175	.153
2400	.113	.082	.064	.053	.045	.039	5200	.468	.343	.268	.218	.187	.163
2600	.131	.093	.075	.061	.052	.045	5400	.500	.368	.287	.235	.200	.175
2800	.150	.110	.085	.070	.060	.052	5600	.535	.393	.307	.258	.215	.188
3000	.170	.125	.097	.080	.068	.059	5800	.570	.420	.328	.270	.230	.200
3200	.192	.140	.110	.090	.078	.066	6000	.610	.450	.350	.290	.245	.215

MATCHED LOW LOSS FITTINGS

20 ga. galvanized steel. Continuous weld. Available for every diameter of spiral duct. Portion of types available shown below.



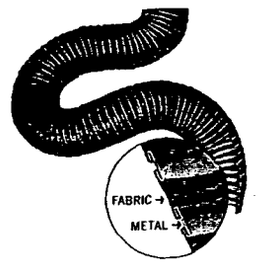
The Wiremold Company

Hartford 10, Connecticut

WIREMOLD®
FLEXIBLE AIR DUCT

Sales representatives in principal cities throughout the United States and Canada

FLEXIBILITY MAKES AIR DUCT VERSATILE



Wiremold Flexible Air Duct provides an efficient, economical means for conducting hot or cold air, gases, fumes, dust, or air-borne particles at high or low velocity. Completely different from any other duct, Wiremold is made by a process that *mechanically locks* the fabric and the supporting metal spiral. There is no glue to dry out and fail with age.

Its principal advantages are these: makes any bend easily, without kinking ... dampens mechanical vibrations ... saves hours of drafting time, because there is no need to work out exact positioning details on installation layouts.... eliminates the need for expensive, special elbows ... one man can easily install even the largest size, without needing any special tools ... no air leakage within recommended operating pressures ... minimum friction loss.

Wiremold Air Duct is made in many types and sizes to meet varying requirements and provide the longest possible service life. Write for full details.

FOR AIR CONDITIONING...

After three years of development, including over 25,000 hours of accelerated aging tests, Wiremold offers three types of air duct specifically for air conditioning applications. Manufacturers of air distribution equipment recommend the use of such flexible connectors on the high pressure side of the attenuation box, as well as for runouts to air distribution units. Because the duct is flexible, diffusers can easily be relocated to meet the changing needs of tenants.

5 major uses in air-conditioning systems



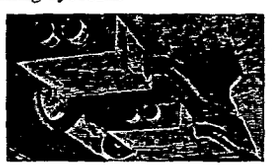
1—To connect underwindow units with vertical risers.



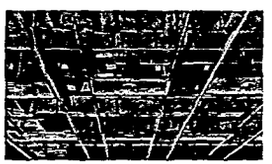
2—To connect branch duct to single ceiling outlet. The duct and diffuser can easily be relocated at any time.



3—From attenuation box to outlets in modular ceiling.



4—Flexible connection from supply duct to attenuation box, on the high pressure side.

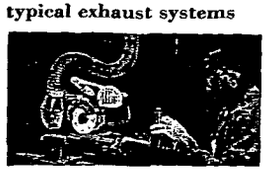


5—Vertical takeoff from branch runout to ceiling pan-type outlet. Note compensation for misalignment.

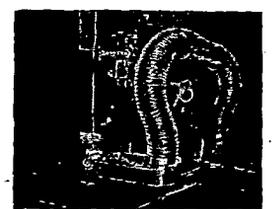
FOR VENTILATING...

Two types of flexible ducts—one to handle dust and abrasive particles, the other to handle corrosive fumes and gases—are available for industrial exhaust systems. They can be hooked up to central or portable collection systems without costly duct work, elbows, fittings and connections.

typical exhaust systems



On surface grinding machine, duct moves back and forth with grinding wheel, to exhaust with maximum efficiency.



Glass dust and particles are removed from glass cutting machine. Height and length of cut are easy to adjust because of flexible duct.

ALCO VALVE CO.

851 Kingsland Avenue, St. Louis 5, Mo.

THERMOSTATIC EXPANSION VALVES . . . CONSTANT PRESSURE (AUTOMATIC) EXPANSION VALVES . . . REFRIGERANT DISTRIBUTORS . . . SOLENOID VALVES . . . SUCTION LINE REGULATORS . . . FLOODED EVAPORATOR CONTROLS . . . REVERSING VALVES . . . LIQUID AND SUCTION LINE STRAINERS.

SEE YOUR ALCO WHOLESALER

For capacities in excess of those listed, write us and give specific requirements

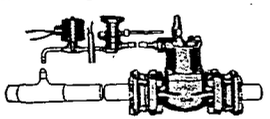
THE COMPLETE LINE OF REFRIGERANT CONTROLS

THERMO® EXPANSION VALVES

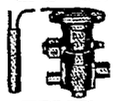
For automatic control of liquid refrigerant on all types of refrigeration and air conditioning systems. Capacities: From fractional tonnage to 200 tons Refrigerant 12 and "Freon-22"; selective charges. Low temperature valves for −40F to −100F.



TYPE 402 with pressure limiting feature



TYPE PO Pilot operated—maintains sensitive control over wide ranges in load



TYPE TCL With either angle or straight-thru connections



TYPE TR Multi-Outlet

SOLENOID VALVES

For all types of service. Liquid, up to 75 tons Refrigerant 12, 110 tons Refrigerant 22. Suction, up to 10 tons Refrigerant 12, 15 tons Refrigerant 22. For discharge gas, brine, water, steam, air or oil service.



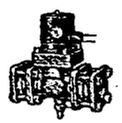
TYPE S1



TYPE S2



TYPE M3



TYPE R2

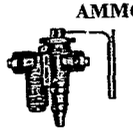
AMMONIA CONTROLS



TYPE M91F



TYPE TG



TYPE UG



TYPE TX



TYPE E with strainer

Solenoid Liquid Valves—up to 172 tons. **Solenoid Suction Valves**—up to 28 ton. **Thermo® Expansion Valves**—from fractional tonnage to 125 ton. **Automatic Expansion Valves**—from fractional tonnage to 60 ton.









VENTURI-FLO DISTRIBUTORS

Wide application range. One piece; replace distributors requiring as many as 6 or 7 nozzles.

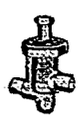
FLOAT VALVES AND SWITCHES

TYPE J56 electric floatswitch. For Refrigerant 12, 22, Methyl Chloride, Ammonia and other non-corrosive liquids having a specific gravity of 0.6 or more. Up to 230 Volts, A.C or D.C.

TYPE HK HIGH PRESSURE FLOAT VALVE. Up to 5 ton Refrigerant 12, 3 ton Refrigerant 22 and 20 ton Ammonia.

908 LECTRO-LEVEL electronic remote control of liquid level. Accurate control for full-flooded evaporators. Adjustable to a wide range of level changes. Easy-to-set control dials conveniently mounted in remote box.

SUCTION LINE CONTROLS



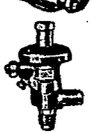
TYPE 771–772 SUCTION PRESSURE REGULATORS

(Hold-Back Valves) Prevent motor overload. Refrigerant 12 and 22.



TYPE EPR EVAPORATOR PRESSURE REGULATOR

For all refrigerants, with connection sizes up to 6 in.



TYPE 760 "EVAPOTROL"

Pressure regulator—½ ton, Refrigerant 12—¼ ton, Methyl Chloride.

BARBER-COLMAN COMPANY

1300 Rock Street, Rockford, Illinois

Field Offices in 95 Cities Throughout the United States and Canada—consult local phone book

A Complete Line of Electric and Electronic Controls

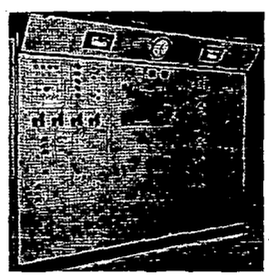
Barber-Colman manufactures a complete *line of automatic controls* and control systems for heating, ventilating, and air conditioning applications for all types of buildings, large or small. Typical product lines are pictured on this page.

A nationwide sales and service organization offers experienced engineering assistance, qualified supervision of installation, and prompt, competent field maintenance assistance.

Only Barber-Colman offers you years of combined experience in both automatic *controls* and air distribution and assumes undivided responsibility for delivering desirable indoor weather. (See Barber-Colman page in the Air System Equipment Section.)

Product literature and application diagrams available. Please state your specific interest.

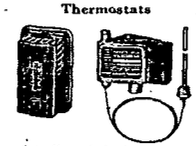
Complete "Electronic" Control Centers



The "Electronic" control center provides visual supervision of the entire heating and air conditioning systems and other mechanical or electrical functions, provides automatic control, and reduces installation and operation costs. It includes remote temperature indication, adjustment, and recording. Schematically diagramed, the control center provides "visual" supervision of the entire *building*.

Ask for new booklet, "Electronic" Control Centers.

Thermostats

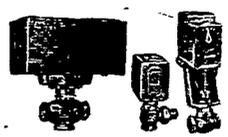


A complete line of single- and two-stage room and remote bulb thermostats for on-off control or proportional control.

Humidity Controls

Hair element or electronic type for two-position (on-off) or proportional control of motor-operated valves, solenoid valves, compressors, relays, etc.

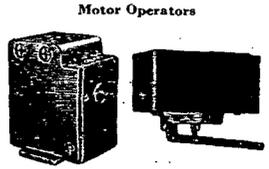
Motor-Operated Valves



Single- or double-seated valves for on-off or modulating control of steam, air, and water, and three-way valves for bypass or mixing applications.

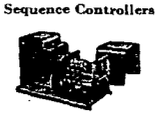
A complete line of compact motor operators is available for two-position and proportional control.

Motor Operators



Spring return, unidirectional, reversible, multi-position, and proportional-type motor operators are available for operation of all types of dampers. Heavy-duty models are oil-submerged type. Adjustable speed is optional.

Sequence Controllers



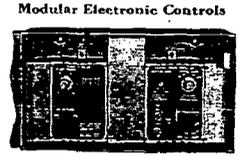
Sequence controllers, consisting of a reversible oil-submerged driving motor to operate from two to six snap-acting switches, are available for multi-step control of compressors, pumps, valves, or relays in refrigeration, air conditioning, or industrial processing applications.

Unit Ventilator Controls



A complete line of controls for all types of unit ventilators, including face and bypass, valve-controlled, electrically heated, and direct gas-fired classroom or auditorium types. "Package" sets are featured, designed for mounting entirely within the unit ventilator enclosure. These sets utilize a dual-element thermostat combined with a motor operator for extremely accurate control of room and discharge air temperatures without the need for wall-mounted room thermostats.

Modular Electronic Controls



These controls translate temperature or humidity changes into voltage outputs *which* are amplified to operate relays. *The relays, in turn, operate valves,* motor operators, or other control *devices.*

Complete custom control panels to meet any requirement are built up of one or more standard bridge circuit modules and one or more standard amplifier modules.

Temperature bridges provide for use of one, two, or three standard sensing elements. Auxiliary bridge circuit modules are available to operate as many as three amplifiers in sequence.

Either two-position or proportional amplifiers are available.

Dual compensator units, used with an amplifier module or combination of amplifier and bridge components, provide control for such applications as: outdoor reset of main element, stabilization of discharge temperature, differential thermostat action, and lag element control.

Accessories

The Barber-Colman accessory line includes power boxes, *transformers,* color-coded thermostat cable, relays, switches, electric time switches, manual rheostats, polarized microrelays, and *other items* which go to make up a complete control system.

GENERAL CONTROLS

MANUFACTURERS OF AUTOMATIC CONTROLS FOR HOME, INDUSTRY AND THE MILITARY

Glendale 1, California

FACTORY BRANCH OFFICES SERVING ALL PRINCIPAL CITIES OF THE UNITED STATES AND CANADA

SIX PLANTS: IRON MOUNTAIN, MICHIGAN · GLENDALE, CALIFORNIA · BURBANK, CALIFORNIA · SKOKIE, ILLINOIS · GUELPH, CANADA · WARWICK, RHODE ISLAND



T-270 T-90H110 T-190

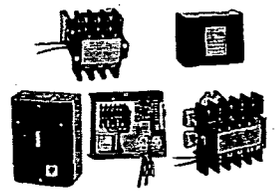
THERMOSTATS

New matched line of thermostats for heating and air conditioning—horizontal and vertical models. Distinguished by compactness and beauty of appearance—for use in all space heating remote control systems. Provide rapid response to radiant as well as convected heat. New mercury switch type. Heat Anticipator optional. Millivolt, 24-v and line voltage types to 13 amperes are available for heating and cooling.



AIR CONDITIONING CONTROLS

Complete control systems for residential, commercial and industrial air conditioning. Thermostats for cooling and heating-cooling combination, system-coordinating Master Control Panels, and a line of refrigeration controls including thermal expansion valves, solenoid valves and refrigerant distributors.



ELECTRICAL CONTROLS

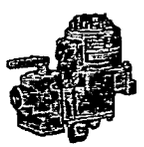
New Master Control Panels are packaged control systems combining all electrical components required for complete automatic control of refrigeration or air conditioning equipment. Electrical components such as relays, contactors, magnetic starters and transformers are also available individually.



K-3A ¼" K-3A 1½" K-3E 2½"

MAGNETIC GAS VALVES

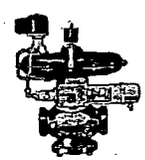
Magnetic, current failure, quiet, two-wire. Opens and closes fuel line supply at demand of room thermostat, limit control or similar device. Widely used for controlling gas to warm air furnaces, gas fired boilers, industrial furnaces, conversion burners, etc. Compact; low current consumption; humless ⅜ in. to 6 in. IPS for pressures up to 5 lb; a-c or d-c. Also available in explosion-proof model.



B-63

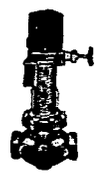
COMBINATION GAS VALVES

For ultimate space economy, an electric diaphragm gas valve combined with pilot valve, gas cock and low pressure regulator. B-53 models for 24-v systems have provision for optional plug-in thermopilot relay for safety shut-off in event of pilot failure. B-63 self-powered millivolt models inherently fail safe upon pilot failure, since operating power is generated from pilot flame heat.



SLOW-OPENING GAS VALVES

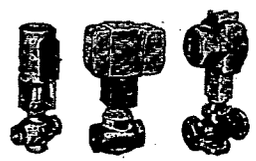
New combustion control afforded by these diaphragm-controlled gas valves. Governor regulates fuel supply to burner in direct ratio to steam pressure, eliminating hunting aspect common to most pilot throttling valves. Low fire adjustment prevents main valve or gas flow from reducing below adjustable set minimum, affording fullest advantage of gas burner installed. Type B-55G-1 pictured, available 1 in. to 6 in. in IPS.



V-110-1

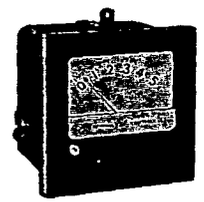
MANUAL RESET VALVES

Equipped with manually-reset electromagnetically-held valve operator for safety shutoff of fuel supply to boilers. Valve can be opened only when current is applied and is closed by spring force on interruption of current in response to high limit or safety switches. ½ in. to 6 in.; normally closed; screwed or flanged body; explosion-proof or weatherproof. *Factory Mutual* and *Underwriters* approved.



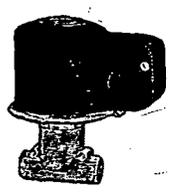
HYDRAMOTOR VALVES

Hydramotor valves, composed of electro-hydraulic actuators and valve bodies, are available in a wide range of types and sizes. Spring return for two-wire, current failure operation. Valve bodies and actuators are offered in combination as complete self-contained units. Bodies available in steel, brass, iron and alloy in sizes from ⅜ in. to 12 in., two-way and three-way bodies. Hydramotor valves are particularly suited to safety shutoff applications and control of steam, hot and/or chilled water systems.



BOILER COMBUSTION CONTROL SYSTEMS

For improved efficiency and better operational maintenance, General Controls offers an outstanding line of combustion controls and instruments suitable for use with oil, gas or stoker-fired boiler room installations in industrial and commercial installations. Included in the line are Flue Gas Temperature Indicator, Draft Gage, Steam Pressure Controller/Indicator, Draft Program Control, Air Pressure Gage. All are flush or surface mounted.



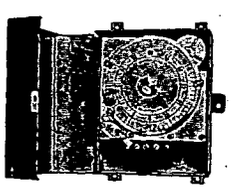
REFRIGERATION AND INDUSTRIAL SOLENOID VALVES

Available in a range of sizes from ⅛ in. pipe size midget valves to 6 in. pipe size. Pressure range to 1000 psi. Body construction in cast iron, forged brass, carbon steel, stainless steel and special alloys. Available in two-way, three-way, four-way, normally open, normally closed and manually reset models. Direct acting types, also pilot operated, diaphragm type, lever acting types. Available in a variety of coils for special applications such as mobile applications.



HAMMEL-DAHL AUTOMATIC CONTROL VALVES

Pneumatic and hydraulic actuators for fluid control from a Micro-Flo basis up to the largest type. Pneumatic diaphragm motors feature removable actuators, reversible superstructures, and a wide variety of material for body, lining and diaphragm. Body assemblies include globe, angle, Saunders, three-way and butterfly designs. Particularly suited for high temperature hot water systems.



TIME SWITCHES

For commercial and industrial systems or processes for automatic on-off control. Available in 7-day models and also 24, 12, 8, 4, 2, 1 hour rotations. Electric or spring-wound models. Snap switch or mercury switch.

HENRY VALVE COMPANY

3215 NORTH AVENUE, MELROSE PARK, ILLINOIS (Chicago Suburb) CABLE: HEVALCO, MELROSE PARK, ILLINOIS

Manufacturers of Valves, Driers, Strainers, Fittings and Accessories for Refrigeration, Air Conditioning and Industrial Applications.



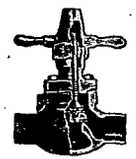
"GOLDEN BANTAM" PACKLESS VALVE—TYPE 516

A small packless line shut-off valve with ¼ in. through ⅝ in. flare, ODS or extended end solder connections. Also "Blue Bantam" and "Standard" Balanced-Action packless line shut-off valves with ¼ in. through ⅝ in. Fl. and ¼ in. through 1⅛ in. ODS size connections.



CROSS-VU® HERMETIC SEAL LIQUID INDICATOR—TYPE LI-16

Brass indicator with glass fused directly to metal body. Maximum working pressure 500 PSI Connection sizes: ¼ in. through ⅝ in. M.Fl., M.Fl. x F.Fl. or with extended ends and solder connections.



WING CAP PACKED VALVES—TYPE 203

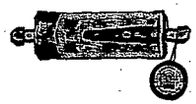
The standard of the industry. Bronze alloy with ODS connections in sizes ⅞ in. through 4⅛ in. (Globe) and ⅞ in. through 3⅛ in. (Angle). Also semi-steel with ½ in. through 2½ in. F.P.T. connections and semi-steel flanged type globe and angle valves with brass ODS or steel butt-weld adapters, and one piece socket weld or F.P.T. flanges in a wide range of sizes from 1 in. to 5⅛ in.



ANGLE RELIEF VALVE—TYPE 52

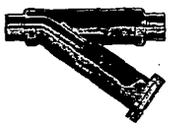
UV *ASME—NB—National Board Certified*

Forged brass automatic pressure relief valve. Pressure setting range 50 to 450 PSI Size connections: (MPT Inlet x Flare Outlet) ¼ in. x ⅜ in., ⅜ in. x ⅜ in., ⅜ in. x ½ in. and ½ in. x ⅝ in. Also ⅝ in. ODS x ⅝ in. ODS.



"DRI-COR" FILTER-DRIER—TYPE V800

Ceramic fired desiccant filter-core in bed of granular desiccant—a blend of molecular sieves and activated alumina. Provides high capacity drying even at relatively high temperatures, micronic filtration, efficient acid removal and low pressure drop. Pressure sealed. Wide range of sizes with ¼ in. through ⅝ in. Fl. connections. "DRI-COR" Filter-Drier Cartridges also available for Henry Cartridge Type Angle Driers.



"Y" STRAINER—TYPE 895

Flanged design permits cleaning without removing strainer from line. Screen cylinders securely located by spring tension prevents by-passing of refrigerant. Forged brass end caps with integral end connections. Screen area: 10 through 150 sq in. Size connections: ⅝ in. through 4⅛ in. ODS. Angle and straight-through type strainers also available.

Write for catalogs describing the complete line.



Catalog 103 Valves, Driers, Strainers and Accessories.



Catalog 202 Ammonia: Valves and Accessories.

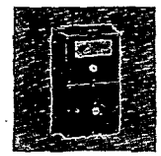


Catalog FF *Forged* Carbon Steel Fittings.

Illinois Testing Laboratories, Inc.

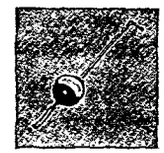
Room 516, 420 N. La Salle St., Chicago 10, Illinois

Alnor® FOR PRECISE MEASUREMENT OF TEMPERATURE... AIR VELOCITY... DEW POINT...



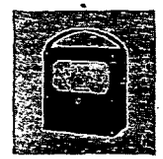
PYROMETERS

A complete line of single or multiple-point pyrometers—portable or mounted—for temperature measurement in all types of service. *Particularly adapted to exhaust, cooling water and lubricating oil temperature measurements on diesel and gas engines.* For furnaces, kilns, ovens, vats, pots, heat exchangers, kettles, reactors, autoclaves, pipe lines and any similar equipment where accurate temperatures are desired.



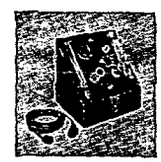
PYROCON

A self-contained, hand-held portable contact pyrometer. Takes any surface temperature. Widely used in plastic molding, chemical processing, electrical, rubber and paper manufacturing industries. Available with fixed or flexible arm.



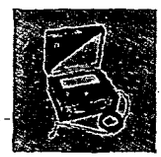
VELOMETERS

An instantaneous air velocity meter giving direct readings in feet per minute without necessity of timing, calculation, or reference to tables or charts. A compact, self-contained, portable unit. For low range readings, Velometer is held in air stream. For high range readings, tube-connected special jets provide convenient use in any location. Standard ranges from 20 f.p.m. to 24,000 f.p.m



DEWPOINTERS

An improved instrument for simple, rapid dew point reading of any gas. Indication takes place in an enclosed observation chamber under accurately controlled conditions. Dew or fog suspended in air—not on mirrored surface—makes it easy for two or more observers to get the same accurate results. The Alnor Dewpointer is a complete, self-contained unit—no external coolants or other apparatus are required.



SURFACE PYROMETER

The Alnor type 2300 Pyrometer sets a new high in precision measurement of low surface temperatures. Special design, from Alnico magnet to quick reacting thermocouple, for accuracy...speed and dependability. This is a self-contained, sturdy unit—easily carried and used anywhere on a wide variety of applications. Direct reading, six-inch mirrored scale is easily read...quickly understood. The instrument is available in five Fahrenheit scale ranges with a span as small as 150°.



PORTABLE PYROMETER

The Alnor type 1500 is designed for work with electrically heated mantles. This lightweight, magnetically shielded pyrometer gives fast, precise temperature reading to production men in all industries. Choice of *ten scale ranges*, 0-400° to 0-3000°F. Automatic cold end compensator is standard equipment.

Alnor® ILLINOIS TESTING LABORATORIES, INC.
Room 516, 420 N. La Salle St.•Chicago 10, Illinois

PRECISION INSTRUMENTS FOR EVERY INDUSTRY

JOHNSON SERVICE COMPANY

AUTOMATIC TEMPERATURE AND AIR CONDITIONING CONTROL

General Office & Factory 507 E. Michigan St. Milwaukee 1, Wis.

Direct Branch Offices in Principal Cities

Johnson Controls Ltd., 120 Bermondsey Rd., Toronto 16, Ont.

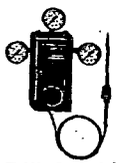
MANUFACTURERS, ENGINEERS and CONTRACTORS—A complete line of automatic temperature and humidity controls for heating, cooling, ventilating, air conditioning, industrial processing, military installations and special temperature and humidity control applications.

SPACE CONTROL—Automatic room temperature and humidity control for convectors, radiators, radiant heating, unit ventilators, unit heaters, and heating and cooling air delivery equipment.

INDUSTRIAL PROCESSING—Automatic temperature and humidity control for every requirement in manufacturing and industrial processing. Thermostats, valves, dampers for tanks, dryers, vats, kettles, curing rooms, coolers, kilns, and other special equipment in textile, rubber, paper, petroleum, meat, dairy, baking, sugar, brewing, distilling, tanning, candy, foods, pharmaceutical, and other industries.

SERVICE ORGANIZATION—JOHNSON sales engineers and trained installation experts at all branches in United States and Canada. No agents, jobbers, or part-time representatives! All Johnson salaried personnel for exclusive service of Johnson Service Company customers.

JOHNSON LITERATURE: Available at your nearest JOHNSON branch office or write to address above.



T-901 Liquid-filled Remote Bulb Submaster Thermostat

JOHNSON THERMOSTATS

Room Thermostats—single and dual temperature (Day-Night, Heating and Cooling). Proportional or two-position action. Maintain temperatures within one degree above or below setting. Wide selection of adjusting features, guards, and mounting. Furnished with or without thermometers.

Insertion and Immersion Thermostats—Liquid-filled temperature measuring elements. Stem type for direct measurement. Remote bulb types for applications requiring location of thermostat at a distance from the point of temperature measurement. Various bulb styles available. Standard capillary length is 8 ft. Lengths of 15, 25, 35, 50, 65, 75, and 85 ft available on special order.

Submaster Thermostats—The set point of these Johnson controllers is changed from a remote location by the action of another (master) controller. Available with liquid-filled stem and remote bulb measuring elements.

Johnson "Duo-Stats," weather compensated primary controllers, control heating system temperatures in accordance with the outdoor temperature. Duo-Stats are available for any heating system, all types of buildings, and furnished in pneumatic, electric or electronic models.



Series T-400 Room Thermostat



T-804 Liquid-filled Immersion Thermostat

PRESSURE CONTROLLERS

Pressure Regulators with ranges of 30 inches of water to 300 psi. Static pressure regulators measure variations in pressure from 0.009 inches to 3 inches of water. Used also as differential regulators measuring difference in pressure between two chambers. Pressure switches and governors also available.



R-316 Air Flow Controller

The Johnson Air Flow Controller is applied to mixing boxes of high velocity, high pressure, dual duct air conditioning systems. Provides positive constant volume air discharge through each mixing box. Operates at any pressure difference from approximately 0 to 1 inch wg and will produce a 1 psi air pressure change on pressure difference changes of less than 0.005 inch wg.

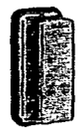


R-1051 Static Pressure Indicator-Controller

JOHNSON HUMIDITY CONTROL

Humidostats—Room and insertion types with humidity measuring elements of wood or animal membrane, the most sensitive of which controls within 1 percent at relative humidities as high as 95 percent, at temperatures of 200 F. Submaster models also available which are adjustable from a remote point by another controller (master) or may be manually readjusted from a remote point by means of a Johnson Gradual Switch.

Humidifiers—Steam grid (perforated pipe supplied with low pressure steam) or pan type with copper evaporating pan, brass or copper heating coils and float control.



T-1210 Room Temperature Transmitter

PNEUMATIC TRANSMISSION

Johnson pneumatic transmission systems provide a greatly simplified and superior means of centralized indication and control for modern air conditioning, heating and ventilating systems. Control centers are custom designed to fit the requirements of the individual building. Johnson offers a complete line of pneumatic transmission and control center instrumentation, including room transmitters, remote bulb transmitters, indicators, receiver-controllers, recorders and related apparatus.



T-1270 Receiver Controller

SPECIAL CONTROLLERS

Aspirating Cabinets—Several models with many combinations of controlling and auxiliary devices required to provide control of temperature and humidity. Cabinets equipped with blowers for positive air circulation and air gauges for checking function of controllers at all times.



12 Point Model Pneumatic Step Controller

Pilot Positioners—for dependable and accurate repositioning power. Applied to valves and damper operators. Respond to smallest changes in air pressure. New Precision Pilot acts 6 times faster than ordinary pilots.

Pneumatic Step Controllers—for capacity control of reciprocating refrigeration equipment or any application where sequence of pressure switches is used.

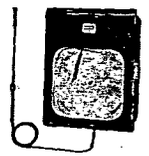
Record-O-Stat—Combination capillary temperature controller and recorder. Available in single-pen, two-pen, and submaster models. Indicating controllers and recorder-controllers also available.



V-530 Precision Pilot Positioner Applied to Valve

Liquid Level Regulators—(Float Type) control within extremely close limits. Mounted through wall of containing vessel by stem with 1-inch pipe thread. Floats of copper, stainless steel or special alloys.

Thermometers—Insertion and immersion types for air and liquids. Stem and remote bulb temperature measuring models with vapor tension or liquid filled elements. Variety of styles, mountings and finishes.



T-1000 Single Pen-O-Stat Record-O-Stat

JOHNSON VALVES

Johnson Diaphragm Valves—Simple, rugged. Diaphragms of special molded rubber, resistant to age and oxidation, operate valve stems against spring pressure. Available also with "Sylphon" seamless metal bellows. In standard sizes and patterns including compact valves for convectors and unit conditioners. Normally open or normally closed. Three-way mixing and by-pass valves, for water, brine and other liquids.



V-153 "Sylphon" Convector Valve

Johnson "Streamline" Diaphragm Valves—With modulating plugs and special internal construction. Smooth proportional control. Where greater power is required for repositioning at slightest demand of controlling instruments, the larger molded rubber diaphragm valves are fitted with pilot positioners.

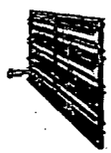
Series V-210 and V-214 Three-Way Valves—for mixing and by-pass applications. 100 percent tight closing. Can be serviced without removal from the line. Sizes 2½ to 6 in. 125 and 250 lb standard construction.



V-210 Three-Way By-Pass Valve

JOHNSON DAMPERS AND SWITCHES

Standard Johnson Dampers—Galvanized 16-gauge steel blades in flat steel frames with adequate bracing to form rigid assembly. Black lacquer or special corrosion-resisting finishes. Angle iron frames optional.



D-225 Proportioning Louver Damper

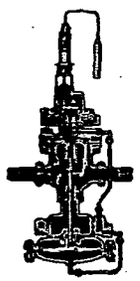
Special Dampers—Galvanized iron frames and monel metal, aluminum, copper or rust-resisting steel blades on special order. Frames of same material as blades, if desired. Brass pins in steel bearings or ball bearings.

Johnson Damper Operators—Similar in operating principle to valves. Specially molded rubber diaphragm or seamless metal bellows operates damper through suitable linkage. Johnson "piston" damper operators afford long travel at full power. With or without pilot mechanism, as described for "Valves."

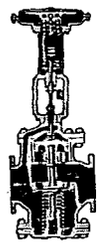


D-251 Piston Damper Operator

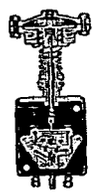
Johnson Pneumatic Switches—For direct operation of dampers and to switch controllers in and out of service from remote locations. Standard switchboard panels are Masonite. Ebony, asbestos, steel, polished oak and genuine or imitation marble panels on special order. Special switchboards including all types of control equipment available on special order.



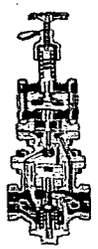
Temperature Regulator
Class T



Diaphragm Control
Valve Class D-1



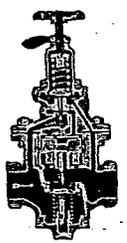
Control Pilot
Type PD



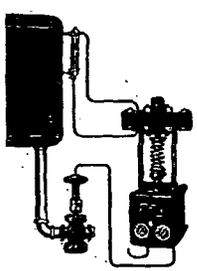
Pump Governor
Class PR



Self-Cleaning
Strainer



Reducing Valve
Class L-3



Floatless Level Control
for Boilers, Tanks,
Evaporators, Heaters,
Standpipes, etc.

LESLIE CO.,
237 Grant Avenue, Lyndhurst, New Jersey

TRADE LESLIE MARK

REGULATORS and CONTROLLERS

for the Regulation and Control of Pressure, Temperature, Flow and Liquid Level • • •

LESLIE COAST-TO-COAST SERVICE—In all principal industrial centers LESLIE factory-trained engineers are available to help with any control problem involving steam, liquid, air or gas. Look for LESLIE REGULATORS under "Valves" or "Regulators" in your classified telephone directory.

INTERCHANGEABILITY—*All internal parts of* LESLIE Regulators are dimensionally interchangeable and have maximum interchangeability in all sizes and classes.

LESLIE SPECIAL FEATURES—*at no extra cost!*

- ★ *Stellited Seat Rings*
- ★ *Hardened Stainless Steel Main Valves, 500 Brinell*
- ★ *Corrosion Resistant Springs*
- ★ *Hardened Stainless Steel Cylinder Liners, 500 Brinell*

Supplied as standard equipment.

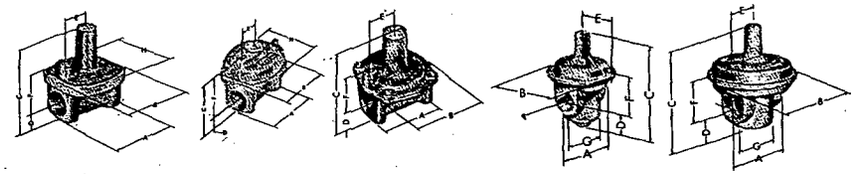
LESLIE PRESSURE REDUCING VALVES, PUMP GOVERNORS, and TEMPERATURE REGULATORS — Self-contained, spring *loaded, internal pilot piston* or diaphragm operated with stellited seat rings, hardened stainless steel trim, bronze, iron, or steel construction. ALL LESLIE Regulators feature:

- ★ *High accuracy of regulation*
- ★ *Single-seated construction for positive dead-end shutoff*
- ★ *Quick valve action for sudden load changes*
- ★ *Maximum resistance to corrosion and wear*
- ★ *Maximum spring range — from minimum to maximum of the range without change of spring or diaphragm*

MAXITROL COMPANY

12200 Beech Road, Detroit 39, Michigan

GAS APPLIANCE PRESSURE REGULATORS



1 2 3 4 5

FOR MAIN BURNER LOAD APPLICATIONS

DRAWING	MODEL	DIMENSIONS A	B	C	D	E	F	G	H	SHIP. WT.	NOMINAL SIZE (INCHES) INLET	OUTLET	OUTLET PRESSURE ADJUSTMENT INCHES w.c.	PRESSURE DROP CAPACITY at 0.3" w.c. Pressure Drop cu ft/hr 0.6 sp gr	Btu/hr 1000 Btu/cu ft 0.7 sp gr Gas	Btu/hr 500 Btu/cu ft 0.6 sp gr Gas	REGULATION CAPACITIES Btu/hr—400 Btu/cu ft, 0.7 sp gr Gas** (MAXIMUM)	(MINIMUM)
	RV-10*	1¾	1²¹/₃₂	2½	⅝					1/₁₆#	⅛	⅛	2-5	15	11,000	7,450	17,000	1,700
1	RV-30*	2¼	2⁹/₁₆	3¹/₁₆	⅛	⅜	1¹³/₃₂		2⁹/₁₆	⅛#	⅜	⅜	2-5	81	60,000	40,500	100,000	10,000
											½	⅛	2-5	81	60,000	40,500	100,000	10,000
											½	½	2-5	88	65,000	44,000	100,000	10,000
	RV-30A*	2¼	2⁹/₁₆	2¹/₁₆	⅛	⅛	1¹³/₃₂		2⁹/₁₆	⅛#	⅜	⅜	3-4	81	60,000	40,500	100,000	10,000
											½	⅜	3-4	81	60,000	40,500	100,000	10,000
											½	½	3-4	88	65,000	44,000	100,000	10,000
2	RV-31*	2¼	2⁹/₁₆	2¹/₁₆	⅛	⅛	2¹/₁₆		2⁹/₁₆	1/₁₆#	⅛	⅜	3-4	81	60,000	40,500	100,000	10,000
											½	⅜	3-4	81	60,000	40,500	100,000	10,000
											½	½	3-4	88	65,000	44,000	100,000	10,000
3	RV-41*	2¹¹/₁₆	2¹⁵/₁₆	3¹³/₃₂	²³/₃₂	1¹¹/₆₄	1²¹/₃₂			⅛#	½	½	2-5	126	93,000	61,000	152,000	15,200
											½	½	2-5	135	100,000	67,500	174,000	17,400
	RV-41A*	2¹¹/₁₆	2¹⁵/₁₆	3¼	²³/₃₂	1¹¹/₆₄	1²¹/₃₂			⅛#	½	½	3.5—4.5	126	93,000	61,000	152,000	15,200
											¾	¾	3.5—4.5	135	100,000	67,500	174,000	17,400
4	RV-42	2½	3⅛	4⁵/₁₆	1⅝	1¹⁷/₃₂	1⁷/₃₂	1½		⅛#	½	½	2-5	170	126,000	85,000	196,000	19,600
	RV-50	3	3¹¹/₁₆	5¹/₁₆	1⅜	1¹³/₁₆	1½	1¹³/₁₆		1#	½	½	2-5	270	200,000	135,000	310,000	31,000
											⅜	¾	2-5	270	200,000	135,000	310,000	31,000
	RV-51	3⅝	4⅜	6¹/₁₆	2¹/₁₆	2¼	1⅜	2⅛		1½#	¾	⅜	2-5	450	333,000	225,000	517,000	51,700
											1	1	2-5	460	340,000	230,000	527,000	52,700
	RV-60	4⅛	5¹/₁₆	7⅛	2¹/₁₆	2¹¹/₁₆	1¹³/₃₂	2⁷/₃₂		2½#	1	1	2-5	676	500,200	338,000	775,000	77,500
											1¼	1¼	2-5	705	521,700	352,500	810,000	81,000
5	RV-81	6	7	8⅝	2⅝	2⅜	3¹/₁₆	3½		5#	1¼	1¼	2-5	1250	925,000	625,000	1,430,000	143,000
											1½	1½	2-5	1260	932,400	630,000	1,450,000	145,000
	RV-91	7¼	9⅛	10⅛	2⅜	3¼	4⅜	4¼		9#	2	2	2-5	2030	1,502,200	1,015,000	2,330,000	233,000
											2½	2½	2-5	2030	1,502,200	1,015,000	2,330,000	233,000
	RV-110	9	12⅛	14¼	3½	4⅛	5¹/₁₆	6		20#	2½	2½	2-5	4200	3,108,000	2,100,000	4,850,000	485,000
											3	3	2-5	4900	3,626,000	2,450,000	5,650,000	565,000

* These models also available A.G.A. certified for main burner and pilot load applications (minimum capacity 120 btu/hr).
** To convert to Btu/hr—0.6 sp gr Gas, multiply by .675.

PRESSURE SETTINGS:

Standard outlet pressure settings for the various models are as follows:

RV-10	3.0 in. w.c.
RV-30, RV-30A and RV-31	3.5 in. w.c.
RV-41¾ in., RV-41A¾ in., RV-41A½ in.	4.0 in. w.c.
RV-41½ in., RV-42, RV-50, RV-51, RV-60, RV-81, RV-91	3.5 in. w.c.
RV-110	4.0 in. w.c.

Upon Request, any Maxitrol gas regulator will be adjusted to any other pressure within the range of the standard spring at no extra charge.

INLET PRESSURES:

All Maxitrol gas regulators can be satisfactorily operated at inlet pressures up to one psi. Under certain circumstances even higher inlet pressures are permissible. Consult factory.

MULTI-POISE MOUNTING:

"Straight-Thru-Flow" series regulators should not be installed in a vertical pipe run if the gas is flowing upward. Otherwise, all Maxitrol gas regulators may be installed in any plane at any angle.

VENTS:

Position—Models RV-42, RV-50, RV-51, and RV-60 are manufactured with horizontal vent outlets. Various vent positions are available. Consult *catalog 57*. The vent boss on all other regulators is located vertically over the outlet end of the regulator.

Boss Size—All models except RV-10, and RV-110 are tapped with ⅛ in. NPT. RV-10, no tapped boss. RV-110, boss tapped ¼ in. NPT.

THE MERCOID CORPORATION

Main Office and Factory, 4201 Belmont Ave., Chicago 41, Ill.

NEW YORK 17, N. Y.
205 E. 42nd Street

PHILADELPHIA 32, PA.
3137 E. Broad Street

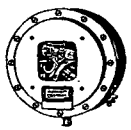
Automatic Controls for Heating, Air Conditioning, Refrigeration and Industrial Applications

Complete line of Mercoid Controls and mercury switches shown in Catalog No. 858A

SERIES "D" PRESSURE CONTROLS



General Purpose Type (Nema 1): For indoor service and other general purpose applications. Equipped with outside adjustments, visible calibrated dial, visible on-off hermetically sealed mercury contact. Available in 23 pressure ranges from 0-30 in. vac. to 300-2500 psi with differential varying from 2 oz to 150 psi depending upon range selected.

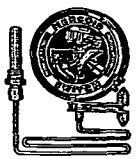


Weather Resistant Type (Nema 1A, 2, 3, 4): For outdoor service and other applications. Has external adjustments, visible dial, hermetically sealed mercury contact. Available with same operating ranges as noted under general purpose types.



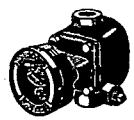
Explosion-Proof, Class 1, Group C (Nema 7), Class 2, Group E, F, & G (Nema 9, 9A): For hazardous locations. A new light weight control (8¼ lbs) equipped with external adjustments, visible calibrated dial and visible on-off hermetically sealed mercury contacts. Available with same operating ranges as noted under general purpose pressure controls.

SERIES "D" TEMPERATURE CONTROLS



For liquids or gases, such as air, oil, water, paraffin or distillate vapors and other industrial applications. Equipped with outside adjustments, visible dial and visible on-off hermetically sealed mercury contact. Operating ranges from 30 to plus 60°F., up to 370-530°F. Available with Weather-Proof or Explosion-Proof enclosures.

LIQUID LEVEL CONTROLS



Various types available such as for boiler feed water pump control, combination pressure and low water control or for use where low specific gravity liquids are used. Can be furnished with explosion-proof case for hazardous locations.

If you have a control problem involving the automatic control of pressure, temperature, liquid level, mechanical operations, etc., it will pay you to consult Mercoid's engineering staff—always at your service.

TRANSFORMER-RELAYS



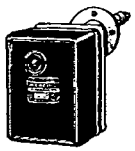
Type V, a reliable low voltage mercury contact relay which also acts as a transformer, inducing low voltage (24 volts) on the pilot circuit. Used for all types of automatic equipment. Available in various voltages, cycles and circuit arrangements.

VISAFLAME CONTROL



For direct burning mounting. Operates direct from the light of the flame instead of from the heat in the stack. Used with a Mercoid panel unit for either intermittent or constant ignition burners. Provides full protection against flame, ignition and power failure.

OIL BURNER SAFETY AND IGNITION CONTROLS



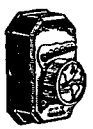
Type JMI, a stack safety control, for positive protection against flame or ignition. Also provides complete protection against low line voltage or power failure. Type JM, available for constant ignition burners.

LOW VOLTAGE THERMOSTAT



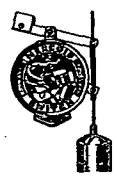
The Mercoid Sensatherm is a very sensitive room thermostat, operating on a differential of ½°F. No internal heater coils or other means of artificial acceleration are used to maintain an even room temperature.

LINE VOLTAGE THERMOSTATS



For direct control of motors operating automatic equipment. No relay required. These thermostats incorporate a liquid charged bellows. Expansion and contraction of the bellows tilts a Mercoid mercury to mercury contact to open or close the electric circuit. Standard ranges: 56-80°, 65-90°, 38-70°, and 25-60°F. Available with manual on-off switch for use with industrial unit heaters. Can be furnished with explosion-proof housing for hazardous locations.

LEVER ARM CONTROLS

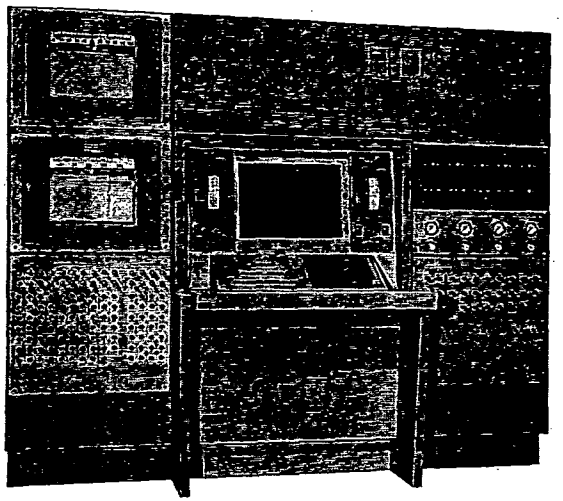


Mercoid lever arm controls have a variety of applications where it is desired to mechanically open and close electric circuits. Mercoid float controls are used for maintaining fluid levels in tanks, or for control of sump pumps, etc.

MINNEAPOLIS-HONEYWELL REGULATOR CO.

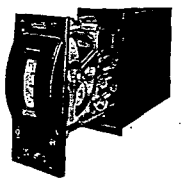
Minneapolis, Minnesota

A COMPLETE LINE OF PNEUMATIC, ELECTRIC, ELECTRONIC CONTROLS AND INSTRUMENTS



Supervisory DataCenter* allows one man to supervise and control all the mechanical systems of a building from a central location; remotely check and adjust temperatures, start and stop compressors, pumps and fans.

A new Selectrafic* panel clearly displays the desired mechanical system on a single small screen, while the multipurpose control board is automatically switched to that system. This new feature permits all graphic layouts to be shown, yet requires less wiring, is smaller and standardized. Panel location and space requirements can be determined early in the building's design. Additional functions can be incorporated by adding modular units.



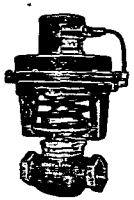
Electronik* temperature indicator gives an unlimited number of remote temperature readings on one instrument. Accurate to ±1/5 of a degree. Remote readings can be taken of any point in the air-conditioning system simply by pressing a button. Requires less than a square foot of panel space.

You can look to Honeywell for all your control needs—centralized panels, instruments, valves—because Honeywell has a complete line available. You'll get quick service from 112 offices across the nation, each staffed with control specialists who will assist and advise you. Contact Honeywell even before blueprints are started to insure efficiency, saving, satisfaction.

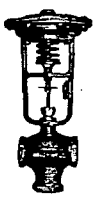
Shown below are three typical control valves in Honeywell's complete line of electric and pneumatic valves for commercial and industrial applications.



Three way electric valve controls liquids in either mixing service or in systems requiring flow be directed from either of two inlets to a common outlet. Constant total flow throughout full plug travel. Strain release mechanism insures tight close-off without motor strain. Sizes: screwed—½ in. to 2 in.; flanged—2½ in. to 4 in.



Single seated pneumatic valve used for normally closed modulating control of steam, hot water, or cold water on heating or air-conditioning coils. Also for general purpose —zone systems, booster coils, cooling units using water, etc. Equal percentage, high lift plug. Sizes: screwed—½ in. to 3 in.; flanged —2½ in. to 6 in.



Industrial diaphragm-control valve. Three-way valve used for mixing service with two inlets, one outlet, or for diverting service. Normally open to bottom port. Linear flow characteristic. Rangeability: 30 to 1 for each port. Maximum operating air pressure 35 psi. Sizes: screwed—½ in. to 2 in.

*Trademark

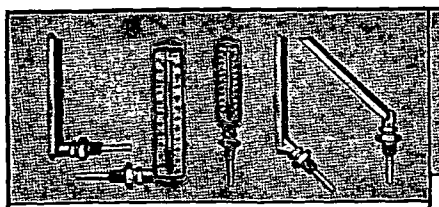
FOR COMPLETE INFORMATION CALL YOUR NEAREST HONEYWELL SALES OFFICE

MOELLER INSTRUMENT COMPANY

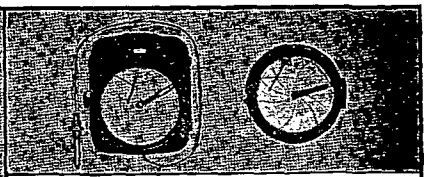
SINCE 1867

132nd STREET & 89th AVENUE, RICHMOND HILL, NEW YORK, N.Y., U.S.A

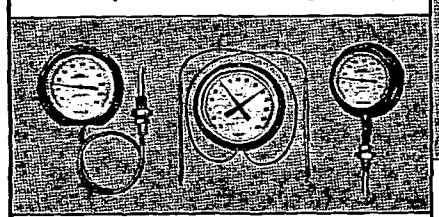
Representatives in Principal Cities



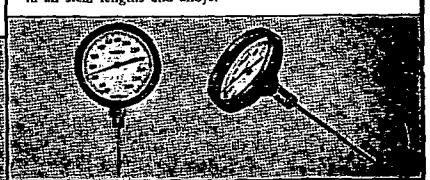
MOELLER INDUSTRIAL THERMOMETERS made in all forms, Straight, Angle, Right Side Angle, Left Side Angle, Reclined and Inclined Angles, in scale ranges from minus 120 to plus 1125°F. or its equivalent in centigrade. Available in 5", 7", 9" and 12" scale sizes with plain stems, union connection stems and separable sockets in all stem lengths and alloys.



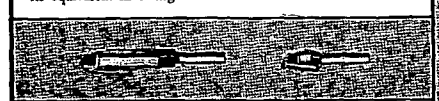
RECORDING THERMOMETERS, Mercury Actuated, made in the round or rectangular cases. Charts 10" or 12", 1 hour, 12 hour, 24 hour or 7 day. Ranges from minus 40 to plus 1000°F. or its equivalent in centigrade. Made in remote reading, direct reading, portable and self contained styles. Plain stems, union connection stems and separable sockets in all stem lengths and alloys.



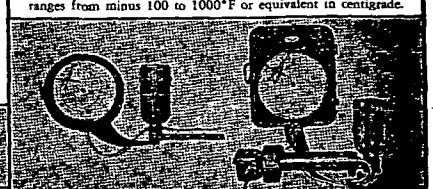
DIAL INDICATING THERMOMETERS, Mercury Actuated, made in 4½", 6" and 8½" sizes with flexible extension tubing. Adjustable rigid stems, single or two pointers, plain bulb, union connection bulb or separable sockets in all lengths and alloys. Scale ranges from minus 40 to plus 1000°F. or its equivalent in centigrade.



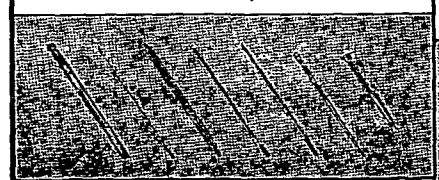
MOELLER BIMET THERMOMETERS are actuated by a powerful, specially processed bi-metal helix. Made in 5" and 3" diameter case size. Rigid stainless steel stem, can be furnished with union connection stem and separable socket in all alloys. Scale ranges from minus 100 to 1000°F or equivalent in centigrade.



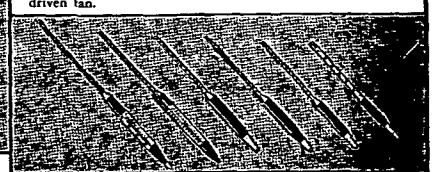
THERMOMETER TEST WELLS, THERMOCOUPLE PROTECTING TUBES and WELLS, made up in all sizes and alloys.



RECORDING PSYCHROMETERS, Mercury Actuated. Round and Rectangular Case, 12 hour, 24 hour or 7 day charts, self contained or remote reading, with or without motor driven fan.



ENGRAVED STEM and PORCELAIN and PAPER SCALE THERMOMETERS made in ranges from minus 150 to plus 1125°F. or its equivalent in centigrade. Available in standard and extreme precision grade. Made to meet specifications of the Government, A.S.T.M. and other authoritative bodies.



HYDROMETERS PLAIN FORM or COMBINED with THERMOMETER. Made in all scales specific Gravity or Baume for heavy and light liquids. Brix, Twaddle, Ballings Scales, etc. available for all liquids. Standard and commercial grades.

PENN CONTROLS, INC.

Goshen, Indiana

In Canada: Scarborough, Ontario **Export Div. 27 E. 38th St., New York 16, N. Y.**

Offices and Representatives

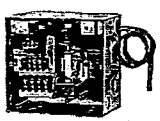
Albuquerque, N. M.—Decatur, Ga.—Berkeley 2, Calif.—Newton 58, Mass.—Chicago 11, Ill.—Cleveland 3, Ohio—Columbus 21, Ohio—Dallas 7, Texas—Dayton 20, Ohio—Denver 2, Colo.—Houston 22, Texas—Los Angeles 26, Calif.—Milwaukee 5, Wis.—Moline, Ill.—Minneapolis 19, Minn.—North Bergen, N. J.—Philadelphia 11, Pa.—Pittsburgh 16, Pa.—Rochester 18, N. Y.—St. Louis 10, Mo.—Salt Lake City 1, Utah—Seattle 1, Wash.—Wichita 2, Kans.

Wholesalers and Warehouses in Principal Cities

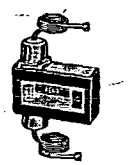
AIR CONDITIONING AND COMMERCIAL COOLING CONTROLS



Fan Center to control the fan coil unit on residential air conditioning with remote cooling equipment. Compact ... only 4½ in. wide and 6 in. high ... it combines a transformer with fan relay, reset relay and heating relay ... *one or all.*



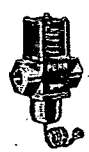
Cooling Center for control of remote condensing unit or water chiller. Only 7 in. high and 8 in. wide, it combines pressure controls, contactor, auxiliary relay for (a) water chiller pump, (b) tower pump or fan relay or both, (c) condenser fan relay.



Series 275 Oil Protection Control for pressure-lubricated compressors. Stops operation of refrigerating equipment before a failure in oil pressure system can damage the unit. Features a 90-second time-delay switch with manual reset to re-start compressor.



Series 273 Dual Pressure, 2-pole, refrigeration control handles polyphase motors without use of magnetic starter. With its 2 separate circuits, each unit is really 2 switches in 1. Available in single function control, also temperature models in single and dual function.

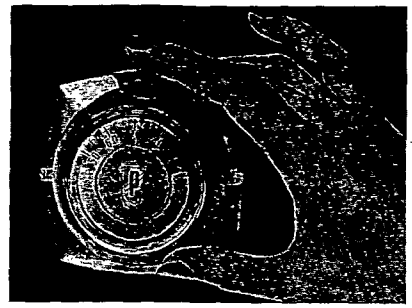


Series 246 Water Valve for pressure or temperature application on water-cooled refrigeration compressors and condensers. All range design handles both R-12 and R-22. Threaded and flange models. Also 2-piece valves for hermetic systems.

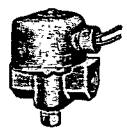


3-Way Water Valve designed for atmospheric or forced draft cooling towers. For use on single or multiple condenser hook-ups to provide the most economical and efficient use of the tower regardless of surrounding air temperature or humidity. Available in ½ in., ¾ in., 1 in. and 1¼ in. sizes.

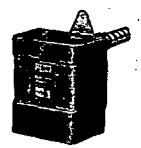
DOMESTIC HEATING CONTROLS



Series 888 Rimset Thermostat designed for close control of room temperatures in heating, cooling or combination year. 'round air-conditioning systems. Thermostat unit is interchangeable with various subbases to handle 12 different heating and cooling jobs. All wiring is on sub-base ... you simply "plug in" the thermostat.



Series 926 Gas Valve features straight-through flow and vertical self-cleaning seat. Designed for ½ lb pressure on all types of gas. Pressure regulators and pilots are also available.



Series 680 Stack Switch for continuous or intermittent ignition oil burners. Available in either single or two-unit models. Features low-voltage protection and automatic recycling on power interruption.



Type 520 Fan, Limit Control for warm air furnaces. Liquid expansion type in either combination fan and limit or separate models for fan and limit control. Safety limit controls for hot water jobs also available.

AUTOMATIC CONTROLS FOR HEATING, REFRIGERATION, AIR CONDITIONING, APPLIANCES, PUMPS, AIR COMPRESSORS, ENGINES

THE POWERS REGULATOR COMPANY

Specialists in Thermostatic Control Since 1891

Factory and General Offices, Skokie 84, Ill.

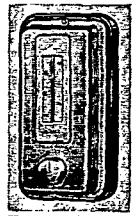
OFFICES THROUGHOUT UNITED STATES AND CANADA

New York 17, N. Y.: 231 East 46th St. • Chicago 26, Ill.: 6655 Ridge Blvd. • Los Angeles 26, Calif.: 3200 W. Temple St. • Toronto, Ont.: 15 Torbarrie Rd.

CONTROL SYSTEMS FOR HEATING, VENTILATING, AIR CONDITIONING and Processes, also Hot Water Supply Controls



Room Thermostat

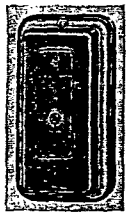


Heating-Cooling Thermostat

When you specify or install a Powers System of Temperature Control you can rely on Powers complete responsibility for a correctly engineered control system, proper installation, continuous successful operation and SERVICE when required from offices in 85 cities.

Some of the many controls employed in a Powers pneumatic system are shown here. A few hot water supply controls are shown at bottom of opposite page. For engineering assistance or bulletins contact your nearest Powers office or write nearest office listed above.

Pneumatic Room Thermostats, Hygrostats, Damper Motors



Hygrostat

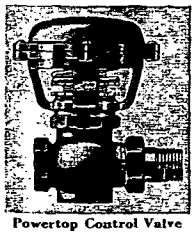
Powers Thermostats provide true gradual action for precise automatic control of room temperatures. They are available for single temperature control with direct or reverse action; two temperature or day-night control with direct action only; heating-cooling for air-conditioning control is also available with remote bulb; and submaster type with control point readjustable by a remote master instrument.

Powers Hygrostats are designed for pneumatic control of valves, dampers and similar devices used with various types of humidifying equipment. They are available for single point control for room or duct installations, direct or reverse action; also for submaster control with reverse action and with direct or reverse adjustment.

POWERSTROKE Damper Motors are piston type for gradual control of dampers or other final control elements. Available in three sizes: 3 in. (2 in. stroke) for limited space applications, 4 in. (3 in. stroke) and 6 in. (4 in. stroke) for smooth gradual control of automatic dampers up to 32 sq. ft. or for other applications.



Powerstroke Motor

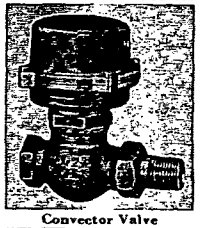


Powertop Control Valve

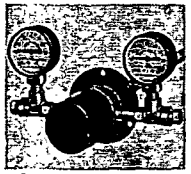
Powers PACKLESS Control Valves eliminate packing maintenance and damage from water leakage.

POWERTOP Valves are designed for smooth proportional control of steam or water flow. With 10 sq. in. effective diaphragm area, it is a compact unit for limited space applications such as unit ventilators, radiant panels, reheat coils, etc. Sizes: ½ in., ¾ in. and 1 in. with angle union or globe union bodies.

Convector Valves are extremely compact and especially adapted to very limited space applications. Available in three body types: angle union, globe union and reverse flow union. Angle and globe type sizes are ½ in. and ¾ in., reverse flow ¾ in.



Convector Valve



Limitem Thermostat

Powers LIMITEM Thermostat is a precision instrument for: (1) low limit control of unit ventilators (2) return air, mixed air and fan discharge temperature control with central fan systems and (3) pipe line insertion when used with a well. It provides convenient adjustment for any control point between 35 F and 185 F. Various mounting arrangements can be supplied.

Series 200 SUBMASTER Indicating Regulator has a control point that is pneumatically readjusted by a master instrument. It is available with sensitivity adjustment, direct or reverse action, and with direct or reverse reset adjustment.

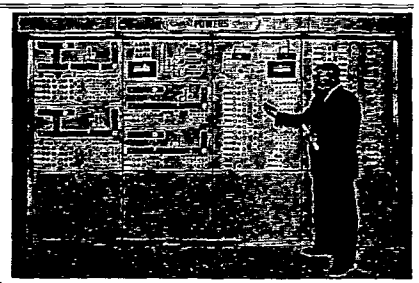


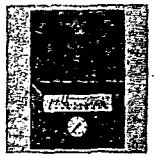
Series 200 Submaster

POWERS GRAPH-O-MATIC PANELS

Powers central control panels combine precise instrumentation with simplicity of operation. They can be equipped to take care of a variety of functions such as remote readjustment of controllers, continuous temperature indicators, remote positioning of dampers and valves. Simplicity of pneumatic controls insures continuous and reliable service.

Graph-O-Matic Panels are also available in console models. All types are custom designed and built to meet the requirements of any comfort or process control system.



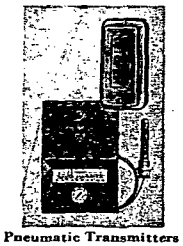


No. 200 Receiver Controller

Transmitters, Receiver Controllers, Gages

The Type HT Transmitter is available for room and duct installation. Ranges are 35 F to 65 F, 55 F to 85 F, 75 F to 105 F, and 95 F to 125 F. The Series 200 Remote Bulb Transmitter is available with bulbs for duct or pipe line insertion and with 15 ft. of capillary tubing. Standard ranges: −25 F to 135 F and 25 F to 250 F.

Series 200 Receiver-Controller receives pneumatic signals from a remotely located transmitter. It indicates the temperature at the remote location and also controls a valve or damper to maintain operation at the selected control point. It is furnished with adjustable sensitivity and with direct or reverse action. Standard indicating scales are −25 to +135 F and 25 F to 250 F.



Pneumatic Transmitters
Type HT above, Series 200 below



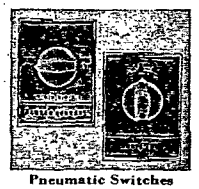
Temperature and Indicator Gages

Temperature Gages continuously show the temperature of important phases of an air-conditioning or process system. They are designed for central panel mounting remote from the transmitter. Diameters are 2½ in. and 3½ in.

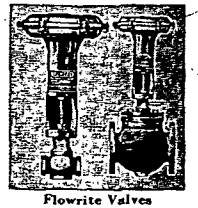
Powers Indicator Gages are used to show the reset range of a controller, the operation of electrical equipment or the position of a valve or damper motor.

Powers Pneumatic Switches and FLOWRITE Valves

Pneumatic Positioning Switches are designed for manual operation of actuators requiring smooth gradual control and also for readjustment of remotely located submaster instruments. They are 2⅜ in. wide by 3⅜ in. high and are compactly designed for panel board use. Powers Selector Switches are designed to deliver or stop or divert a flow of compressed air to selected control devices remotely located.

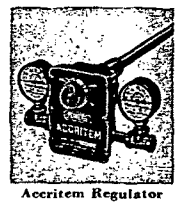


Pneumatic Switches



Flowrite Valves

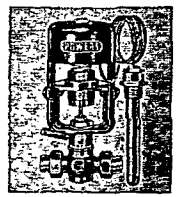
Powers FLOWRITE Valves are used to control the flow of water or steam. These valves are furnished with Powers #590 Teflon Graphited-Asbestos Packing. They are available in single or double seat and V-port types, direct or reverse acting — also in a three-way water mix type.



Accritem Regulator

Hot Water Supply Temperature Controls

Shown here are a few of Powers complete line of self operating and air operated controls for water heaters, heat exchangers, hot water line control, various types of baths and processes. All are described in POWERS WC CATALOG.

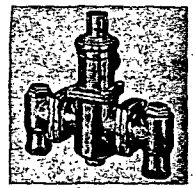


No. 11 Regulator

No. 11 Self Operating Regulator controls temperature of liquids or air; operating temperatures from 60 to 260°F; sizes ½ thru 4"; various types of valve bodies; available with 4" dial thermometer. Is widely used for water storage heaters; heat exchangers and processes.

ACCRITEM Regulator is air operated; controls temperature of liquids or air; temperature ranges 50 to 250°F and 150 to 350°F. Used for water storage heaters, instantaneous heaters, heat exchangers, submerged indirect heaters and processes.

HYDROGUARD Thermostatic Water Mixers automatically blend hot and cold water to deliver a mixture at exact temperature desired. Used for hot water line control, various types of baths, and processes. Models available for various capacities and temperature ranges.



Type N Hydroguard

Robertshaw-Fulton CONTROLS COMPANY

FULTON SYLPHON DIVISION
Knoxville 1, Tenn.
Sales Representatives in Principal Cities

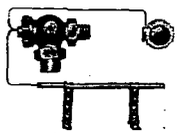
For additional information, write for Product Bulletin listed in each product description

SPACE HEATING CONTROLS



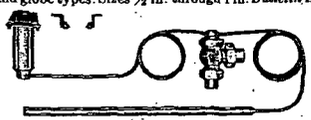
Automatic Radiator Valves #885 & #886

For exposed radiators and other heating units, either steam or hot water. Self-contained—no wiring, piping or auxiliary equipment. Self-operated—continuously positions valve to maintain desired temperature. Packless, non-leaking valve. Quickly installed—simply insert in place of regular radiator valve. Adjustable. Self-powered valves also available for concealed radiation units. Sizes ½ in. through 1 in. *Bulletin HVG-E.*



Automatic Radiator Valve For steam, #895

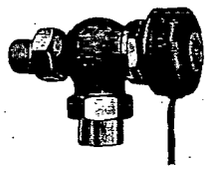
For any 2-pipe steam heating system of 15 lbs pressure or less. Suitable for convectors and enclosed radiators . . . only dial and adjustment knob are visible. Packless design. Two bulbs provide throttling action that assures very close temperature control. Minimum setting above freezing point. Control will lock at any adjustment to prevent tampering. Available in angle and globe types. Sizes ½ in. through 1 in. *Bulletin HVG-E.*



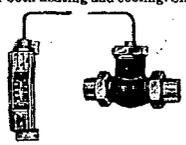
Radiator Valve #522-A1

For hot water heating systems. Positive, accurate and reliable control of the flow of hot water. Used primarily on radiators or convectors to control room temperature in heating. Can be applied to heating surfaces of induction units in some year-round air conditioning systems to control room temperature for both heating and cooling. Sizes ½ in. only. *Bulletin HVG-E.*

Radiator Valves for Steam, #890 & #891



An electrically operated valve for steam only. Suitable for pressures up to 15 lb. Used mainly for two-pipe steam heating systems . . . for exposed or concealed radiators and unit heaters . . . for control of individual rooms, zones, or entire buildings. A complete controlling unit includes one or more #890 (angle type) or #891 (globe type) valves, a wall thermostat and a simple transformer. Local or remote thermostatic control. Packless, no-leak valve. No wearing parts . . . nothing to lubricate. Sizes ½ in. through 1½ in. *Bulletin HVG-E.*



Temperature Regulator #7

For unit heaters, wall or ceiling type radiators, steam-heated duct systems. Self-powered through filled bellows thermostat system. Packless valve continuously regulates steam supply according to setting on ultra-modern wall thermostat. 20°F ranges up to 170°F. Pressures to 75 psig. Sizes ½ in. through 4 in. *Bulletin HVG-E.*

PACKLESS EXPANSION JOINTS



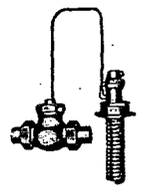
Sylphon® Packless Expansion Joints simplify building plans, save on the height of buildings, reduce cost of material and labor, and improve exterior and interior appearance. Successfully used in thousands of prominent buildings. Easy installation . . . no repacking. They are made possible by the Sylphon Bellows, which makes these joints as permanent as other structural parts of the building.



Low Pressure Type PACKLESS EXPANSION JOINT #1290

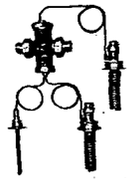
For steam or hot water lines. Simple, compact means of absorbing expansion of "baseboard" radiation, finned convectors, supply lines, etc. Emergency packing feature gives extra protection and lengthens life should bellows ever leak. Comes completely assembled. Traverse: ¾ in. Maximum working pressure: 70 psig. Sizes ¾ in. through 2 in. *Bulletin HVG-E.*

HEATING AND AIR CONDITIONING CONTROLS



Compensated Temperature Regulators #928 H1, -J1, -K1, -L1

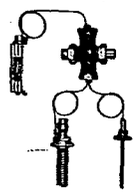
Low-cost single valve regulators that provide modulating control of heating and ventilating systems. Packless valve modulates steam flow according to dial setting on bulb. Control bulb automatically adjusts to prevent "droop" when constant discharge temperature is wanted. Self-operated. Steam pressures up to 75 psig. Temperature ranges of 60°F between 35°F and 170°F. Sizes ½ in. through 2 in. *Bulletin HVG-E.*



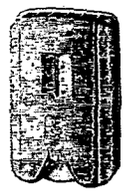
Regulators #889-D & #889-E

Actually two self-powered regulators in one, providing highly advanced modulating action for duct systems. One bulb serves to regulate room temperature . . . the other the low-limit discharge air temperature. A third bulb "compensates" for changes in demand, raising the *"low-limit" setting if outside* air temperature falls. Dial-type adjustment for ranges between 35°F and 170°F. *Bulletin HVG-E.*

Regulators #889-F and #889-G



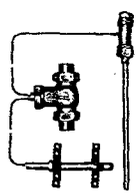
These regulators provide the same multiple control actions as #889-D and #889-E. However, room temperature is maintained through an adjustable wall thermostat instead of a thermostatic bulb. Steam pressures to 15 psig. Adjustable temperature ranges between 35°F and 170°F. *Bulletin HVG-E*



Room Thermostats #801 & 802

For heating, cooling or combination heating and cooling. Compact, attractive design. Available in Super Sensitive (#801) or Sensitive (#802) models. Suitable for direct control of many types of electric heating units, and for direct line voltage control of stokers, burners, motors, pumps, air conditioners, fans, etc. *Bulletin HVG-E*

INDOOR-OUTDOOR CONTROL FOR STEAM CONVERTORS



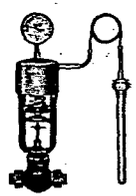
Regulator, #928-P1

This self-operated regulator varies hot water temperature as desired by regulating steam supply to steam-liquid convertors. Outdoor bulb automatically resets control bulb, and increases water temperature as outdoor temperature falls. For steam pressure to 15 psig. Dial-type control bulb adjustable over any specified 60°F range between 35°F and 170°F. *Bulletin HVG-E*

CONTROL FOR MISCELLANEOUS USES

Regulators #1003 Series

Completely self-powered, these sturdy units maintain hot water temperature by regulating flow of steam, to water heating units. Powerful, seamless Sylphon® Bellows actuates valve. Available with direct-or reverse-acting valve or 3-way valve. Adjustable temperature ranges between minus 35°F and plus 455°F. *Bulletin HVG-E*



PACKLESS EXPANSION JOINTS



High Pressure Type PACKLESS EXPANSION JOINT #1291

For steam or water lines—vertical or horizontal. *Internally* guided stainless steel multi-ply Sylphon Bellows. Completely assembled ready for easy installation. Traverse: 2 in. (also available to give 4 in. traverse). Maximum working pressure: steam 125 psig; water 150 psig. Sizes ¾ in. through 3 in. *Bulletin HVG-E.*



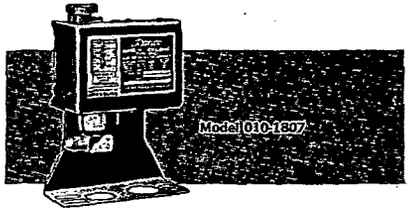
PACKLESS EXPANSION JOINTS #110-M and #111-M

These simple, rugged units absorb up to 1-¼ in. movement in horizontal or vertical lines. Two-ply Monel bellows will not corrode or leak at pressures up to 100 psig. No. 110-M, shown on opposite page, is for steam lines; No. 111-M is for water service. Sizes from ¾ in. through 3 in. *Bulletin HVG-E.*

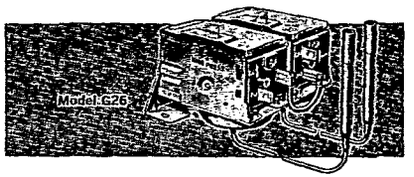
Ranco® INCORPORATED

COLUMBUS 1, OHIO

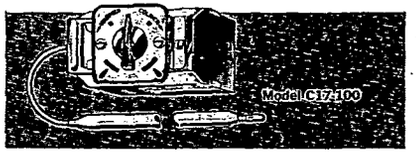
REFRIGERATION AND AIR CONDITIONING CONTROLS



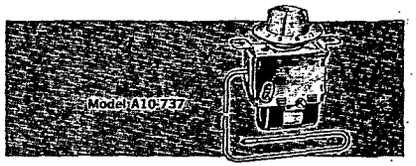
"O" type Commercial Refrigeration Controls, in Single or Dual Types adaptable to temperature or pressure operation; automatic cycle or manual reset. Single pole, single throw or single pole, double throw switches with front access terminals. Temperature range adjustment: −37°F to 110°F. Adjustable differential: 5° to 20°. Pressure range adjustment: 20 in. vac. to 450 psi. Adjustable differential: 5 to 25 psi and 30 to 100 psi.



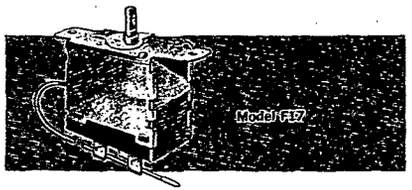
"G" type Commercial Air Conditioning and Refrigeration Controls for low and high pressure application, non-adjustable in the field. Small in size, readily adaptable to panels. Screw or quick-connect type terminals. With or without manual reset on either single or dual pressures in combinations. Electrical Ratings from Pilot Duty through 18 amp., full load.



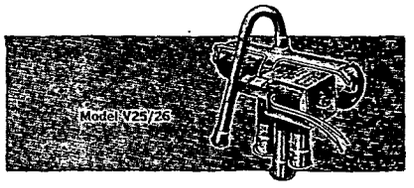
"C17" Air Conditioning Control for automatic changeover on heat pumps or two-stage cooling or heating operation that features two electrically separate switches operated by one temperature-sensitive power element. Liquid filled power element controls from the bulb. Switches are single pole, double throw with quick-connect type terminals.



"A10" type Refrigeration Controls for use on household refrigerators, freezers, beverage coolers, water coolers, window air conditioners, etc. Single pole, single throw switch. 15° to 35° dial range from −37°F to 95°F. Differential: 5° to 24°.



"F17" Air Conditioning Control is a cycling temperature type plus an auxiliary switch built-in for fan operation. Dial knob operates and controls both fan and unit. Switches are single pole, single throw. Maximum range of 25°F. Control differential: 4°F.



"V25/26" Instant Reversing Solenoid Valves for unit air conditioning heat pump systems to reverse the flow of refrigerant for either cooling or heating or both without an "off" period. Capacity of V25 is 2½ through 7½ tons; of V26 is 1 through 2 tons using Refrigerant 22.

Ranco Also Manufactures Motor Relays, De-Icing Controls, Clothes Dryer Controls, Automatic Defrost Timer Controls, Sequence Timers and Automobile Water Heater Modulating Valves. Write for descriptive literature.

SPENCE ENGINEERING COMPANY, INC.

Owners of Rider-Ericsson Engine Co., Founded by Capt. John Ericsson 1842
31 Grant St., Walden, N. Y.
Automatic Regulating Valves, Desuperheaters and Strainers
Offices in all Principal Cities

SPENCE AUTOMATIC REGULATING VALVES

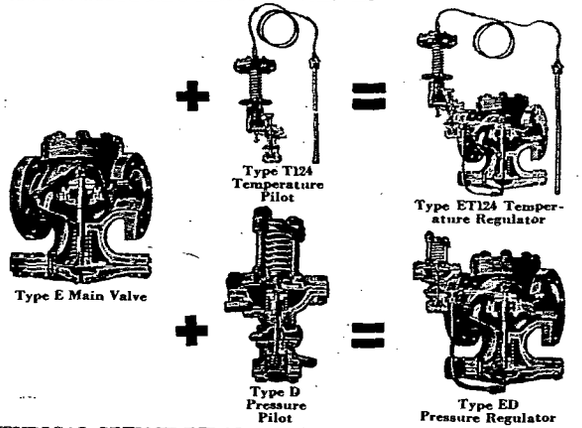
One or more separate, sensitive pilots control the main valve in Spence Regulators. These pilots are connected to the main valve by means of unions. This Spence feature offers the following advantages:

(1) **It's easier** to inspect and maintain Spence Regulators.

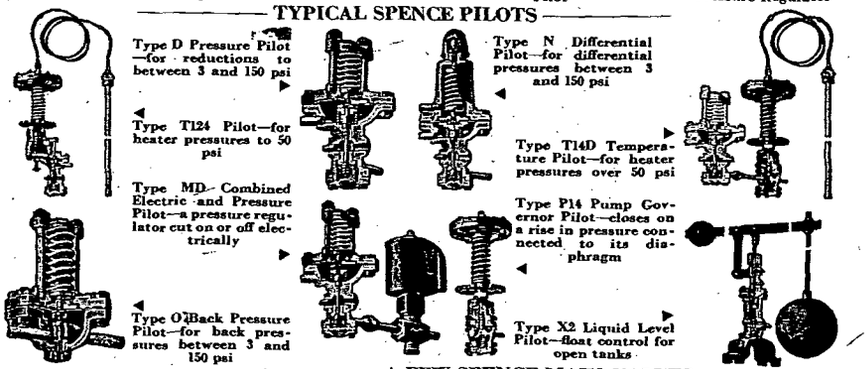
(2) **It's simple** to change the control function of a Spence Regulator by changing the pilot. All main valves, regardless of size, can be controlled by a wide variety of Spence pilots.

(3) **It assures** that you get the most effective and economical regulator for your needs because of the many possible combinations of main valves and pilots.

The illustration at right shows how a single main valve is combined with two pilots to form two different regulators.



TYPICAL SPENCE PILOTS



A FEW SPENCE MAIN VALVES



For accurate water pressure regulation—Type D24 Direct Acting Water Pressure Reducing Valve

Type E2 Main Valve for low initial pressures and low differentials

Type E3 Main Valve for high initial pressures and low differentials

Type C20 Main Valve same as Type E but double seated

WHITE-RODGERS
COMPANY

1209 Cass Ave., St. Louis 6, Mo.

In Canada, White-Rodgers Limited, Toronto

New York, N. Y. (Long Island City)
Buffalo, N. Y.
Atlanta, Ga.
Chicago, Ill.
Detroit, Mich.
Los Angeles, Calif.
Cleveland, Ohio
Philadelphia, Pa. (Havertown)
Columbus, Ohio

Distributors in Principal Cities

HYDRAULIC-ACTION CONTROLS

The complete line of White-Rodgers controls includes:

- Warm Air Controls
- Low Voltage Thermostats
- Line Voltage Thermostats
- Hot Water and Steam Controls
- Relays
- Gas Valves
- Oil Burner Controls
- Refrigeration and Air-Conditioning Controls

Hydraulic-Action temperature controls operate on the principle of expansion and contraction of a solid liquid charge against a diaphragm.
This system, developed and perfected by White-Rodgers provides accuracy comparable to that of a fine thermometer . . . and other outstanding features guaranteeing excellent performance.

ADVANTAGES OF HYDRAULIC-ACTION

Hydraulic-Action controls provide maximum sensitivity—positive dial settings—full range accuracy—consistent differentials—eliminates temperature drift—simplified installation—extra sturdy switch mechanism—and other qualities that assure long satisfactory operation.



"Fashion" Thermostat, low voltage, shown with hinged cover partly open. Pleasing shape blends with any room decor. Adobe Beige finish. Bi-metal operation.



Push Button subbase using "Fashion" thermostat to give finger-touch selection of heating, air-conditioning or off.

HOT WATER ZONE CONTROL PACKAGE

Low cost system which uses no flow valves and after first zone no additional relays or circulators are necessary. Low voltage wiring. Uses common return to save on piping, fittings etc.



Line voltage thermostat for unit heaters or air conditioning installations.



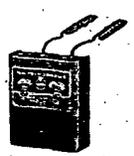
Zone Valve. Quite automatic water valve for inexpensive zoning of Motels, Hotels, Split-Level Homes, Ranch-type homes.

PRIMARY CONTROLS

In addition to Hydraulic-Action Controls, White-Rodgers manufactures Oil Burner primary controls, Gas Valves (both diaphragm and solenoid types) and automatic pilots for gas installations.



***Dual Hot Water* Controls. Limit and Circulator, High and Low Limit combinations. Also made in single controls.**



Fan and Limit controls in single case. Note the flexible elements. Also in single units.



Remote bulb heavy-duty thermostat, also available with self-contained bulbs.



Oil Burner primary control (stack switch). Intermittent or Constant Ignition types.



"Cushioned Power" Solenoid Gas Valves. Quiet, soft seat, spring loaded. for ALL gases.

Contact your nearest White-Rodgers office for complete information. Catalog and Engineering data furnished quickly.

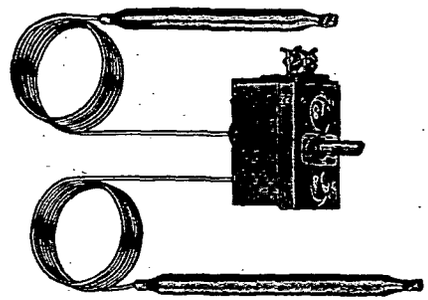
1001 Newark Ave., Elizabeth, N.J.
In Canada: Wilcolator (Canada) Ltd., 221 Evans Ave., Toronto 14, Ont.
Export Address: Wilcolator, 1010 Schaff Bldg., 1505 Race St., Philadelphia 2, Pa.

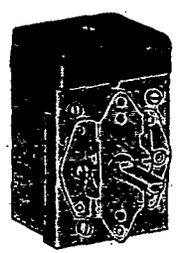
THE Wilcolator COMPANY

TYPE G2-E

New Effective-Temperature Control

The Wilcolator G2-E Effective-Temperature Control is a thermostat designed to sense both dry bulb and wet bulb temperatures and to combine these signals in such a way that the thermostat controls at any selected effective temperature. In application, a wick or sleeve surrounding the wet bulb cools it by evaporation of the water supplied from the evaporator. As humidity increases, evaporative cooling decreases, and temperature of the wet bulb rises. Hence an increase in humidity appears to the thermostat as an increase in temperature, just as it does to the human body. Volume of the wet bulb is so proportioned to volume of the dry bulb that the thermostat automatically corrects the controlled temperature to compensate for changes in humidity.

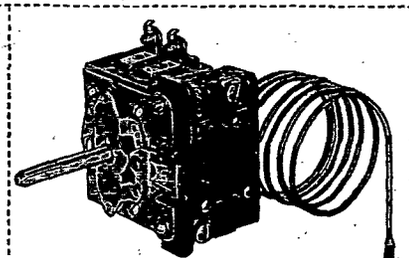




TYPE G

Thermostat with Positive "Off" or "Constant Cool"

The Wilcolator Type G liquid expansion electric thermostat features an improved, snap-action, vibration-proof control mechanism permitting much higher ratings than previously available in such a compact unit (2¾ x 1⅝ x 1⅞ in.). Suggested applications include room coolers, space heaters, and air conditioners. Ratings are the same as for Type GA. Standard temperature range for both G and GA controls is 0°F minimum; 585°F maximum. Special temperature ranges to customer's requirements.



TYPE GA

Thermostat with Multiple-Circuit Switch

The Wilcolator GA is a versatile new thermostat for both cooling and heating applications. Higher rated for new heavy duty appliances, it consists of a standard Type G thermostat, plus a special switch—cam operated by the dial shaft—permitting control of several circuits with a single dial. Contact rating, heater load (Type G1-A): 30 amp, 125 and 250 v a-c noninductive load. Cooler ratings (Type G2-A): 120, 208 and 240 v a-c; running current—14 amp; locked rotor—60 amp.

Field Control Division

H. D. Conkey & Company, Mendota, Ill.

Field Barometric Draft Controls

Field M+MG2: A Three-Fuel Control, in Sizes 10 in. through 32 in., for Oil, Oil-Gas, Gas, and Coal Fired Furnaces and Boilers.

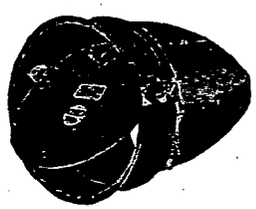
The **M+MG2** will serve gas, oil or coal with equal efficiency, and requires only minor, in-the-field changes to adjust from one fuel to another. A plant might convert from coal, to oil, to gas, then to oil-gas, without obsoleting this control. With coal it serves as a combination barometric draft control and check damper. With oil it serves as a single-acting barometric control. With gas it becomes a double-acting control, opening inward to regulate up-drafts, opening outward to relieve down-drafts. Where gas-oil is used interchangeably, the control change takes less than a minute.

The Field M+MG2 is a specialized, precision-made, three-fuel control. Because it is a control of extreme sensitivity, it is, of necessity, a control of extreme structural strength. For example, accuracy is increased when the clearance between a ring and gate is held to a minimum. To permit this, in the M+MG2, without risk of binding, Field developed a die-formed, heavy gage ring. Its deflection strength exceeds many fold the normal requirements of shipping, installation or use. The gate is also very heavy—39 pounds for a 32-inch control. Such weight obviously provides extreme rigidity. Heavy weight is also, paradoxically, a requirement of the virtually friction-free gate mounting that gives the M+MG2 its extreme sensitivity. Only a heavy gate could be so delicately balanced. The same weight factor also prevents the spurious responses to which so sensitive a gate might normally be subject.

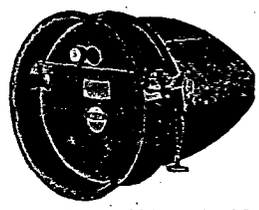
The M+MG2 gate is mounted on a long, thin, stainless steel knife edge, which in turn rests on self-aligning bearings. The bearings are self-cleaning. Soot or dirt cannot accumulate to cause binding or friction. They are also self-leveling—assuring that the knife edge is always in contact across the full width of the bearing, for an even distribution of weight. When the gate moves, only the sharp corner of the knife edge rests on the bearings—an area of contact so minute that friction is virtually non-existant. The result is extreme and instantaneous sensitivity to draft change. The gate is deeply recessed in the ring, to direct the flow of air over the gate and avoid response to random, variable air conditions outside the control.

The M+MG2 permits an unusually wide range of draft settings. The 18 in. control, for example, will maintain a draft setting as low as 0.002 in. The same control will maintain a draft as high as 0.3 in., a requirement where there is a very high draft loss through the appliance. Compensation for differences between horizontal and vertical installation is fully provided, by the simplest means available on any Field control. Where a safety switch must be specified, to meet gas code requirements, attachment is easily, effectively, and compactly made. Compactness is a characteristic of the M+MG2. There are no protruding parts. Even the stainless steel bead chain is short, holding setting weights neatly against the side of the control.

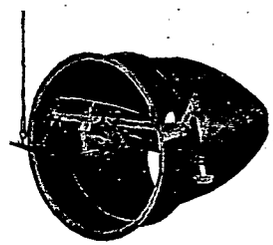
Unless otherwise specified the M+MG2 control is shipped as a single-acting control, with patterned collar. For gas, the installer will set control for double-action. If solid fuel is involved, order the M+MG2 Barochek. If control is to be shipped less collar, so specify.



M + MG2 Double Acting for Gas, opens inward to regulate up drafts, and opens outward to relieve internal pressures



M + MG2 Single Acting for Oil



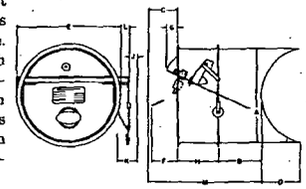
M + MG2 Barochek for Solid Fuels

PHYSICAL DATA

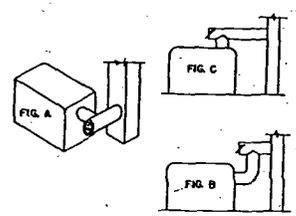
Control Size	Nominal Capacity Sq. in.	Recom. Pipe Sizes Dia.	Recom. Pipe Sizes Circum.	A	B	C	D	E	F	G	H	J	K	L	M	Shpg. Wt. Comp.	Shpg. Wt. L.C.	Vol. (Cu. Ft.) Comp.	Vol. (Cu. Ft.) L.C.
10	78.5	9 & 10	28¼ to 31½	10	4⅜	10	4¼	11	3⅛	1½	4¾	1	3½	1	19¾	9½ #	7½ #	.76	.48
12	113	11 & 12	34½ to 37¾	12	5¾	12	5	13¼	4⅝	1⅜	4¾	1	3½	⅛	22½	11½ #	8½ #	1.32	.70
14	154	13 & 14	40¾ to 44	14	7⅛	14	5¾	15½	5⅜	2⅛	5¼	1	3½	⅛	26¼	17 #	11 #	2.12	1.03
16	201	15 & 16	47 to 50¼	16	8¾	16	6½	17½	6¼	2¾	5⅝	1½	4	⅛	30	28 #	17¾ #	3.1	1.41
18	255	17 & 18	53½ to 56½	18	8¼	18	7½	19½	6⅞	2¾	7⅜	1½	5¼	1¼	33⅜	34 #	22½ #	3.98	2.09
20	314	19 to 21	59¾ to 66	20	9¾	20	8¼	22	7¾	3	7¾	1½	6¼	1¼	37⅜	42½ #	28¼ #	5.65	2.90
24	452	22 to 25	69 to 78½	24	12⅜	24	10½	26	9¼	3⅝	8⅝	2	6¼	1¼	45¼	65 #	44 #	7.18	4.35
28	616	26 to 30	81¾ to 94¼	28	13¼	28	*	30½	10⅝	4⅛	11⅜	2	6¼	⅜	52½	95 #	69 #	7.80	7.80
32	804	31 to 34	97½ to 107	32	15⅜	32	*	35	12¼	4¾	12⅝	2	6¼	⅛	59½	123 #	85 #	12.20	12.20

*No Points On These Collars

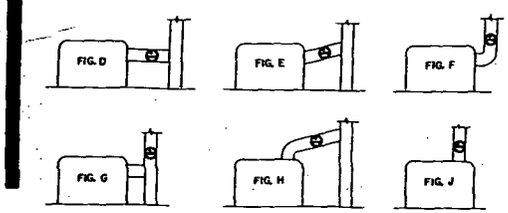
SWITCHES: A draft hood provides relief for down draft pressures, but gives no visual sign of spillage. The double-acting M+MG2 gate opens outward to relieve such pressures, and by its position signals any spillage. In addition, the M+MG2 can be equipped with an automatic switch where the engineer finds conditions which could create frequent and prolonged down drafts. A time delay switch is usually recommended for an atmospheric burner; a thermal switch for a power burner. These switches provide an obvious means to measure the extent of a down draft problem so that an engineer can judge if and when some remedial action is indicated.



BEST LOCATIONS FOR GAS



BEST LOCATIONS FOR OIL OR SOLID FUELS





FIELD TYPE M: For automatic heating equipment, designed to assure finer performance, greater fuel economy, through highly accurate control of drafts. Available in sizes from 6 in. through 9 in. for pipe diameters of 5 in. through 10 in. Features "Rocking Chair" gate action, off-center gate mounting, sidewings, extended housing. Widely used in the heating industry.



FIELD TYPE RC: Calibrated draft control in sizes 6 in. through 9 in. for 5 in. through 10 in. flues. Features extreme capacity, and reversible wing flap on gate for more precise adaption to either vertical, horizontal or sloping flues. Friction-free gate mounting, single strap attaching, easily leveled on a sloping flue.



FIELD MG1: Double-acting control for lower-input gas-fired furnaces and boilers; widely used in conversion burner installations. Opens outward to relieve down-drafts; opens inward to regulate up-drafts. Sizes 7 and 8 in. for 6 through 8 in. flues.

COMBUSTION CONTROL DIVISION

Electronics Corporation of America

One Memorial Drive • Cambridge 42, Mass.

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Shreveport, La.
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Tampa, Fla.
Tulsa, Okla.
Washington, D. C.

FIREYE SAFETY INTERLOCK SYSTEMS FOR SEMIAUTOMATICALLY AND MANUALLY-LIGHTED BURNERS.

Complete safeguard systems for up to four burners per boiler—oil, gas, or combination oil-gas. Assures safe light-off and firing. Includes System FP-4 electronic control in pre-wired cabinet assembly; plus all external valves, switches, and alarms.

FIREYE ELECTRONIC SAFEGUARDS AND PRIMARY CONTROLS.

System FP-2. Flame failure protection and programming for automatic industrial oil, gas, and combination oil-gas burners. *UL-FM* approved.

System FJ-2. Flame failure protection and programming for automatic commercial light oil, gas, and combination light oil-gas burners. *UL-FM* approved.

System FP-4. Flame failure protection for semiautomatically and manually-lighted oil, gas, and combination oil-gas burners. *UL-FM* approved.

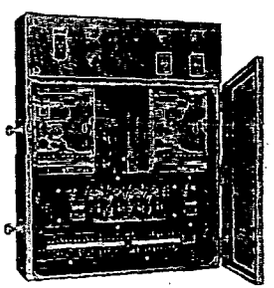
FIREYE VALVES AND INTERLOCKS.

Oil Valves. Manual reset or solenoid operated safety shutoff valves for all oil burner applications.

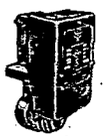
Gas Valves. Automatic or manual reset types, 1 in. to 6 in. IPS. Solenoid operated pilot gas valves, ¼ in. and ½ in. IPS. Supervisory *FM* Cock, manually operated, lubricated plug cock. Untamperable position indicating switch.

FIREYE PHOTOELECTRIC SMOKE DENSITY INDICATOR.

System FE-3. Provides low cost, *easy-to-install* smoke density indication with assured accurate readings. Helps keep smoke within legal limits. Saves fuel dollars. Chart recorders also available.



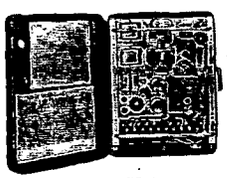
Safety Interlock System



Automatic Gas Valve



Manual Reset Valve



System FP-2



System FE-3



FM Supervisory Cock



Scanner



Light Source

SIMPLEX MANUFACTURING COMPANY

198-206 North Main Street **Fond du Lac, Wisconsin**

SIM-TROL Barometric Draft Controls—Types "A" "C" "F" "H-HG"

Breechings and stacks must provide sufficient draft under adverse atmospheric conditions. During normal and high barometric periods, cold months, and windy days, excessive draft intensities are generated with resulting inefficiency and fuel waste.

ADVANTAGES

SIM-TROLS automatically maintain the predetermined minimum draft required for good combustion, fuel economy, increased heat transfer, by reduction of gas velocity through the boiler or furnace bonnet, minimizing cold air infiltration and inrush causing sudden shrinkage of boiler parts. They eliminate local hot spots, floating and pulsating flame and sucking out of pilots. They improve feed water regulation, super-heater results, boiler-life and boiler-room ventilation.

SIM-TROLS aid materially in smoke abatement and boiler cleanliness by reduction of unburned combustibles and air dilution of stack gases. Lowered stack temperatures prolong liner life and reduce fire hazards.

CONSTRUCTION

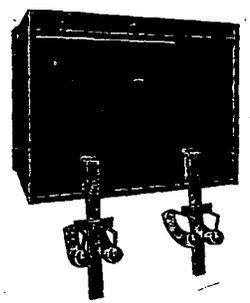
SIM-TROLS, properly sised, adjust easily to any desired draft intensity.

The races of the Type A SIM-TROL adjust laterally, the race assembly vertically, and the arm angle may be varied in relation to the plane of the gate.

The races of the Type C SIM-TROL adjust laterally, the assembly vertically, but the angle *in* relation to the gate is not variable.

Gates of Type A and Type C SIM-TROLS rotate on cold rolled arbors and prelubricated dust proof ball bearings. Curved tubular counter-balances contain metal balls which constantly change position to compensate for varying rotation angles of the gate, maintaining overfire draft within 0.01 in. water, plus or minus.

Gates of Type F SIM-TROLS rotate on cold rolled arbors and dust proof ball bearings in sizes from 12 in. x 12 in. upward. Adjustable weights hung on slide loops permit regulation along the curved slide arms.



Type "A" SIM-TROL

Gates of Type H-HG SIM-TROLS rotate on heavy gage black oil tempered wire extending into drilled holes in the metal frame of the control at an off-center horizontal point. Counterbalance members are similar to that of the Type F, except as to angle of application.

SIM-TROLS are protected by hard, heat resisting boiler room enamels. All counterbalance members are outside, free from corrosion and encrustment by products of combustion.

DESIGN FACTORS

Sising is important. The input opening should equal the stack area, plus 10 percent for each 35 ft of stack over 65 ft thereof. *Preference should be given to control width, rather than for height, for ease of adjustment and smooth gate rotation.*

Horizontal clearance from installation point approximates gate height plus 30 in. Over-all height approximates gate height plus 37 in. Over-all width approximates gate width plus 8 in. Designs for limited space may be made if plans are submitted.

SIM-TROLS *of types* A, C and F are individually designed to plant specifications, including breeching and stack dimensions, clearances, fuel, method of firing and number of boilers on the line. Detailed sketches are submitted for approval or suggested change without obligation.

SIM-TROLS—Types A, F and H-HG are designed to serve plants fired with solid, liquid or gaseous fuels under any method of firing. An adjustable "stop" is provided above the gate to allow dual gate action when used in connection with gaseous fuel, the stop being turned upward. For use with solid or liquid fuels the stop is turned downward effecting single action required for those fuels. Full capacity and efficiency are available for both draft control and relief of back pressure from any cause.

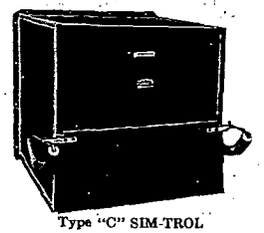
The Type C SIM-TROL is not adaptable to dual gate action, though it is efficient in connection with powered gas burners, or plants using induced draft equipment, or plants fired with solid or liquid fuels. We do not recommend the Type C SIM-TROL for plants burning gas and designed as purely atmospheric type.



Type "F" SIM-TROL



Type "H-HG" SIM-TROL



Type "C" SIM-TROL

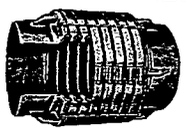
ADSCO DIVISION

Yuba Consolidated Industries, Inc.

20 MILBURN ST. BUFFALO 12, N. Y.

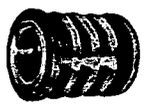
Yuba Plants and Sales Offices Nationwide

ADSCO CORRUFLEX PACKLESS EXPANSION JOINTS



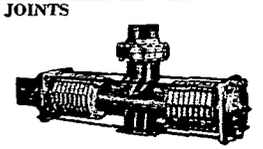
Equalizing, Welding Ends

Adsco's Corruflex packless expansion joints absorb pipe expansion and are adaptable to any piping condition. The equalizing joint is used for various types of motions and is suitable for pressures to 300 psi and higher. They are made in sizes ranging from 3 in. to 54 in., with single or multiple corrugations, with flanged as well as welding ends. They can be supplied with covers and telescoping inner sleeves.



Non-Equalizing, Flanged Ends

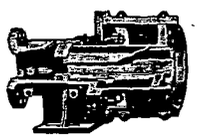
This type is generally used to absorb small amounts of expansion and to isolate vibration under vacuum or pressures to 50 psi. Depending on the metal used in the expansion element, both the equalizing and non-equalizing types are suitable for temperatures from sub zero to 1600 F. Adsco also manufactures a close-pitch type and a semi-equalizing type with root rings to reinforce the corrugations.



Special Types

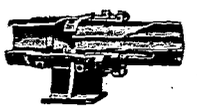
Special types of Adsco Corruflex expansion joints, like the pressure-balanced above, can be designed and manufactured to absorb axial, lateral, or angular rotation, or any combination of these movements. Some of these special types of expansion joints are the universal, hinge, swing, gimbal, tie-rod, and internally-guided. Consult Adsco for details on the performance of these special expansion joints in pipe lines.

ADSCO SLIP-TYPE EXPANSION JOINTS



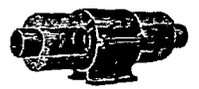
"PG" Joint, Flanged Ends

Slip-type expansion joints, designed for axial motion, offer long traverse, low initial cost, and produce minimum stresses on pipe anchors. The piston-ring joint, Type "PG", can be *unpacked and repacked under full line pressure* because of its piston-ring feature. The polished, chrome-plated slip cannot be scored by metal-to-metal contact. Packing can be quickly lubricated. In the externally-internally guided piston-ring joint the slip is fully supported at both ends throughout its circumference over the entire length of travel.



"P" Joint, Welding Ends

The internally-guided piston-ring joint, Type "P", being more compact, is especially recommended for use where space is limited. This unit, like the "PG" Type, can also be *unpacked and repacked under full line pressure*. Adsco slip joints are furnished in both single and double units with flanged ends or welding ends. Sizes range from 1¾ in. up, with traverse per slip from 4 in. to 12 in. Semi-steel joints can be used for working pressures to 250 lb and temperatures to 450 F. Steel joints can be used for working pressures to 400 lb and temperatures to 800 F.



Pipe Alignment Guides

Proper guiding of a pipe line, as obtained by Adsco pipe alignment guides, is essential to insure straight axial movement of the pipe to the expansion joint. Misalignment may damage equipment. A pipe line should be considered as a load-bearing column and pipe alignment guides should be used accordingly to prevent bowing and bending. Pipe supports such as hangers, chairs, or rollers are not to be considered as guides. Write for details on the latest designs in Adsco pipe alignment guides.

ADSCO STANDARD HEATERS



Storage Water Heaters

Adsco storage heaters are built in accordance with the *ASME* code for working pressures desired. Tanks are made of flange-quality steel, nickel-clad or stainless-clad steel, or copper silicon alloy. Where alloy tanks are used, tube sheets are of corrosion-resisting metal, so that service water does not come in contact with any ferrous parts. Steel tanks can be lined with all standard linings, including Adsco Phenolic 6-2 or Adscote bitumastic to prevent corrosion. Heating element is easily removed for cleaning.



Instantaneous Heaters

Adsco instantaneous heaters are made in a large variety of types, including the simple U-bend instantaneous shown above, which is used to heat water for showers, dishwashers, laundries, and other purposes. These heaters are also used as converters for furnishing hot water to space heating systems. Many straight-tube, float-head units are used as closed feed-water heaters, boiler blow-down heaters, and for various uses in the process industries.



Oil Pre-heaters

Adsco oil pre-heaters help obtain highly efficient performance from fuel oil by heating it properly for complete combustion in the burner. In many cases, the work done by oil pre-heaters permits the use of heavier grade, less costly fuel oil. These heaters frequently are used with Adsco tank suction heaters, the two units making an efficient team. Tank suction heaters, which heat only the liquid to be pumped and not the full contents of the tank, also are available.

Flexonics Corporation

EXPANSION JOINT DIVISION

1329 South Third Avenue — Maywood, Illinois

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District Offices

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In Canada: Flexonics Corporation of Canada, Ltd., Brampton, Ontario

FLEXONICS EXPANSION COMPENSATORS FOR STEAM AND HOT WATER HEATING SYSTEMS

Model L for pressures to 60 psig

The Flexonics Model L Expansion Compensator consists of a two ply phosphor bronze bellows with copper tube end connections enclosed in a protective shroud of brass. It is especially designed for control of expansion in finned type convectors, baseboard, radiator or heating supply and return lines.

Sizes from ¾ in. through 2 in. Suitable for temperatures from −60 F to 250 F and for pressures up to 60 psig. A single unit will handle motion up to ⅝ in. (½ in. compression, ⅛ in. extension). Available with all commonly used fittings.



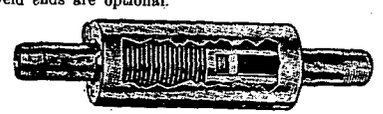
Sizes and Dimensions

Tube Size (in.)	Overall Length, in.		Max. O. D., in.	
	Sweat Ends	Female Threaded Ends	Sweat Ends	Female Threaded Ends
¾	6½	8 1/16	1⅛	1⅛
1	6 3/16	8 1/16	1⅜	1⅜
1¼	7	9⅛	2¼	2¼
1½	7 3/32	10 5/32	2½	2½
2	8¼	10½	3	3

Model H for pressures to 175 psig

The Flexonics Model H Expansion Compensator is used primarily on vertical risers for high pressure steam and hot water lines. However it may also be used on horizontal runs. The Model H consists of a corrosion resistant Flexonics Bellows enclosed in an all-welded steel housing. The unit is provided with a positive anti-torque device to prevent possible bellows damage during installation.

Sizes from ¾ in. through 3 in. Suitable for temperatures from −60 F to 750 F. Maximum test pressures 50 psi over working pressures shown in table. A single unit will handle motion up to 1¾ in. (1½ in. compression, ¼ in. extension). Mild steel male NPT pipe nipples on both ends are standard. Ground joint unions or weld ends are optional.

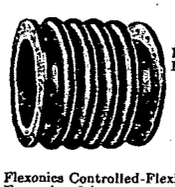


Sizes and Dimensions

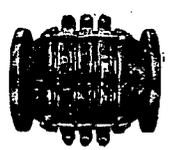
Pipe Size (in.)	O/A Length (in.)	Max. O. D. (in.)	Working Press (psi)	Unit Wt. (lb.)
¾	9 5/8	2⅜	175	2.6
1	9 5/8	2⅜	175	2.8
1¼	13 1/8	3½	150	7.7
1½	13 1/8	3½	150	8.0
2	12 3/4	4¼	125	10.3
2½	13 3/16	5	125	13.7
3	14 1/16	5¼	125	17.4

FLEXONICS EXPANSION JOINTS FOR POWER AND PROCESS PIPING

For medium and high pressure piping, Flexonics offers a complete line of corrugated expansion joints in free-flexing and controlled flexing types. Free-flexing units are available in standard sizes from 3 in. to 48 in. ID, for pressures to 30 psi, temperatures to 850 F. Designed for expansion travel up to 3¾ in. per unit. Controlled-flexing units are available in standard sizes from 3 in. to 48 in. ID, for pressures to 300 psi, temperatures to 850 F. Designed for expansion travel up to 7½ in. per unit. Both free-flexing and controlled flexing types are available with flanged or welding ends.



Flexonics Free-Flexing Expansion Joint



Flexonics Controlled-Flexing Expansion Joint

U. S. Flexible Metallic Tubing Co.

Since 1905

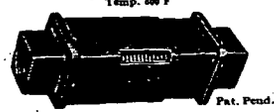
KEFLEX MANUFACTURING DIVISION

General Offices—63 Main St., San Francisco 5, Calif.

Factory—Los Angeles

Sales Offices—Agents in Principal Cities of United States and Canada

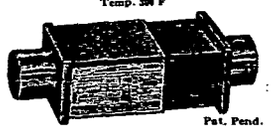
TYPE 7 Q WITH BRASS FEMALE THREADED ENDS
Working Pres. 300 PSI Maximum
Temp. 500 F



Pat. Pend.

Nom. I.P.S.	Overall Length	Max. O.D.	Wt. #
¾"-1.00"-1.25"	6.50"	2.00"	3
1.50	6.50	2.50	4
2.00	6.50	3.00	5
2.50	6.50	3.50	6
3.00	6.50	4.00	9

TYPE 7Q-T WITH COPPER TUBE ENDS
Maximum Pres. 200 PSI Maximum
Temp. 300 F



Pat. Pend.

To Fit Copper Tubing		Overall Length	Max. O.D.	Wt. #
Nominal I.D.	Actual O.D.			
¾"	⅞"	6.50"	2.00"	1½
1	1⅛	6.50	2.00	1½
1¼	1⅜	6.50	2.00	1½
1½	1⅝	9.00	2.50	2½
2	2⅛	10.00	3.00	4½
2½	2⅝	10.75	3.50	5½
3	3⅛	11.50	4.00	8½

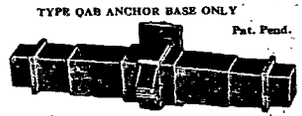
TYPE 11Q-T WITH COPPER TUBE ENDS
Maximum Pres. 100 PSI Maximum
Temp. 300 F



Pat. Pend.

To Fit Copper Tubing		Overall Length	Max. O.D.	Wt. #
Normal I.D.	Actual O.D.			
¾"	⅞"	4.50"	1.50"	1½
1	1⅛	4.50	1.50	1½
1¼	1⅜	4.50	1.75	1½
1½	1⅝	4.50	2.25	3
2	2⅛	4.50	2.75	4

TYPE QAB ANCHOR BASE ONLY



Pat. Pend.

Size	0.75"–1.00"–1.25"	1.50"	2.00"
C/L to Base	1.25	1.50	1.75
H—Total Depth or Height	3.38	4.13	4.63
L—Base Length	4.00	4.88	5.38
W—Base Width	2.00	2.00	3.00

KEFLEX QUADRA-SIDE EXPANSION COMPENSATORS

FOR STEAM AND HOT WATER HEATING SYSTEMS

KEFLEX Type 7 Q and 7Q-T packless type, expansion compensators are designed to compensate for both expansion and contraction in low or high temperature heating supply lines as well as for fin tube baseboard radiators and convectors.

Type 7 Q has square brass ends with female tapered pipe threads.

Type 7Q-T has female copper tube ends for soldering to copper pipe.

The bellows section, made from highly flexible, multi-ply (two or more plies), Type 321 or 347 stabilized stainless steel, is protected from torsion, squirm, misalignment, and external damage by a square, brass, telescoping case having positive stops against over compression.

The force to compress KEFLEX Quadra-Side Compensators is less than 100 lb; Standard units will accommodate a total traverse of 2 in.; 1½ in. in compression and ½ in. in extension. Units for traverse in compression of ¾ in., 2 in. and 3 in. are available with proportionate extension. Made in sizes ¾ in. through 3 in.

All KEFLEX Quadra-Side Compensators are provided with a removable screw, set for the indicated traverse in compression.

KEFLEX TYPE 11Q-T QUADRA-SIDE EXPANSION COMPENSATORS

USED WITH CONVECTOR AND FIN TUBE BASEBOARD RADIATION UNITS IN SCHOOLS, HOSPITALS, OFFICES AND COMMERCIAL INSTALLATIONS

KEFLEX Type 11Q-T, packless type, expansion compensators are designed for use in low pressure heating systems where maximum test pressures do not exceed 150 psig. They will compensate for expansion in compression up to ¾ in. and ¼ in. in extension—a total traverse to 1 in.

The bellows section, made from Type 321 or 347 stainless steel, is enclosed in a square brass case which slides over a square guide to provide positive protection against torsion, squirm and external damage during installation. Sizes: ¾ in., 1 in., 1¼ in., 1½ in. and 2 in. Maximum Working Pressure 100 psig. Maximum Temperature 300 F.

KEFLEX TYPE QAB INTERMEDIATE ANCHOR BRACKET

Type QAB is a heavy bronze bracket for use with KEFLEX Type 7 Q Dual Expansion Compensators as an Intermediate Anchor. The bracket is provided with two holes to facilitate attaching to floor, wall or ceiling. A set screw cinches the bracket tightly to the 7 Q unit and limits the movement to each individual compensator. Suggested for use in long runs of fin tube piping where lines should be segmented between the main anchors.

KEFLEX VIBRATION ABSORBERS

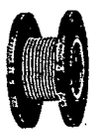
The high flexibility of KEFLEX Vibration Absorbers makes them extremely suitable to absorb vibration from pumps, chillers, and compressors. Keflex Vibration Absorbers are packless type as they have a bellows made from laminated stainless steel, type 321 or 347.

They may also be used as low pressure expansion joints (75 psig) where axial movements of ½ in. in compression and ⅛ in. in extension are desired. A lateral movement of ½ in. is also obtainable. The close face to face or overall length measurement permits their use in very close quarters where longer lengths of flexible hose would be impracticable.

KEFLEX Type 301 Vibration Absorbers are designed for higher pressure service than Type 151, or Type 71. Reinforcements have been added to withstand pressures over 150 psig. A liner is not standard equipment but is available if required. Dual units with long spacer nipple for lateral movement or correction of misalignment are furnished to specification.

Type 71-21V is fitted with carbon steel mechanical flanges having special elongated bolt holes on series 15 bolt circle. Keflex reattachable flanges may be applied jobsite in making emergency repairs.

Type 151 is fitted with either carbon steel weld ends or 150 lb type rigid flanges having special elongated bolt holes on a series 15 bolt circle to permit easy installation.
Type 301 is furnished with either carbon steel weld ends or 150 lb type lap joint flanges; 300 lb type flanges are also available.



71-21V



151-2V



151-12V



301-2V



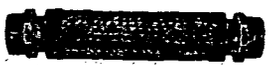
301-14V
301-4-14V (braided)

TYPE	71-21V	151-2V	151-12V	301-2V	301-14V 301-4-14V (braided)
Overall Length					
Sizes 2"-2¼"-3"	4.75"	11.00"	4.75"	11.00"	11.00"
Sizes 3½"-4" Inc.	4.75"	11.00"	4.75"	11.00"	11.00"
Max. Work Press. psig	75	150	150	300	300
Max. Temperature Fahr.	500	500	500	500	500
Traverse Compression	0.50"	0.50"	0.50"	0.50"	0.50"
Extension	0.125"	0.125"	0.125"	0.125"	0.125"
Lateral Deflection	0.125"	0.125"	0.125"	0.125"	0.125"

U.S. FLEX FLEXIBLE CONNECTORS

BRONZE METAL HOSE with High Tensile Bronze Braid

Install as near to pump as possible in each suction and discharge line for chilled water, hot water, steam, and force pump systems.



FLEXIBLE "BMH" Connectors



"USF" BRONZE VIBRATION ABSORBERS

FLEXIBLE "BMH" CONNECTORS
BMHN—IPT Nipples Brazed on.
BMHF—125 # Cast Iron Flanges Threaded on Nipples.
Suggested Lengths for Average Installations:

Hose I.D. (Inches)	Overall Length	Working Pressure PSI	Approx. Offset Motion	
			Intermit.	Permanent
½"	10"	750	¼"	1"
⅝"	11"	500	½"	1"
1"	12"	350	½"	1"
1¼"	13"	300	½"	1"
1½"	15"	250	½"	1"
2"	17"	150	½"	1"
2½"	20"	150	½"	1"
3"	22"	150	½"	1"
3½"	27"	150	½"	1"
4"	27"	150	½"	1"

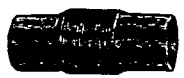
DAMPEN NOISE
PREVENT PIPE LEAKAGE
ABSORB VIBRATION
CORRECT MISALIGNMENT

"USF" BRONZE VIBRATION ABSORBERS WITH COPPER TUBE ENDS

To Fit Copper Tubing		Part Number Female Sweat Couplings	I.D. Flex Hose	Overall Unit Length	
Nominal I.D.	Actual O.D.				
½"	⅝"	USF-6	¼"	9¾"	Working Pressure and Motion same as BMH Flex Connectors
¾	⅞	USF-8	¾	11½	
1	1⅛	USF-9	1	13	
1¼	1⅜	USF-10	1¼	14¾	
1½	1⅝	USF-11	1½	17	
2	2⅛	USF-12	2	20	
2½	2⅝	USF-13	2½	24	
3	3⅛	USF-14	3	27	
3½	3⅝	USF-15	3½	32¾	
4	4⅛	USF-16	4	33¾	
5	5⅛	USF-17	5	39¾	
6	6⅛	USF-18	6	45	

KEFLEX EXPANSION JOINTS
Corrugated, Packless Type
FOR STEAM AND HOT WATER HEATING SYSTEMS AND PROCESS PIPING

KEFLEX packless type Expansion Joints are made from highly flexible laminations of Type 321 or 347 stabilized stainless steel. Standard equipment in both KEFLEX Types 158 and 308 consists of a packless type, stainless steel bellows; weld ends or carbon steel flanges; self guiding, full bore, flow liner; and self equalising contour control rings in sizes over 4 in., also available for smaller sizes. Type 308 also has reinforcing shoulder rings. Sizes 2 in. through 48 in. The force to compress KEFLEX Expansion Joints is usually less than 100 lb.



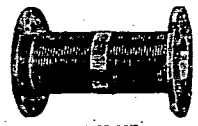
158-9E



158-15-9E



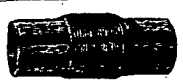
158-12E



158-15-12E

Rigid flanges supplied on standard Types 158-12E and 158-15-12E assemblies have a series 15 drilling, but with special elongated bolt holes to permit easy installation in the event companion flange bolt holes are misaligned.

TYPE	158-9E	158-15-9E	158-12E	158-15-12E
Overall Length				
Sizes 2"-2½"-3"	13.00"	19.75"	7.25"	14.00"
Sizes 3½"-48", Incl.	13.00"	19.25"	6.75"	13.00"
Maximum Working Pressure psig	150	150	150	150
Maximum Temperature Fahr.	800	800	800	800
Traverse Compression	2.00"	4.00"	2.00"	4.00"
Extension	0.50"	1.00"	0.50"	1.00"



308-9E



308-15-9E



308-14E

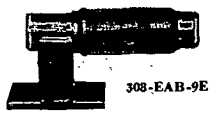


308-15-14E

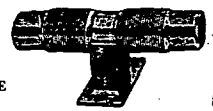
When flanges are required, Type 308 is fitted with lap joint or swivel flanges. 150 lb type is standard; 300 lb type and heavier are available.

TYPE	308-9E	308-15-9E	308-14E	308-15-14E
Overall Length				
Sizes 2"-2½"-3"	14.50"	22.00"	14.50"	22.00"
Sizes 3½"-24", Incl.	14.00"	22.00"	14.00"	22.00"
Maximum Working Pressure psig	300	300	300	300
Maximum Temperature Fahr.	800	800	800	800
Traverse Compression	2.00"	4.00"	2.00"	4.00"
Extension	0.50"	1.00"	0.50"	1.00"

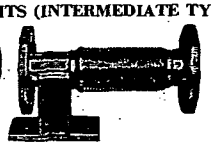
KEFLEX ANCHOR BASE UNITS (INTERMEDIATE TYPES)



308-EAB-9E



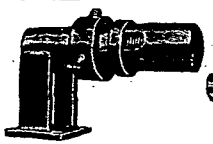
158-CAB-9E



158-EAB-12E



308-CAB-14E



KTO 300-EAB-2E
(Patent Pending)



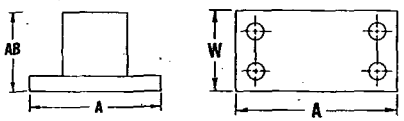
KTO 300CAB-2E
(Patent Pending)



KTO 300-15EAB-14E
(Patent Pending)



KTO 300-15CAB-14E
(Patent Pending)



ANCHOR BASE dimensions for standard Keflex Units are listed below.

Size	2.00	2.50	3.00	3.50	4.00	5.00	6.00	8.00	10.00	12.00	14.00	16.00	18.00	20.00	24.00
A—Base Len.	6.00	6.00	6.00	8.00	8.00	8.00	10.00	10.00	10.00	10.00	12.00	12.00	12.00	12.00	12.00
AB—C/L to Base	4.50	4.75	5.25	5.50	6.00	6.50	7.25	8.50	9.75	11.25	11.25	12.25	12.50	13.50	16.00
W—Width	3.38	3.88	4.50	5.00	5.50	6.68	7.68	9.68	11.75	13.75	15.00	17.00	19.00	21.00	25.00

KEFLEX-KETROL CONTROLLED EXPANSION JOINTS FOR EXTRA SAFETY
POSITIVE PROTECTION AGAINST TORSION. FOR USE IN PIPE LINES TO COMPENSATE FOR EXPANSION AND CONTRACTION
(Patent Pending)

KEFLEX-KETROL, self guiding, anti-torsion, packless type, expansion joints are manufactured with multi-ply, Type 321 or 347, stabilized stainless steel and enclosed by a special alloy steel shroud having two matched sections. Each section has at least four equally spaced finger-type projections at one end, which fit axially into the space between the finger projections of the companion section so as to serve as a positive guide to the bellows element and prevent torsion and deflection. The shroud cage also protects the bellows from exterior damage.

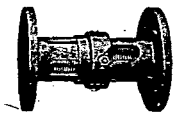
The lengths of the finger projections determine the full extent to which the bellows may be compressed, and also provide sufficient travel to accommodate extension of the bellows in the event ambient temperatures drop below normal and the pipe lines contract accordingly. Steel rings to prevent over extension of the bellows element are attached to the ends of the finger projections of each companion section. The rings also maintain complete concentricity of the matched shroud-cage and insure structural stability.



KTO-150-2E



KTO-150-15-2E



KTO-150-12E



KTO-150-15-12E

Rigid flanges supplied on standard Types KTO-150-12E and KTO-150-15-12E assemblies have a series 15 drilling, but with special elongated bolt holes to permit easy installation in the event companion flange bolt holes are misaligned.

STANDARD UNITS

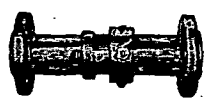
TYPE	KTO-150-2E	KTO-150-15-2E	KTO-150-12E	KTO-15-12E
Overall Length				
Sizes 2"-2½"-3"	13.50"	22.00"	13.50"	22.00"
Sizes 3½"-24", Incl.	13.00"	22.00"	13.00"	22.00"
Maximum Working Pressure lb	150	150	150	150
Maximum Temperature F	800	800	800	800
Traverse Compression	2.00"	4.00"	2.00"	4.00"
Extension	0.50"	1.00"	0.50"	1.00"



KTO-300-2E



KTO-300-15-2E



KTO-300-14E



KTO-300-15-14E

When flanges are required, KTO 300 are fitted with lap joint or swivel flanges. 150 lb type are standard; 300 lb type and heavier are available.

STANDARD UNITS

TYPE	KTO-300-2E	KTO-300-15-2E	KTO-300-14E	KTO-300-15-14E
Overall Length				
Sizes 2"-2½"-3"	17.50"	29.00"	17.50"	29.00"
Sizes 3½"-24", Incl.	17.00"	28.00"	17.00"	28.00"
Maximum Working Pressure lb	300	300	300	300
Maximum Temperature F	800	800	800	800
Traverse Compression	2.00"	4.00"	2.00"	4.00"
Extension	0.50"	1.00"	0.50"	1.00"

TYPE KTO 450

Pressures to 450 psig—Temperatures to 800 F.
For 2 in. Traverse in Compression and ½ in. in Extension
Type KTO-450-2E with Ends Beveled for Welding
Type KTO-450-14E with 300 lb, Type Lap Joint Flange
For 4 in. Traverse in Compression and 1 in. in Extension
Type KTO-450-15-2E with Ends Beveled for Welding
Type KTO-450-15-14E with 300 lb, Type Lap Joint Flange

NOTE: Type 450 measurements are essentially the same as type 300 listed above, except 450 standard flanges are series 30 (300-lb type)

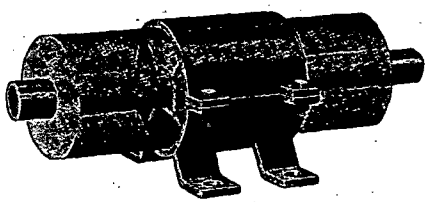
KEFLEX-MAVE TYPE M PIPE GUIDES

FOR CONCENTRIC GUIDING OF PIPE LINES

KEFLEX Type M pipe guides control the longitudinal movement of pipe and protect expansion joints from lateral deflection caused by bowing or bending of pipe lines. The fabricated steel shell and the inside clamp permit complete insulation of pipe at all points. Type M pipe guide is made of fabricated steel protected inside and out with a suitable finish. The body consists of a guiding cylinder with an anchor base welded on. The top half is removable for easy installation.

The pipe clamp is made in halves bolted together with the bore being ⅛ in. larger than the OD of the pipe. Set screws, in sizes 5 in. and larger, center the pipe and secure the clamp which moves through the body with the expansion and contraction of the pipe. The "ears" of the clamp ride on the inside surface of the cylinder.

Available for traverses of 3 in.-4½ in.-6 in.-9 in.-12 in., and for insulation thicknesses of 1 in.-1½ in.-2 in.-3 in.



KEFLEX-PULSCO TYPE P SHOCKTRAP FOR PIPE LINES, REFINERIES, CHEMICAL PLANTS, AIRCRAFT AND MARINE FUEL LOADING SYSTEMS

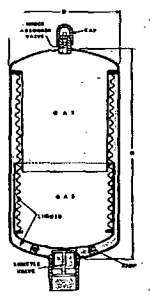
KEFLEX Type P Bellows shock traps are used to eliminate "water" hammer and reduce pressure surges in pipe lines operating under extremely high velocities, pressures, and hazardous "stop and go" flows which result in broken lines, meters and overstressed loading equipment, particularly where quick closing valves are used.

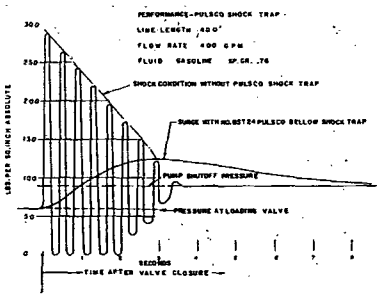
Type P is constructed of A-285 grade C steel enclosing a KEFLEX laminated Type 347 stainless steel bellows with a gas volume sealed in. The entrance to the shock trap is fitted with a special shuttle valve, consisting of an orifice restriction and a bleed hole for fluid exiting from the shock trap, and is guaranteed not to recycle. Surge resulting from the kinetic energy of the pipe line is absorbed by compressing the gas within the KEFLEX Bellows and by friction through the orifice in the shuttle valve.

The orifice in the shuttle valve is so sized as to provide the minimum gas volume for absorbing the surge. Kinetic energy in the pipeline is then converted into friction heat and into the work of compressing the gas in the KEFLEX Bellows gas chamber.

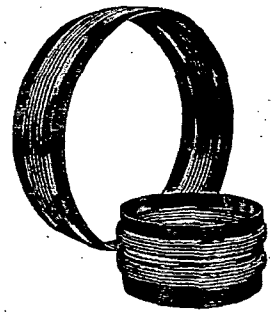
DIMENSION SHEET
PULSCO BELLOWS SHOCK TRAPS

MODEL NO.	CAPACITY cu. in.	CONN.	DIMENSION D	DIMENSION H
BST-6	200	1½	6	10
BST-8	400	2	8	13
BST-10	1130	2½	10¼	19
BST-16	3450	3	15	24
BST-20	5090	3	20	27¾
BST-24	9975	4	24	33¾





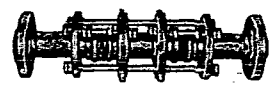
KEFLEX EXPANSION JOINTS



24 and 48 in. Type 151-2E Expansion Joints

KEFLEX Type 151-2E expansion joints in large sizes are particularly adaptable to ducting lines and wind tunnel installations as their high flexibility reduces anchor load requirements. The force to compress KEFLEX expansion joints is usually less than 100 lb.

The use of multiple laminations (or plies) of thin stainless steel, longitudinally welded, assures a long cyclic life for the unit and great resistance to pressures.



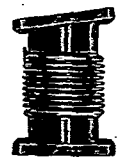
Type KST300-15-14E Dual Springtrol Expansion Joint

KST300-15-14E KEFLEX Springtrol assemblies are for use as controlled expansion joints where lateral deflection also occurs. Spring controlled tie rods support the long center section, between the bellows sections, and prevent over compression and extension.



Type KG300-14E Gimbal Type Expansion Joint

KG-300-14E KEFLEX Gimbal (hinged) joints are used when controlled lateral deflection is required and axial movement must be prevented or held to a minimum. The force to deflect KEFLEX joints is usually less than 100 lb.



Type 158-12 VSP for Diesel Exhaust

KEFLEX Type 158-12VSP Vibration absorbers are designed for use on diesel engine exhaust manifolds. They are usually fitted with a telescoping, self-guiding, full bore, liner to prevent direct impingement of flame on the bellows. They will absorb axial movement as well as the required lateral deflection. Flanges are furnished to match the drilling on the engine.

U.S. FLEX-KEFLEX EXPANSION JOINT MULTI-UNIT



Type 308-15CAB-14E. Traverse Provided for Job Requirement

MILITARY SPECIFICATIONS U. S. FLEX KEFLEX

Expansion Joints may be furnished to conform with Military Specifications MIL-E-17813a, Type I or II and Class 1 or Class 2.

THE BABCOCK & WILCOX CO.

TUBULAR PRODUCTS DIVISION · FITTINGS DEPT.

3839 WEST BURNHAM ST. MILWAUKEE 46, WISCONSIN

SEAMLESS WELDING FITTINGS AND FORGED STEEL FLANGES

DISTRICT OFFICES

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HOUSTON 19, TEXAS	2134 Welch St.	TULSA 3, OKLA.	427 S. Boston St.


90° Long Radius Elbow


180° Long Radius Return


Straight Tee


Concentric Reducer


Cap


Stub End


Welding Neck Flange


Slip-on Flange


Blind Flange

B&W Welding Fittings and Flanges are available in carbon, alloy and stainless steels in a wide range of sizes. Typical types and size ranges are listed below. For information on the complete line of B&W Seamless Welding Fittings and Flanges, write for B&W's Welding Fittings *Catalog FB-76*.

Type of fitting	Carbon and Alloy Steel				Stainless Steel			
	Std Weight	Extra Strong	Schedule 160	Double Extra Strong	Schedule 5S	Schedule 10S	Schedule 40S	Schedule 80S
45° Elbows—Long radius	⅛–36	¾–36	1–24	¾–8	⅛–24	⅛–24	⅛–24	¾–24
90° Elbows—Long radius	⅛–36	⅛–36	1–24	¾–8	⅛–24	⅛–24	⅛–24	⅛–24
90° Elbows—Reducing—Long Radius	2x1 to 12x10	2x1 to 12x10	—	—	—	—	—	—
90° Elbows—Short Radius	1–30	1¼–30	—	—	—	—	—	—
180° Returns—Long Radius	⅛–30	⅛–30	1–24	2–8	⅛–24	⅛–24	⅛–24	⅛–24
180° Returns—Short Radius	1–30	1¼–30	—	—	—	—	—	—
Tees—Straight	⅛–36	⅛–36	⅛–24	⅛–8	⅛–24	⅛–24	⅛–24	⅛–24
Tees—Reducing Outlet	½x½ to 30x26	½x½ to 30x26	¾x⅜ to 24x20	¾x½ to 8x6	½x½ to 24x20	½x½ to 24x20	½x½ to 24x20	½x½ to 24x20
Reducers—Concentric and Eccentric	1x¾ to 36x30	1x¾ to 36x30	1x¾ to 24x20	1x⅜ to 12x10	1x¾ to 24x20	1x¾ to 24x20	1x¾ to 24x20	1x¾ to 24x20
Caps	1–36	1–36	1–24	1–8	1–24	1–24	1–24	1–24
Stud Ends—Lap Joint	⅛–24	⅛–24	1–12	1–8	⅛–24	⅛–24	⅛–24	⅛–24
Laterals—Straight and Reducing	1–24	1–24	—	—	On Application			
Crosses—Straight and Reducing	⅛–24	⅛–24	—	—	On Application			
Swaged Nipples	On Application		—	—	—	—	—	—
Bull Plugs	On Application		—	—	—	—	—	—
Shaped Nipples 90°—45°	2–12	2–12	—	—	—	—	—	—
Saddles—Sleeves	1–30	—	—	—	—	—	—	—

Type of Flange	150 lb	300 lb	400 lb	600 lb	900 lb	1500 lb	2500 lb
Welding Neck	⅛–24	⅛–24	⅛–24	⅛–24	⅛–24	⅛–24	⅛–12
Slip-on	⅛–24	⅛–24	⅛–24	⅛–24	⅛–24	⅛–24	⅛–12
Lap Joint	⅛–24	⅛–24	⅛–24	⅛–24	⅛–24	⅛–24	⅛–12
Threaded	⅛–24	⅛–24	⅛–24	⅛–24	⅛–24	⅛–12	⅛–12
Blind	⅛–24	⅛–24	⅛–24	⅛–24	⅛–24	⅛–24	⅛–12
Socket Welding	¼–24	¼–4	—	¼–3½	—	—	—
Reducing . . . Slip-on and Threaded	¾x½–24x4	¾x½–24x4	¾x½–24x4	¾x½–24x4	¾x½–24x4	¾x½–12x3½	¾x½ – 12x3½

FA-8907

LADISH CO.

Cudahy (Milwaukee Suburb) Wisconsin

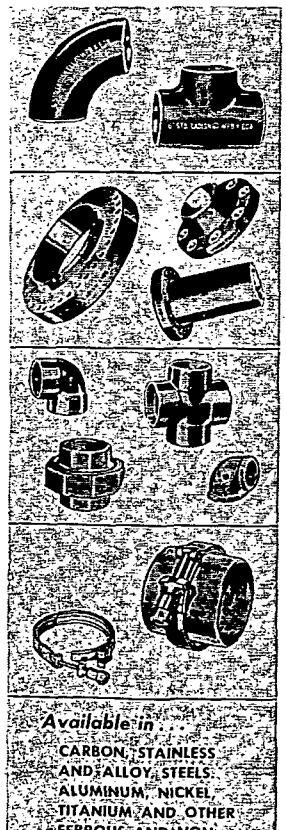
PLANTS IN: Cudahy, Milwaukee, Kenosha, Wis., Los Angeles, Calif., Houston, Texas, and Brantford, Ontario, Canada

TO MARK PROGRESS

LADISH SALES OFFICES:

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		ST. LOUIS, MO.	3615 Olive Street
		ST. PAUL, MINN.	2300 University Ave.
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		SAN FRANCISCO, CALIF.	420 Market Street
		SEATTLE, WASH.	314 Fairview Ave., N.
		TORONTO, ONT., CAN.	11 Yorkville Ave.
		TULSA, OKLA.	1502 1st Nat'l Bank and Trust Bldg.

LADISH . . . A COMPLETE LINE OF *Controlled Quality* PIPE FITTINGS



BUTT WELDING FITTINGS (IPS and Tube O.D.):

Sizes ½ in. through 42 in. Ranging from IPS Schedule 5S through double extra strong and special wall thicknesses through 5 in., plus complete range in Tube O.D. Fittings. These butt welding fittings align accurately for sound, fast welds to form an integral piping system of continuous efficiency. Smooth, flush, leak-proof joints reduce turbulence and pressure loss . . . and provide assurance of product uniformity.

FLANGES, LONG NECKS, ROLLED RINGS:

ASA, MSS, Light type and Corrosion Weight Flanges in sizes ½ in. through 24 in. —in pressure ratings 150 lbs through 2500 lbs; Large O.D. and T.E.M.A. Flanges in sizes up to 24 feet dia; Long Necks in sizes 1 in. through 24 in.—in pressure ratings 150 lbs through 2500 lbs; Rolled Rings, seamless or welded, are available in diameters up to 24 feet and in weights up to 140,000 lbs. All parts forged under exacting metallurgical controls, accurately machined to full dimensions.

SCREWED & SOCKET WELDING FITTINGS & UNIONS:

Forged Fittings in sizes ⅛ in. through 4 in.—in pressure ratings 2000 lbs through 6000 lbs. Sharp, clean threads of Ladish screwed fittings assure tight, leak-proof joints. Wide reinforcing bands provide ample wrench grip surface and extra strength at stress points. Socket Welding Fittings have deep, true welding sockets to assure proper alignment and slip fit. 150 lb STAINLESS STEEL SCREWED FITTINGS are also available in a complete range of sizes and types.

INDUSTRIAL TRI-CLAMP STAINLESS STEEL FITTINGS:

The industrial type Tri-Clamp Stainless Steel line presents a unique design concept in a dismountable fitting with particular application in process lines demanding contamination-free product flow. Available in sizes 1½ in. through 12 in., in IPS or Tube O.D., in type 304 and 316 stainless steel.

Tri-Clamp fittings make possible a light weight system with smooth inner walls . . . unobstructed flow . . . and no pockets and crevices to cause hazards of contamination. Simple assembly features . . . combined with light weight . . . assure fast, easy assembly and dismounting of lines for thorough cleaning, brushing and sterilization. Tri-Clamp line is assembled by either expand-on or weld-on methods.

A COMPLETE SERVICE TO MEET ANY REQUIREMENT:

Ladish offers complete service in pipe fittings regardless of type, size, pressure rating, wall thickness or material specification. Special emphasis on metallurgically sound materials and forging procedures . . . combined with advanced engineering . . . and rigid manufacturing controls assures users of Ladish fittings a uniformly high standard of dependability.

CATALOGS AVAILABLE

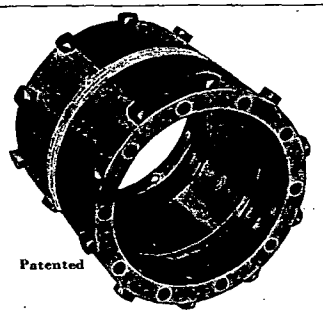
304 page Ladish Carbon and Alloy Fittings Catalog No. 55 . . . and 86 page Ladish Stainless Steel Fittings Catalog available on request. Contact your authorized Ladish Distributor, your local Ladish sales office, or write Ladish Co., Cudahy, Wisconsin.

THE PIPE LINE DEVELOPMENT COMPANY

5700 DETROIT AVENUE CLEVELAND 2, OHIO

TELEPHONE: ATLANTIC 1-3233

EXPORT OFFICE: R. S. STOKVIS & SONS, INC., 17 BATTERY PLACE, NEW YORK 4, N. Y.



Patented

WELD+ENDS SPECIFICATIONS

SIZE	Weight Pounds	OD Inches	Length Inches	Max. Working Pressure, psi (Prior to Welding)
2"	13	3⅜	6¼	2,000
3"	15	4¾	6¼	2,000
4"	28	6	8¼	2,000
6"	42	8⅝	9	2,000
8"	87	10¾	10	2,000
10"	99	12¾	10½	2,000
12"	115	14¾	10½	1,200
14"	190	15¾	14	1,200
16"	310	18¾	14	1,200
18"	361	21	16	1,200
20"	395	23	16	1,200
24"	433	25	16	1,200
24"	458	27	16	1,200
26"	561	26	16	1,000
30"	633	33	18	1,000
32"	939	36	18	1,000
34"	1173	38	18	1,000
36"	1408	40	18	1,000

Weld+Ends Permits Welding a Pipe Line While Pumping Through It

WELD+ENDS is used as a safety welding coupling on crude oil, gas, gasoline, propane, asphalt, tar, steam and other pipe lines. *As soon as packing rings* and clamping screws are tightened, pipe line flow is resumed. After pipe line is restored to service, WELD+ENDS is welded to pipe, making a solid, all-welded joint.

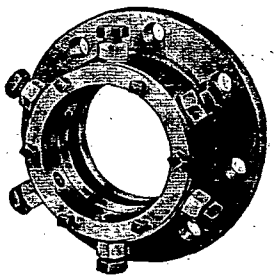
PLIDCOFLANGE

Plidcoflange is brand new. There are *countless* applications where this fitting excels. It slips over the end of the pipe, seals off like WELD+ENDS and mates up with standard flange facings.

Plidcoflange may be welded permanently to the pipeline while pumping or it may be bolted in place, permitting removal.

Plidcoflange is outstanding for use as a blind plate flange. Simply slide Plidcoflange back to change plates. Flange spreaders and jacks for springing pipeline are not needed.

Available in sizes 2 in. through 12 in. with 150 pound *A.S.A.* standard steel flanges. Other sizes and pressure classes on application.



New Smith+Clamp Permits Plugging and Welding Pit-Hole Leaks in a Pipe Line While Pumping Through It

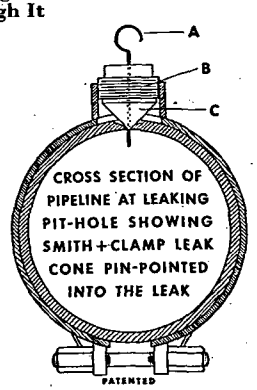
SMITH + CLAMP utilizes a unique pilot pin. (See "A" in illustration at right.) Pin projects from Neoprene leak cone (C) and guides point of cone directly into leaking hole. Pilot pin is removable.

After cone is pin-pointed on leak, pressure is applied directly behind cone by means of Force Screw (B), thus forcing cone point into hole and stopping leak. Makes it possible to control higher line pressures with lower bolting pressure.

SMITH + CLAMP is available in sizes 1 in. through 36 in. or larger.

WELD PLATES are also available in sizes from 4 in. through 36 in. WELD PLATES are companion pieces, required only where welded repairs are specified. They fit over the Clamp and weld to pipe, making a permanent, all-welded, steel plate repair.

Leak Cones, available in *various diameters*, are made of Neoprene in the standard models. Also available in Silicone (for higher temperatures), Teflon, Steel, Lead and other materials.



TUBE TURNS®

DIVISION OF CHEMETRON CORPORATION

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TULSA.....704–05 Wright Bldg.—Cherry 2-9193

In Canada: TUBE TURNS OF CANADA LIMITED, Ridgetown, Ontario
Toronto, Edmonton, Montreal & Vancouver
Distributors in Principal Cities

TUBE-TURN® *Welding Fittings and Flanges*

In sizes ½ in. through 42 in. . . . in all types . . . all wall thicknesses . . . and all piping materials

Tube Turns is the originator and pioneer manufacturer in the development and use of seamless, forged welding fittings. This company offers fittings and flanges in all sizes and types for every pipe welding need. A few of the many advantages of TUBE-TURN fittings are listed here. Complete details are included in the Tube Turns *Catalog 511*, sent on request.

Outstanding Advantages of TUBE-TURN Welding Fittings

These sound advantages are solid reasons why TUBE-TURN welding fittings are *universally preferred today*: (1) Stronger, lighter, tighter piping systems. (2) Permanently leak-proof, trouble-free construction. (3) *Uniform* wall thickness of elbows and returns throughout, making possible exact alignment for better, *faster, lower-cost* welding. (4) Minimum maintenance cost. (5) Easy sweeping changes in direction reduce pressure loss. (6) Smooth inner walls for more efficient flow, less corrosion and longer life. (7) Proper grain structure with no internal stresses. (8) Easier to insulate. (9) Accurate dimensions that permit prefabrication of whole sections.

Materials and Construction

Regular carbon steel TUBE-TURN elbows and returns, in both long and short radius types, are made in Standard, Extra Strong and (in long radius only) Schedule 160 and Double Extra Strong pipe thicknesses. These fittings conform to *ASTM* specifications A-234. TUBE-TURN welding fittings in other schedule numbers and special thicknesses are like*wise available. They are also made of* types 304, 347 and 316 and other stainless steels, carbon-moly and chrome-moly steels, copper, brass, nickel, Monel, Inconel, wrought iron and aluminum.

The uniform high quality of TUBE-TURN elbows and returns is due to their unique and original manufacturing process —in which short sections of seamless tubing are forced over a curved expanding mandrel at a forging temperature. TUBE-TURN tees are also made by an exclusive process and to a design that provides many special advantages. Other TUBE-TURN *fittings offer similar quality* standards.

Fine Grade Forged Steel Flanges

Tube Turns manufactures a complete line of forged steel flanges of outstanding strength and dimensional accuracy.

Valuable Technical Data Available

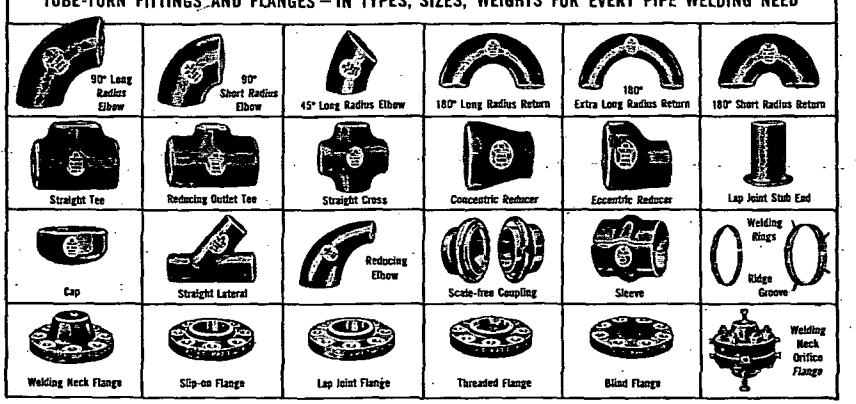
Several items of authoritative technical literature on fittings, flanges and piping in general are available as are bulletins giving specification information on types, sizes and materials. Write for descriptive leaflet.

Distributor Stocks Throughout U. S. and Canada

TUBE-TURN welding *fittings* and flanges are available from distributors located in every important city and area in the *United States and Canada.*

"tt" and "Tube-Turn" Reg. U. S. Pat. Off.

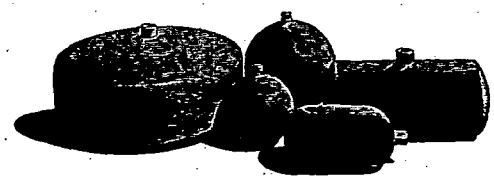
TUBE-TURN FITTINGS AND FLANGES — IN TYPES, SIZES, WEIGHTS FOR EVERY PIPE WELDING NEED



Arthur Harris & Co.

Dept. H. 210–218 N. Aberdeen Street Chicago 7, Ill.

Engineers—Metal Float Manufacturers



Harris Floats

Harris Floats are made in a wide range of sizes in ball, column, disk, conical, cylindrical and rectangular shapes, and can be made of virtually any workable metal for open tank or high pressure and temperature installations. Some of the metals used in the manufacture of Harris Floats are Copper, Copper Plated Steel, Brass, Aluminum, Admiralty, Everdur, Monel, Nickel, Inconel, Stainless Steel, types 304 and 316, etc., and Steel, depending on the corrosion factors of the installation and the pressures under which they must operate.

Harris Copper Ball Floats

Harris Copper Ball Floats are made of two separate bodies, soldered together and then heavily copper plated to provide a seamless surface. They are usually supplied with a hexagonal female spud although other special connections can be furnished. These seamless copper ball floats are carried in stock in 3 in., 4 in., 5 in., 6 in., 7 in., 8 in., 10 in. and 12 in. diameters . . . for 25 lb, 50 lb, 100 lb and 150 lb working pressures. Special sizes to 18 in. diameter and special high pressure copper floats are made to order.

Harris Stainless Steel Ball Floats

Harris Stainless Steel Ball Floats for low or high pressure and Corrosion Resistance are carried in stock in diameters of 2½ in. to 12 in. Sizes from 12 in. to 14 in. diameters are made to order.

All Harris Stainless Steel Floats are joined by Atomic Hydrogen welding which produces an unusually smooth, uniformly strong and ductile weld, free from oxides or pin holes.

Corrosion Resistant Floats

Floats that must operate in corrosive liquids can be made of any workable metal that is resistant to the specific liquid in which they are to be used.

Plated Floats

Floats can be plated with copper, nickel, chrome, zinc, tinned or lead coated where called for by the installation.

Float Connections

A variety of connection styles are available, suitable for practically any type of attachment, including a tube through the center.

Harris Float Catalog



A very complete catalog, with much technical engineering data is available to prospective float users on request. The Harris engineering staff will be glad to consult with you on any float problems.

BURGESS-MANNING COMPANY

Architectural Products Division

5970 Northwest Highway, Chicago 31, Ill.

Distributors in Principal Cities

A suspended metal pan ceiling that supplies:

1. **RADIANT PANEL HEATING**
2. **RADIANT PANEL COOLING**
3. **ACOUSTICAL CONTROL**

The radiant panel principle of this modern ceiling provides HEATING AND COOLING independently of air movement.

For the heating cycle, hot water is circulated through the pipe grids. The aluminum ceiling panels, which are clipped directly to the grids are heated by conduction; thereby making the entire BURGESS-MANNING FUNCTIONAL CEILING a radiator which in turn radiates to every surface and object in the room, nature's way of controlling human comfort.

For the cooling cycle, cold water (above the dew point) is circulated and the above described heating cycle is reversed.

Uniform air temperatures are maintained, convection drafts and heat shadows are minimized. Concentrated heat sources and overheated air are eliminated.

THE BURGESS-MANNING 3-WAY FUNCTIONAL CEILING CREATES A NEW STANDARD FOR COMFORTABLE WORKING AND LIVING CONDITIONS

The BURGESS-MANNING CEILING carries the *Entire Heating Load.*

The B-M CEILING carries the *Entire Sensible Cooling Load* (in most cases).

Air requirements are limited to latent heat removal and ventilation requirements.

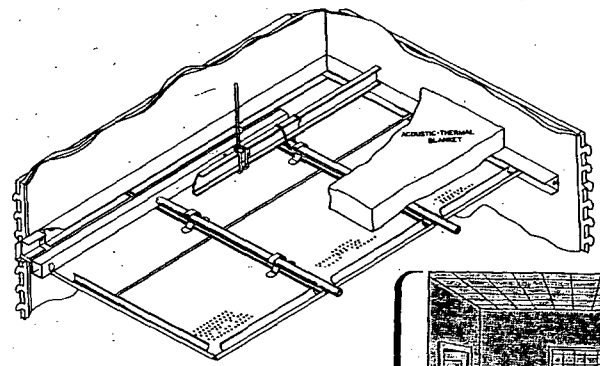
Reduced air handling equipment and smaller ducts SAVE BUILDING CUBAGE.

Heating and/or cooling source, entirely in ceiling, gives unrestricted use of the TOTAL FLOOR AREA.

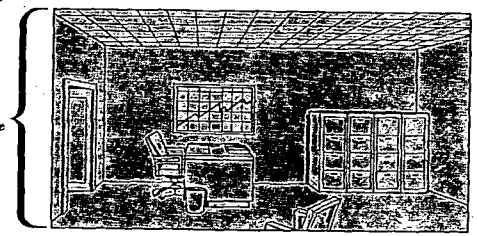
Standard hot water controls are utilized. Response to temperature change is instantaneous—*No Lag or Overrun.*

Simple modular system lends itself to flexibility in layout of partitions, lighting and air distribution.

For design procedure and performance curves send for *catalog A-138-2-G.*



Burgess-Manning Ceiling



Uniform Temperature

GENERAL AUTOMATIC PRODUCTS CORP.

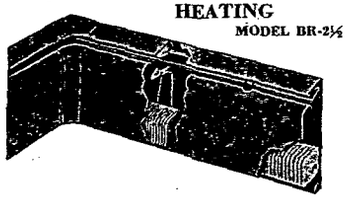
2300 Sinclair Lane Baltimore 13, Md.

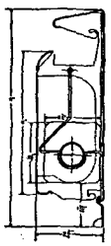
FLOORLEVEL BASEBOARD HEATING

Manufacturers of a Complete Line of Residential Gas and Oil Heating Equipment and Summer Air Conditioning

floorlevel® HOT WATER BASEBOARD HEATING

MODEL BR-2½ A





RATINGS Reg. U. S. Pat. Off	WATER Average Water Temperature, F	Btu/hr per Linear Foot 500 lb/hr	3000 lb/hr
Floorlevel BR-2½ A ¾" Copper Tubing Fins: Rectangular Aluminum 2¼" deep × 2½" high × *0.012, 54 fins per foot*	165	430	450
	170	470	500
	175	500	540
	180	530	570
	185	570	610
	190	600	640
	195	630	690
	200	670	720
	205	700	760
	210	730	800
	215	770	840
	220	800	870

Floorlevel Model BR-2½ A has been tested and approved by the *Institute of Boiler and Radiator Manufacturers* under the 3d edition of the I=B=R Rating Codes. Approval date June 10, 1958.

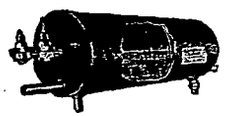
CARTON "A" *contains:* 10 ft of Floorlevel Baseboard complete with heating element and necessary brackets; 2 rubber grommets, 2 splice pieces. (Approximate shipping weight 28 lb.)

CARTON "B" contains: 10 ft of Floorlevel *Baseboard complete* with heating element and necessary brackets; 10 ft of extra enclosure only with necessary brackets, 4 splice pieces and 2 rubber grommets. (Approximate shipping weight 50 lb.)

CARTON "C" *contains: 10 ft of* Floorlevel Baseboard complete with heating element and necessary brackets; 3 inside corners, 1 *outside corner, 4 right hand corners,* 4 left-hand corners, 1 *wood* block (used where wood baseboard meets enclosure), 2 rubber grommets, 2 splice pieces. (Approximate shipping weight 33 lb.)

CARTON "D" contains: two 10 ft sections of Floorlevel *Baseboard* complete with heating element and necessary brackets; 4 rubber grommets, 4 splice pieces. (Approximate shipping weight 55 lb.)

CARTON "E" contains: two *10 ft lengths* of front and back panel, 10 brackets, 2 top moulding splice pieces, 2 front panel splice pieces. (Approximate shipping weight 44 lb.)



DUAL DE-AIRATOR TANK

A complete unit with feed and relief valve, for positive elimination of air in hot water heating systems; also acts as an expansion tank. Three sizes—500, 1,000 and 1,500 sq. ft.

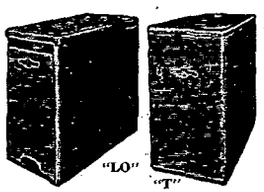
SERIES "LG" GAS-FIRED BOILER-BURNER UNIT



Available for natural, manufactured, mixed or L.P. gas. All units only 31 in. high, completely assembled. Controls available to meet all local code requirements.

SPECIFICATIONS:

Model	A.G.A. Ratings Btu Input	Btu Output	Sq Ft Output
LG-4 to LG-10	84,000 to 210,000	67,200 to 168,000	448 to 1120



"LO" "T"

OIL-FIRED BOILER BURNER UNITS

"LO" is pre-wired and pre-assembled, complete with burner, 3 controls and circulator. Available in 2 sizes:

Model No.	Btu Input	Btu Output	Sq Ft Output
LO-6	140,000	105,000	560
LO-7	168,000	126,000	672

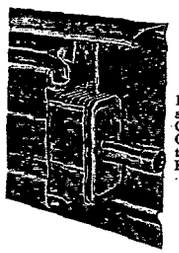
Series "T" Steel water-tube Boiler-Burner unit, constructed to *ASME* Code. Available in following sizes and ratings:

Model No.	Boiler Ratings EDR Hot Water Sq Ft	Steam Sq Ft
T-16 to T-40	580 to 4800	220 to 2600

Also manufacturers of a complete line of summer and winter air-conditioning equipment.

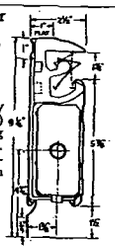
KRITZER RADIANT COILS, Inc.

325 NORTH RIVER STREET · BATAVIA, ILLINOIS



"K" LINE Radiant Baseboard for RESIDENTIAL, COMMERCIAL, INDUSTRIAL and INSTITUTIONAL HEATING

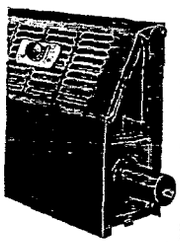
Quick and easy to install with many KRITZER features (at no extra cost) such as Coil Support Brackets, Sliding Coil Shoes, and Thermovane Damper. Capacities shown below are representative of the wide selection available with KRITZER Radiant Baseboard.



Capacities per linear foot. Air at 65F

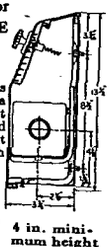
Element Symbol	Cover Symbol	1 psi STEAM Sq Ft per Lin. Ft	Btu per Lin. Ft	200F Water (1 gpm)
375-40A	E-1	3.50	840	645
21-40A	KU-1	4.40	1055	820
21-40	KU-1	2.90	935	720
SBC-40	KS-1	—	—	780

Get BULLETIN 203 for complete information



STB Radiant Baseboard for SCHOOLS, HOSPITALS, OFFICE BUILDINGS, INSTITUTIONS, and FACTORIES

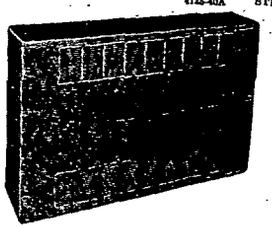
Sturdy. Lifetime of efficient service. Has such KRITZER features (at no extra cost) as Adjustable Coil Support Brackets, Coil Shoe, and Damper and Control. Capacities shown below are but a few of the wide selection available with KRITZER STB Radiant Baseboard.



Capacities per linear foot. Air at 65F

Element Symbol	Cover Symbol	1 psi STEAM Sq Ft per Lin. Ft	Btu per Lin. Ft	200F Water (3 gpm)
21-40	STB2-1	5.1	1225	970
4125-40	STB4-1	6.7	1610	1220
42-40	STB4-1	6.3	1510	1090
4125-40A	STB4-1	7.9	1895	1445

4 in. minimum height from floor Write for complete information





Vertical



Horizontal

Cabinet Unit Heaters

Quiet, economical units used in areas calling for rapid concentration of heat such as meeting halls, churches, and large public recreation rooms.

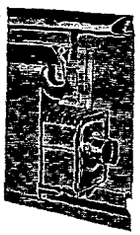
- Six basic sizes. Standard or high capacity coils—500 to 1200 cfm. Up to 430 EDR.
- Floor, wall, inverted wall, or ceiling mounted.
- Only 32 in. high and 10 in. deep.
- 1050 rpm motors with permanently lubricated bearings.
- Trim strips, 3 or 4 sides, for full or partial recessing.
- 4 speed control available.
- All piping can be kept inside of cabinets.

Send for *BULLETIN 1200-T*

Unit Heaters

All units have dynamically balanced, propeller type fans in deep venturi for quiet, efficient operation. Rubber mounted motors. Heavy gage steel cabinets. Baked-on enamel finish. Wide selection of sizes for steam or hot water. Write for *BULLETIN 800.*

WE WELCOME YOUR INQUIRIES ON SPECIAL APPLICATIONS

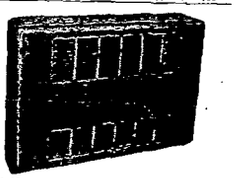


"SPECIAL" Radiant Baseboard for HOMES, APARTMENTS, HOTELS, MOTELS, OFFICES

Low cost baseboard with many KRITZER quality features: Thermovane *Damper, snap-in Coil Support Brackets,* Sliding Coil Shoes, Built-in Air Diverter, packaged in convenient lengths. Capacities *shown below* are representative of wide selection available. *BULLETIN 350.*

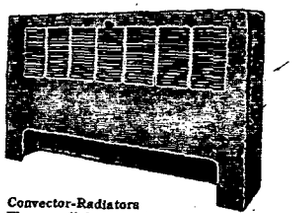
Capacities per linear foot. Air at 65F

Element Symbol	Cover Symbol	1 psi steam Sq ft per lin ft	Btu per lin ft	200F water (1 gpm)
175-50	SP	2.7	650	550
175-50A	SP	3.4	815	655
175-50	SPU	3.1	740	600
175-50A	SPU	3.8	900	730



Dual-Vectors

Fan-coil units for heating and cooling with hot or chilled water. Two models: Free standing (DVF), and Semi-recessed (DV) *which fits between standard studs.* Optional outside air intake and dual room conditioning accessories. Sizes in 200, 300, 400, and 600 cfm. Write for *BULLETIN 1001-T.*



Convector-Radiators

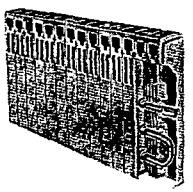
Floor, wall hung and recessed models. Famous KRITZER coil construction for steam and *hot water.* Bronze headers. Heavy, reinforced steel cabinets. Full range of sizes. Chain or knob operated dampers. Cabinets phosphatized. Gray prime baked-on finish. Write for *BULLETINS 600 and 601.*

Shaw-Perkins Manufacturing Company

201 East Carson St. Pittsburgh 19, Pa.

Manufacturers of SHAW Panel-Vectors delivering AIR-e-ATED Radiant Heat

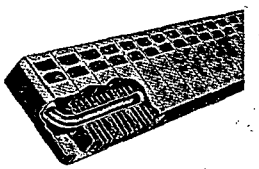
All Shaw Panel-Vectors and Perkins AIRadiators are complete one-piece factory assembled units ready for installation. Manufactured of heavy gage steel with full-length heating fins and vertical serpentine copper heating coils, pressure fitted for excellent heat transfer. Numerous sizes and models available to meet any area heat loss. Institutional models possess perforated paneling for low surface temperatures. Perkins AIRadiators are designed for industrial installations, and are readily mounted on either ceilings or walls.



SHAW Panel-Vectors (front or top air outlets)

DEPTH, front to back—3 in. **HEIGHTS**—8 in., 11 in., 14 in., 17 in., 20 in., 23 in., 26 in. **LENGTHS**—11 in. to 111 in. inclusive, in 2 in. increments. **TAPPING**—supply and return same end, right or left hand; also furnished on opposite ends. **PIPING**—connections for ½ in., ¾ in., or 1 in. piping. **RATINGS**—3.9 **EDR** to 117.7 **EDR,** based on 1 lb steam and 65° F inlet air. **INSTALLATION**—wall hung, free standing, open or closed recess. **APPLICATION**—hot water, high temperature hot water, steam to 150 psi. End panels, air chambers, wall brackets available for any application.

SHAW BASEBOARD Panel-Vectors



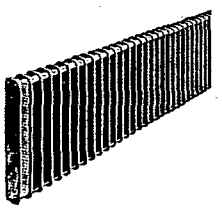
DEPTH, front to back—3 in. **HEIGHT**—8 in. **LENGTHS**—2 ft 5 in. to 72 ft 7 in. inclusive, in 2 in. increments. **TAPPING**—supply and return same end, right or left hand; also furnished on opposite ends. **PIPING**—connections for ¾ in. piping on same end, 1 in. piping on opposite ends. **RATINGS**—6.7 **EDR** per linear ft based on 1 lb steam and 65° F inlet air. **INSTALLATION**—wall hung with inside and outside corner runs. **APPLICATION**—hot water, steam to 150 psi. End panels, air chambers, and mounting screws included as standard equipment.



SHAW INSTITUTIONAL Panel-Vectors

DEPTH, front to back—3 in. **HEIGHTS**—8 in., 11 in., 14 in., 17 in., 20 in., 23 in., 26 in. **LENGTHS**—11 in. to 111 in. inclusive, in 2 in. increments. Steel housing has ⅛ in. diam. perforations on 3/16 in. staggered centers. **TAPPING**—supply and return same end, right or left hand; also furnished on opposite ends. **PIPING**—connections for ½ in., ¾ in., or 1 in. piping. **RATINGS**—5.0 **EDR** to 100.9 **EDR** based on 1 lb steam and 65° F inlet air. **INSTALLATION**—flush mounting for violent areas, offset mounting for non-violent areas. **APPLICATION**—hot water, high temperature hot water, steam to 150 psi. End panels for complete pipe covering.

PERKINS AIRadiator



DEPTH, front to back—5 in. **HEIGHTS**—14½ in., 23½ in., 32½ in. **LENGTHS**—15 in. to 125 in. inclusive, in 2½ in. increments. **TAPPING**—supply and return on opposite ends; top and bottom arrangement reversible by reversing radiators. **PIPING**—connections for 1 in. piping; may be bushed to ½ in. or ¾ in. as specified. **RATINGS**—14 **EDR** to 187 **EDR** based on 1 lb steam and 65° F inlet air. **INSTALLATION**—ceiling or wall hung. **APPLICATION**—hot water, high temperature hot water, steam to 150 psi. Ceiling hangers or wall brackets available for mounting.

Represented in principal cities. Send for catalog.

AIRFLOOR Company of California, Inc.

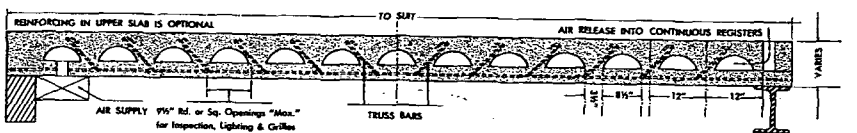
13729 E. Rosecrans Avenue, Sante Fe Springs, California

HOLLOW CONCRETE FLOOR PANEL FOR SINGLE AND MULTI-STORY CONSTRUCTION

Warm Air Radiant Heating and Ventilating or Complete Air Conditioning for Schools, Churches, Residences, Hospitals, Industrial Buildings



Airfloor forms create a strong, hollow floor serving as a common plenum for conditioned air, radiant heat, and positive mechanical ventilation for complete air-conditioning systems. Airfloor introduces all air passing through the floor into the occupied space. For use with Gas, Oil, Electric, or Coal Fired Furnaces, fan-coil units, multi-zone units, etc.

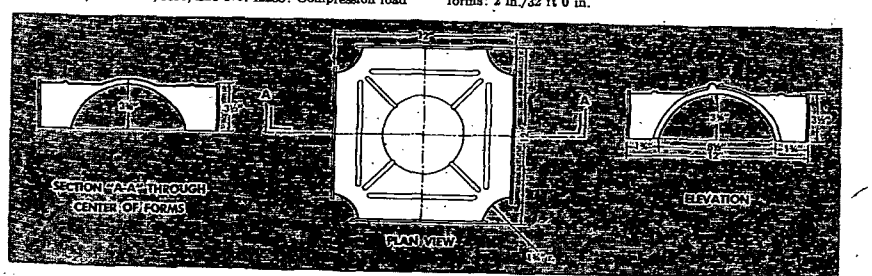


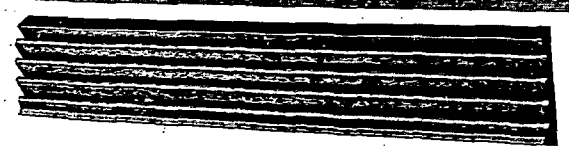
AIRFLOOR Truss Slab Provides:

1. A one-way or two-way truss within the slab.
2. A flat slab construction, which may be poured in place or lifted in place allowing flexibility of partition locations.
3. A ceiling already in place.
4. A reduced load of 35 pounds per square foot which in turn reduces concrete weight, supporting column, beam and foundation sizes.
5. A considerable reduction in seismic forces due to floor dead loads.
6. A plenum or duct through which conditioned air passes thus providing a radiant panel and simultaneous convection heating or cooling.
7. More utilizable volume in multi-story buildings, by eliminating required space for exposed ductwork at each floor level. This is especially important where limit-height buildings are concerned.
8. The ease of installation of hidden piping and electrical work. The necessity for rigid electric conduit is eliminated.
9. Flexibility in partition locations.

AIRFLOOR FORM SPECIFICATIONS

STANDARD MATERIAL: 26 gauge cold rolled steel (galvanized steel forms optional at extra cost).
STANDARD WEIGHT: One pound per sq ft.
SIZE: 12 in. x 12 in. x 3¾ in.
CONCRETE DISPLACEMENT: The concrete displaced by one Airfloor form is 0.223 cu ft.
COMPRESSION LOAD TEST: *By California Testing Laboratories, Inc.*, on June 8, 1955, Lab No. 42288: Compression load was applied to a six inch diameter (0.196 square foot) disc which centered over the one inch thickness of concrete at the center of the form.
The following tabulation gives the results of this test:
COMPRESSION LOAD IN POUNDS: 9,250.
RESULT: No cracks appeared until failure which fractured the concrete by punching shear at the edge of the disc.
FIELD CONDITIONS: Linear gain over installed Airfloor forms: 2 in./32 ft 0 in.





AIRFLOOR REGISTER SPECIFICATIONS.

Material: 20 gage cold rolled steel.
Length: 48 in. standard or cut to suit architectural requirements.
Height: 6¼".
Backplate & Dampers: Optional.
Finish: Canterbury Blue metal primer.

WRITE FOR COMPLETE CATALOG AND SPECIFICATIONS

National Clay Pipe Manufacturers, Inc.

1820 "N" St., N. W., Washington 6, D. C.

311 High Long Bldg., 5 E. Long St., Columbus 15, Ohio
703 Ninth and Hill Bldg., Los Angeles 5, Calif.
100 N. LaSalle St., Rm 2100, Chicago 12, Ill.
Room 406, 1401 Peachtree St., N.E., Atlanta 9, Ga.

CLAY PIPE HEATING, VENTILATING, AND AIR CONDITIONING DUCTS

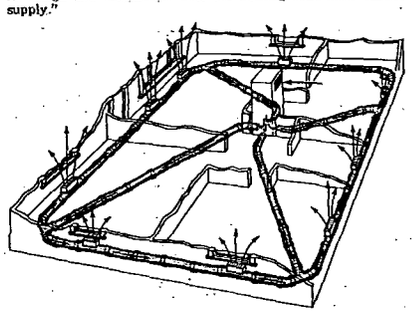
Information contained here includes the basic data, needed by *engineers, architects, builders,* and heating contractors to evaluate and install low-cost Vitrified Clay Heating Ducts in basementless slab-floor homes and other structures. This new development has become so popular that many requests for authoritative information have followed the publication of articles about the new system in the trade press. There are three major reasons for its popularity:

ECONOMY—The complete Clay Pipe installation is economical and efficient. It combines the advantages of radiant heat and controlled warm-air circulation at greatly reduced costs.

SERVICE—Clay Pipe is chemically inert and cannot be eaten away by lime in concrete. It does not rust, corrode, or disintegrate from moisture and furnace gases. It will not crush or deform as concrete is poured over it.

INSTALLATION—Clay Pipe's handy lengths and easy-to-make joints permit fast installation and lower labor costs. A wide variety of fittings is available to solve construction problems with no on-the-job delay, and Clay Pipe cannot float as the concrete is poured.

PERIMETER LOOP SYSTEM—The data and instructions contained here apply to the Perimeter Loop system, since it is more highly recommended by heating authorities. Its pattern eliminates the "waterfall" of cold air at the outer walls and effectively combats "radiant cold" at the outside edges of the slab. It combines the advantages of forced warm-air heating with heat by radiation through the concrete slab. Heat is concentrated where people are sitting—not at the ceiling. In test installations, the new heating method has cut operating costs from 20 to 30 percent by eliminating high heat-loss at ceilings or attics. Successful research conducted by the National Warm Air Heating and Air Conditioning Association has been followed with great interest by the building and heating industries. According to the Association, "Warm Air Perimeter Heating is a highly successful method of heating a basementless building.... It produces the ultimate of occupancy comfort. It is economical to install; can be designed to use a minimum amount of scarce materials; and requires no unusual skill in its design and installation that the heating contractor cannot supply."



The perimeter loop design shown here is used in structures built upon a concrete slab directly on the ground. Its air distribution system includes a warm air duct that extends completely around the perimeter of the building in a continuous loop. The ducts are buried in the slab 2 to 18 inches from its outside edge, and the tops of the bells will be 1½ inches below the slab surface. Heat is delivered into the rooms by warm air discharged from registers placed in the floor beneath the windows, and by radiant heat from the warm floor surfaces over the embedded ducts. The loop is supplied by feeders extending from a plenum chamber in the slab.

ADVANTAGES—It is well known that the greatest heat loss in a well constructed and insulated building—commercial or residential—will be found at the outside walls and glass areas. This loss will usually average 65 percent of the total structural heat loss. Concrete slab floors will account for approximately 18 percent of the total heat loss. The rest of the heat will be lost through the ceiling or attic. Perimeter heating introduces the heat along the outside walls and along the glass area. The heat rises and the resulting blanketing effect reduces the radiant heat loss from all outer walls and windows. Furthermore, this blanketing effect also eliminates air drafts which are caused by the cooling influence of the walls. The heating ducts, embedded in the concrete slab floors, help to maintain the floor surface temperature at desired comfortable levels. It can be seen that perimeter heating accomplishes three important basic functions: first, it replaces the heat that is lost; second, it replaces the heat at the same point where it is lost; and third, it replaces the heat as fast as it is lost. Tests conducted by the *National Warm Air Heating and Air Conditioning Association* at its University of Illinois Research Center and in actual installations throughout the country indicate that temperatures in residential structures can be maintained well within the limits established for comfort conditioning.

VERSATILITY—Clay Pipe heating ducts can also be adapted for ventilation and air conditioning purposes. This is another advantage of the under-slab design—the required amount of ventilation air can be circulated through the same distribution ducts used for heating.

FURNACES—Although a down-flow type of furnace is generally associated with the perimeter warm air heating system when installed in basementless structures, all types of winter air conditioning units can be used. Successful installations have been made using up-flow types of equipment as well as horizontal units. When up-flow furnaces or horizontal suspended units are used, down-comer ducts are used to connect the warm air plenum of the furnace unit with the distribution plenum in the floor. Also, two or more furnaces can be installed—each unit having separate control for its own distribution system so that various sections of the structure can have individual temperature control. However, one unit sized for the entire heat loss can be also installed, using two or more bonnet outlets with automatically controlled dampers in each.

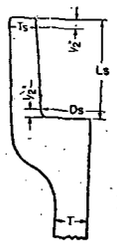
National Clay Pipe Manufacturers, Inc.

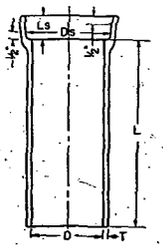
Washington 6, D. C.

VITRIFIED CLAY PIPE FOR UNDERFLOOR HEATING DUCTS

Conforming to ASTM Specifications C-13







Size, in. (D)	Laying Length (L)		Maximum Difference in Length of Two Opposite Sides, in.	Outside Diameter of Barrel, in.		Inside Diameter of Socket at ½ in. Above Base, in. (Ds)		Depth of Socket, in. (Le)		Thickness of Barrel, in. (T)		Thickness of Socket at ½ in. from Outer End, in. (Ts)	
	Nominal, ft	Limit of Minus Variation,* in. per ft of length		Min.	Max.	Min.	Max.	Nominal	Min.	Nominal	Min.	Nominal	Min.
4	2, 2½, 3	¼	5/16	4 7/8	5 1/8	5 3/4	6 1/8	1 3/4	1 1/2	1/2	7/16	7/16	3/8
6	2, 2½, 3	¼	3/8	7 1/16	7 1/16	8 3/16	8 5/8	2 1/4	2	5/8	9/16	1/2	7/16
8	2, 2½, 3	¼	7/16	9 1/4	9 3/4	10 1/4	11	2 1/2	2 1/4	3/4	11/16	9/16	1/2
10	2, 2½, 3	¼	7/16	11 1/2	12	12 3/4	13 1/4	2 5/8	2 3/8	7/8	13/16	5/8	9/16

* There is no limit for plus variation.

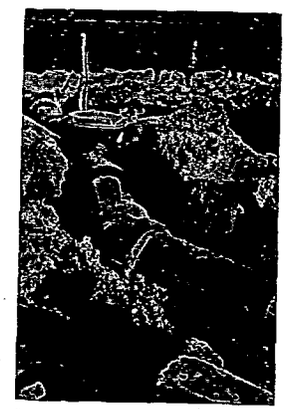
STORAGE—There are no storage problems, because vitrified clay pipe heating ducts can be exposed to weather in any season.

PERFORMANCE IN USE—Clay pipe ducts have such advantages as permanence, resistance to shock loads, and resistance to corrosion from lime in the concrete. They create no odor as warm air flows through them during furnace operation. The smooth inner surface eliminates turbulence and minimizes register noise. The clean smooth surface of the Vitrified Clay Duct is free from dust.

DESIGN DATA—Detailed design recommendations have been prepared by the *National Warm Air Heating and Air Conditioning Association* and are contained in the new revised edition of Manual 4 ($1.50) and Manual 9 ($0.75), which are available from the association's offices, 145 Public Square, Cleveland 14, Ohio. Manual No. 4 describes warm air perimeter heating systems as applied to structures without basements. Manual No. 9 contains detailed information on design and installation of warm air winter air conditioning systems and all necessary engineering data, tables, and recommendations for selection of proper heating equipment.

CASE HISTORIES—For data on actual case histories of Clay Pipe heating, ventilating and air conditioning duct installations in homes, schools, churches, hospitals, factories, gasoline stations, garages, stores, and theatres, write the National Clay Pipe Manufacturers, Inc., 1820 "N" St. N.W., Washington 6, D. C.

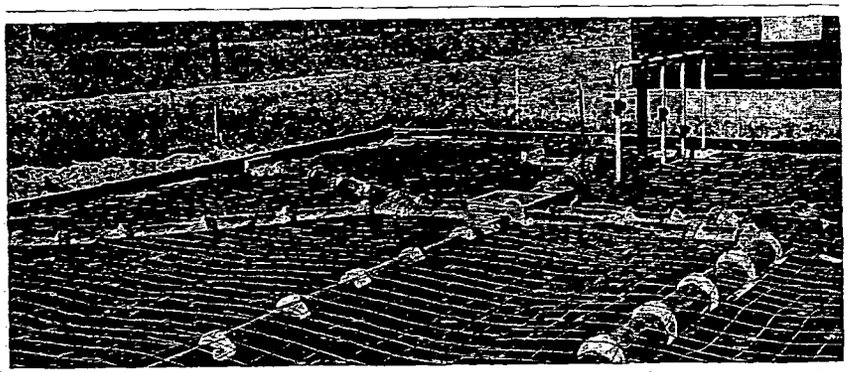
DRAINAGE—When planning basementless homes, great care should be taken to select only well-drained sites. Perforated Clay Pipe should be used to drain the building site. The drainage line should be laid around the outside of the footings and connected to an adequate lot-drainage system, in the herring bone, gridiron, or parallel patterns. Pipe with perforations half-way around the barrel provides good drainage yet carries off a minimum of dirt in suspension. For home drainage, Standard Strength (ASTM C-211) Perforated Clay Pipe is recommended.



Large diameter clay pipe heating duct and plenum connection used in commercial structure. This type of duct, when installed in garages and maintenance or repair shops, can be used for heating, ventilating, air conditioning, and removal of exhaust gases. When registers are located under the windows, the warm air will blanket the glass area and eliminate frosted and wet windows.

National Clay Pipe Manufacturers, Inc.

Washington 6, D. C.



Clay Pipe Heating Ducts ready for the concrete pouring. Wire mesh may be installed at the contractor's discretion to reinforce the floor slab.

CONSTRUCTION DATA—All ground beneath the building should be cleared of organic material. A four to six inch layer of coarse fill is then laid—washed gravel, crushed rock, or stone that has been passed over a coarse screen. Cinders or fine sand should never be used for the fill.

The fill must be covered with a moisture barrier before the slab is poured. This membrane should be the equivalent of a good grade of roll roofing at least 55 lb per 108 sq ft. Strips should be overlapped at least four inches and tarred. The barrier must completely cover the area within the foundation walls and extend up the walls to the top of the slab. See Figure 1.

Edge insulation must be placed completely around the concrete slab. Two-inch thick insulation is recommended, but one-inch is acceptable.

It is not necessary to anchor the Clay Pipe ducts to stakes, or to surround them completely with concrete (as with lighter materials). The Clay Pipe is laid directly upon the membrane, and gravel or sand is used to level it. Small depressions should be made in the fill or gravel so the pipe bells "hang free". This facilitates proper pipe alignment and prevents buckling at the joints. See Figure 2.

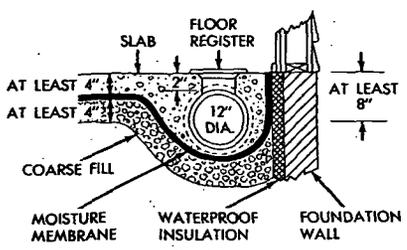


FIGURE 1

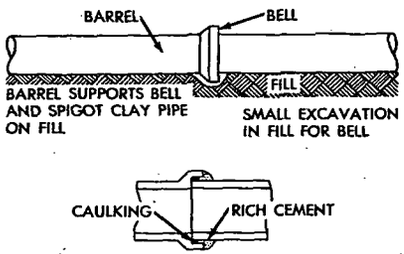


FIGURE 2

Registers must be located before the slab is poured. Wooden forms like those shown in Figure 3 are secured to the pipe by a wire wrapped around the duct. After the slab has hardened, the form can be removed and an opening cut into the pipe with a heavy hammer or bar. Some contractors prefer to retain the wooden register forms in the slab for register mounting frames.

The Clay Pipe bell at the plenum should be butted firmly against the wooden forms of the plenum pit and the concrete poured around it. See Figure 4. The pit floor is poured after the wooden form is removed. The moisture membrane must extend under the pit floor. Recommended distance from the slab floor to the top of the pipe bell is five inches.

Clay Pipe in the Perimeter Loop should be installed so that the top of the bell closest to the perimeter will be 1½ inches below the slab surface. Feeder ducts should slope downward toward the plenum, so that the top of the bell next to the plenum will be five inches below the floor slab surface. See Figure 5. All Clay Pipe joints must be waterproof.

Care must be taken when the slab is poured not to puncture the moisture barrier or dislodge the ducts. Wheelbarrows should not be moved over the ducts without suitable planking.

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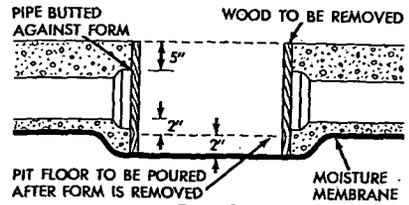


Figure 3

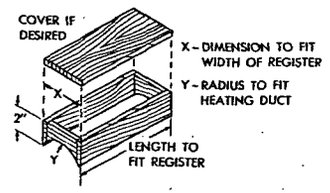


Figure 4

A strip of mesh should be placed over the top of the ducts to give added strength to the slab. Mesh may be extended over the entire slab at the contractor's discretion.

Perimeter insulation—Perimeter insulation material should have not more than 0.40 Btu conductance at 70° F per square foot per hour per degree temperature difference. The material must be moisture proof, vermin and insect proof, and resistant to deterioration from contact with organic materials in the ground. If the perimeter insulation is installed on the exterior of the foundation wall, it should be protected against damage. If installed vertically, it should extend downward not less than 18 inches and in extremely cold climates down to the frost line. Care must be taken not to rip or puncture the insulation material during installation. All openings and punctures should be closed with a suitable compound.

Fill placement—The coarse granular fill used under the concrete floor slab must consist of 4 to 6 inches of coarse material of uniform size, preferably ranging from ½ to 1 inch. It is recommended to use gravel, crushed stone, or slag that has been passed over a coarse screen. The material should be free of "fines" to insure maximum volume of air space in the fill. Sand, unscreened gravel, rock, or cinders should not be used. Cinders will pulverize in due time and their corrosive action may damage plumbing and utility lines. The fill should be distributed and leveled over the entire floor surface. After the fill has been leveled to a desired grade, it should be thoroughly compacted.

Trenches—Trenches should be of sufficient depth to accommodate the heating duct with 4 inches of coarse fill underneath it and a strip of moisture barrier laid on the top of the fill the entire length of the trench. The moisture barrier should be lapped at least 4 inches over the adjacent sections. The duct should be laid directly upon the barrier and gravel should be used to level and support the duct. The trench should be partially filled with gravel after the ducts are installed, aligned, and jointed.

Slab pouring and curing—Concrete must be made with durable, well-graded aggregate and should contain not more than 6 gallons of water per sack of cement, including the moisture contained in the aggregate. When using average moist sand, no more than 5 gallons of water per sack of cement should be added. The mix should consist of 1 part portland cement to 2¼ parts of fine aggregate well graded in sizes from ¼ inch down, and 3 parts of coarse aggregate well graded in sizes from ¼ inch up to 1 inch. Concrete should be placed and compacted thoroughly by vibrating or by tamping and spading. After the concrete is properly tamped, it should be screened to grade.

Concrete should be worked with a wood float, taking care to compact the surface properly. Depressions and inequalities of any kind should be avoided. After the concrete has hardened sufficiently to prevent fine material from working to the top (this will be indicated by the disappearance of "sheen" or shiny film of water on the surface) it should be steel-troweled, but excessive troweling should be avoided.

Concrete should be kept moist for at least 2 days. If the finished floor is to be exposed concrete, at least 5 days of moist curing is required. The floor slab should be covered with moist burlap, canvas or waterproof concrete curing paper during the curing period.

A full 28-day aging period should elapse before heat is introduced into the slab. The heating duct air temperature should be raised gradually during the first 96 hours of operation. Too rapid drying of a "green" slab may result in cracked concrete.

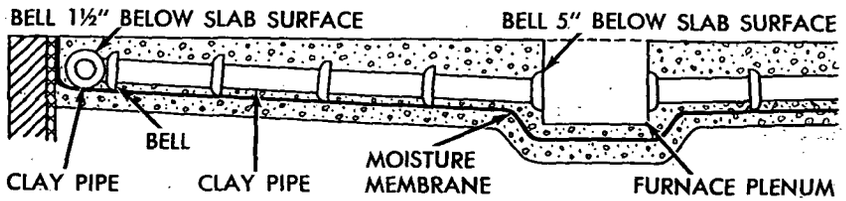


Figure 5

AMERICAN-Standard

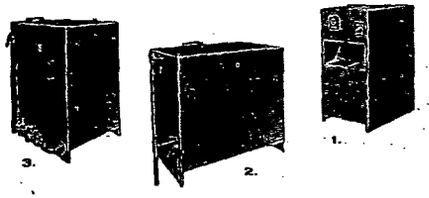
PLUMBING AND HEATING DIVISION

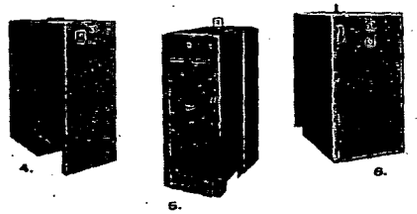
American Radiator & Standard Sanitary Corporation, New York 18, N. Y.

BOILERS—ratings, applications, fuels

Boiler	Net Ratings		Application	Fuel
	steam—sq. ft.	water—Mbh		
1. G-2*	150-750	36-180	Smaller homes, with or without basements	Gas—all types
2. G-4*	450-1,500	108.0-360.1	Larger homes, small commercial	Gas—all types
3. G-6*	1,625-13,458	390.1-3229.8	Medium to very large buildings	Gas—all types
4. Arcoleader*	——	75.0-129.8	Smaller homes, with or without basements	Oil
5. Arcoliner*	303-588	72.8-141.0	Smaller homes, with or without basements	Oil
6. Oakmont*	447-900	107.3-216.1	Larger homes, small commercial	Oil
7. A-5*	816-2,594	195.8-622.7	Larger homes and buildings	Oil
8. #2 Redflash	769-1,794	184.7-430.5	Medium to very large buildings	Hand fired
9. Severn*	456-869	109.5-208.6	Larger homes, small commercial	All fuels
10. #3 Redflash	1,503-2,400	360.6-576.0	Medium to very large buildings	All fuels
11. A-7	2,707-6,949	649.7-1667.7	Medium to very large buildings	All fuels
12. Exbrook	863-1,938	207.1-465.1	Larger homes and buildings	Stoker

*Cast iron sections of hot water boiler guaranteed 25 years in residential applications

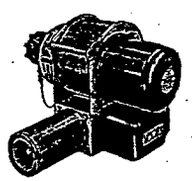


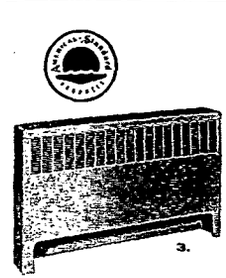


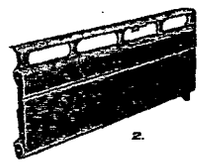
American-Standard boilers are designed for maximum efficiency in a minimum of space. They are noted for their dependability and durability, which is achieved partly through their use of corrosion-resistant cast iron sections. Many are available with tankless water heaters.

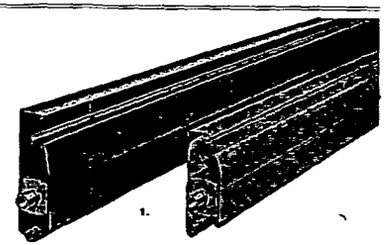
ARCOFLAME OIL BURNER—*Model PH2*—Newly designed inner assembly results in efficient, quiet operation and clean flame. Four disk-turbulator combinations provide an efficient firing range from 0.65 to 3.00 gal/hr. Hanger, pedestal and flange mountings. Tube lengths from 5" to 23". *Other models* designed for efficient operation from 0.70 to 7.00 gal/hr.

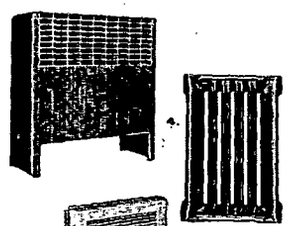


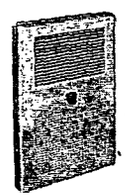


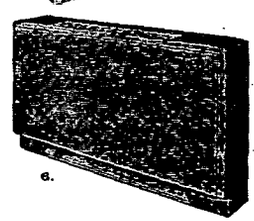


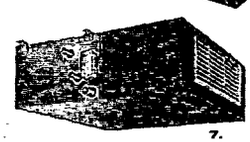


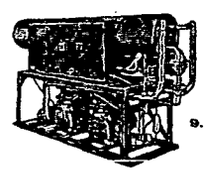












1. HEATRIM BASEBOARD PANELS—Nonferrous panels provide convected and radiant heat with forced circulation hot water systems only. Adaptable to mainless or series loop installations. Available in 4, 5, 6 and 8-ft lengths. Two models: 8"-high Model L and 10"-high Model H both available with ¾" or 1" element.

2. RADIANTRIM BASEBOARD PANELS—Replace ordinary baseboards. Cast iron panels provide all the advantages of convected and radiant heat. Model 10: 10⅜" high for forced hot water or two-pipe steam. Available in 12, 18 and 24" lengths. Sheet-metal accessories available for complete installation. Model 8: 8" high. Available in 12 and 24" lengths.

3. CONVECTORS—With cast iron (Arco) or non-ferrous (Multifin) heating elements. Styles and sizes for every need. Special designs for hospitals, institutions, etc. Multifin Type K (shown) available in 68 packaged stock sizes.

4. RADIATORS—SUNRAD: Recessed or free-standing. Needs no enclosure. Two sizes: 5" deep x 20" high, and 7½" x 23". Inlet grilles if desired. ARCO: Highly efficient, slim-tube radiators. Available in four widths—3, 4, 5 and 6 tubes—and four heights—19", 22", 25" and 32". PEERLESS: Cast iron radiators for economical heating where floor space is at a premium. May be installed on walls, ceilings or hanging from girders. Available in 2 sizes, 8 and 11 sq. ft. per section.

5. TYPE 10 REMOTAIRE ROOM CONDITIONER—Year 'round, central plant air conditioning for homes and small commercial buildings. Individually controlled room units recess into walls between studs. Units cool and heat by passing air over finned coil supplied with water from central chiller and boiler. Quiet fan motor in each unit. 150-cfm unit (shown) 24" high, 6" deep and 16" wide; 300-cfm unit 32" wide.

6. TYPE 20 REMOTAIRE ROOM CONDITIONER—Remote-type vertical unit provides year 'round hydronic air conditioning for multiroom installations such as office buildings, hotels, apartments and hospitals. Features individual room control. 200, 300, 400 and 600 cfm sizes—all 9" thin—for free-standing, partially and fully recessed installation. Test-rated capacities.

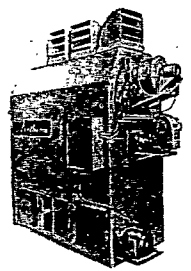
7. TYPE 30 REMOTAIRE ROOM CONDITIONER—Horizontal-style hydronic unit. Installed overhead, exposed or concealed. Four sizes: —200, 300, 400 and 600 cfm—three models to meet all design requirements.

8. TYPE 40 REMOTAIRE ROOM CONDITIONER—Through-the-wall-type year 'round heating-cooling unit has a compact self-contained refrigeration circuit. Connects to two-pipe steam or hot water system or, equipped with integral electric heating coil. May be installed room or floor at a time without disturbing normal operations of a building. One unit can heat a room while another cools adjoining room. Two capacity sizes–8200 and 11000 Btuh.

9. PACKAGED WATER CHILLERS. Models BCS and ADS—Central chilled water source for medium- and large-size air conditioning systems. Model BCS (shown) in 4 sizes, 7.5 to 20 hp; Model ADS (direct drive) in 8 sizes, 25 to 125 hp. Completely packaged unit includes compressor and motor, evaporator, condensor, heat exchangers, controls, piping, framework. Model BCS tested and shipped with operating charge of refrigerant; Model ADS with holding charge.

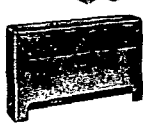
Airtherm Manufacturing Company

708 S. Spring Ave. • St. Louis 16, Missouri



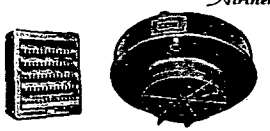
Airtherm HEAVY DUTY FORCED AIR HEATERS

HEATING SURFACES—Large, elliptically shaped stainless steel combustion chamber. Four complete passes of flame across air stream insure maximum heat extraction. One complete pass occurs in combustion chamber, with 3 separate passes through secondary heat exchanger where specially designed Turbolators swirl the hot gases. **LOW INSTALLATION COST**—Packaged, self-contained—requires connections only to fuel, power and exhaust. **APPROVED BY AMERICAN GAS ASSOCIATION AND LISTED BY *UNDERWRITERS LABORATORIES***—All control systems approved by Factory Mutual Insurance Companies. **10-YEAR WARRANTY**—All parts in direct contact with flame carry standard 10-year guarantee. **BURNERS**—Power type and available for oil, gas or combination gas-oil. Combination burners can effect immediate change-over from one fuel to another. Controls available for on-off control, high-low fire, or full burner modulation. Complete electronic protection provides power and fuel shut-off if flame fails. Heaters can be located on floor, or horizontally or vertically suspended from ceiling. Radiation shields reduce radiation and transmission losses. Casing temperature does not exceed 10 deg above ambient temperatures. **CAPACITIES**—The eight sizes have outputs of 400,000 to 2,000,000 Btu/hr. **STANDARD FAN CAPACITIES**—Range from 4900 to 24,500 cu ft per minute.



Airtherm CONVECTORS

Rounded corners and edges enhance safety and appearance. Cabinets are ruggedly constructed. Front panels are reinforced with formed edges and channel stiffeners welded to the back side. The heating elements are for use on all hot water and two-pipe, steam systems. Constructed with copper tubes and aluminum fins, with the tubes hydraulically expanded into the fin collars. Available in floor, wall or recessed cabinets and standard or heavy duty institutional construction.



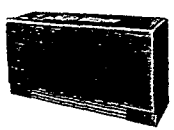
Airtherm PROPELLER FAN UNIT HEATERS

Airtherm Horizontal and Vertical Unit Heaters are available in a total of 46 Standard and Low Outlet Temperature Models. Basic capacities range from 18,600 to 555,000 Btu/hr, offering a selection to meet every job requirement. The coils are constructed of copper tubes, aluminum fins and steel headers. The tubes are expanded into the fin collars with 2,000 lbs hydraulic pressure. The coils are tested with 500 lbs hydrostatic pressure. Casings are made of heavy gage steel, phosphatized and finished with attractive gray-green enamel.



Airtherm CABINET UNIT HEATERS

Feature efficient direct-drive power assembly with aluminum centrifugal fans and permanent split capacitor motor. The cabinets are made of 16 gage steel, reinforced with angle and channel stiffeners, phosphatized and finished with attractive gray-green enamel. Coils are constructed of copper tubes, aluminum fins and steel headers. Available in 8 models with basic capacities ranging from 375 to 1,630 cfm and 25,100 to 124,800 Btu/hr. Units are offered for Floor, Wall or Ceiling mounting and in exposed or recessed cabinets.



CLASSROOM HEATING AND VENTILATING UNITS

Feature functional design for heating and ventilating classrooms and similar spaces. Damper is arranged to provide from 0 percent to 100 percent fresh air. Damper can be operated manually or automatically by application of temperature controls. The cabinets are constructed of 16 gage steel, well reinforced, phosphatized and attractively finished. Adequate end pockets for piping and controls. Coils are made of copper tubes, aluminum fins and steel headers. Power assembly is efficient direct drive with aluminum centrifugal fans and permanent split-capacitor motor. Offered in 4 capacities, 375 to 960 cfm.



Airtherm AIR CONDITIONING UNITS

Airtherm Air Conditioners provide healthful all season conditioned air comfort using a central source of chilled water for cooling and hot water for heating. Designed for dehumidifying, ventilating, filtering and introducing outside air in new or existing buildings, such as offices, hotels, apartments, motels, schools, churches and hospitals. The units are available in 4 types, for exposed floor and ceiling mounting and for concealed floor and ceiling installations. Offered in 4 models ranging from 220 to 600 cfm and ½ to 2 tons nominal cooling capacity.



Airtherm CENTRIFUGAL FAN UNIT HEATERS

Designed to deliver a large volume of air at proper velocity and temperature through outlet nozzles or through a central duct system for heating and ventilating all types of institutional, industrial and commercial buildings. Available with mixing box and dampers, face and by-pass dampers, filter section and insulated discharge plenum. Three types of coils are available, hot water, standard steam and steam distributing tube. The units are offered in 8 models with basic capacities ranging from 1,800 to 22,000 cfm and 95,000 to 1,440,000 Btu/hr.

BOILER ENGINEERING & SUPPLY CO., INC.

Phoenixville, Pa.

A HALF CENTURY OF BOILER MANUFACTURING EXPERIENCE

Continental AUTOMATIC BOILERS

20 TO 600 HP

STEAM . . . 15 TO 250 PSIG

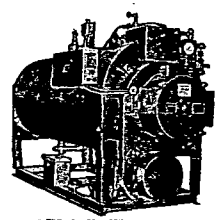
HOT WATER . . . LOW OR HIGH TEMPERATURE

ON-OFF, LO-HI-LO, AND MODULATING CONTROLS

OIL, GAS OR COMBINATION FIRED

BOILER-BURNER UNIT: COMPLETE — AUTOMATIC — ECONOMICAL

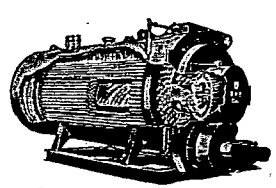
CONTINENTAL Automatic Boilers, fully-equipped for firing oil, gas, or for interchangeable oil or gas firing, generate steam at a guaranteed efficiency of not less than 80% of their conservative ratings. Units are shipped complete with wiring, glass fiber insulation, metal jacket and paint finish, ready for service connections. No special stack or foundation is needed. Necessary trim and fittings are supplied, and unit is mounted on structural steel base.



40 HP CONTINENTAL Boiler, arranged for firing light oil.

BOILER: SIMPLIFIED — RUGGED — ACCESSIBLE

Lower fuel and maintenance costs and longer service life are assured by simplified two-pass design and CONTINENTAL's unique "Spinning Gas" technique, which imparts a spiral motion at high velocity to the gases in the tubes and results in a high rate of heat transfer. Boiler Series F has five square feet of heating surface per horsepower. Built-in steam separator assures 99% dry steam. Hinged front and davited rear doors give easy access to heating surfaces. No refractory or brick-work needed, despite high furnace temperature and low stack heat. Boilers designed and built in accordance with ASME code requirements.



Rugged CONTINENTAL Boiler Design

BURNER: CLEAN — RELIABLE — EFFICIENT

Simple, rugged burner, of pressure-atomizing type, produces uniform, highly radiant flame for oil-fired burner. For gas burner, fuel is mixed with air at point of ignition; baffle-ported nozzle ends clogging or flashback. Blower drive is standard 1800 RPM motor, for forced draft at correct pressure and volume for most efficient combustion.

CONTROLS: MODERN — AUTOMATIC — DEPENDABLE

Control system, designed to meet Underwriters' Laboratories and Factory Mutual requirements as well as all local and national codes, provides such safeguards as electronic monitoring of flame, immediate shutoff and purge of burner after flame failure, low water cutoff and feed water control, front-panel signal lights.

DIMENSIONS and RATINGS

Dimensions in Inches				Nominal Rating and Capacity				Nominal Firing Rate		
Boiler HP	Length Over-all	Width Base	Height Floor to Flue Outlet	Shipping Weight (Lbs)	Steam lb per hour 212°	EDR Steam (Sq Ft)	EDR Hot Water (Sq Ft)	Light Oil (GPH)	Heavy Oil (GPH)	Natural Gas (1,000 Btu per Cu Ft)
20	100½	33½	55¼	3,770	690	2,790	4,465	6		840
30	112½	40	61¼	4,070	1,035	4,185	6,695	9		1,260
40	110¾	46	70	4,690	1,380	5,580	8,930	12		1,680
50	128¾	46	70	5,220	1,725	6,975	11,165	15		2,095
60	134⅛	53	75	6,220	2,075	8,370	13,395	18	15.7	2,520
70	139½	53	75	7,050	2,415	9,765	15,625	21	18.6	2,940
80	153⅜	53	75	7,580	2,760	11,160	17,860	24	22.7	3,360
100	147	56	83½	10,510	3,450	13,950	22,325	30	27.2	4,190
125	172	68	83¾	13,330	4,313	17,438	27,906	37.5	34.3	5,235
150	180	71	92½	15,510	5,175	20,933	33,500	45	41.3	6,280
200	217	71	92½	19,920	6,900	27,915	44,665	60	54.6	8,375
250	223	83	107	27,290	8,625	34,895	55,830	75	68.5	10,470
300	253	83	107	30,690	10,350	41,875	67,000	90	82.5	12,565
350	289	83	107	34,090	13,075	48,855	78,128	105	97.5	14,660
400	264	93	119	38,090	13,800	55,830	89,320	120	109.8	16,750
500	307½	93	119	46,090	17,250	69,790	111,665	150	137.1	20,940
600	343	93	119	52,040	20,700	83,750	134,000	180	165.0	25,120

Ten Reasons Why It Pays to Buy a CONTINENTAL Automatic Boiler

- Guaranteed 80+% efficiency . . . Fire-tested before shipment . . . Range, 20 to 600 HP.
- Simple, practical two-pass design, for uniform flow of combustion gases through all return tubes.
- Maximum radiant heat transfer achieved by "Spinning Gas" technique for thorough mixing of air and atomized fuel.
- High-CO_2 and low-temperature stack gases — proof of high combustion efficiency.
- Free, rapid water circulation, to improve heat transfer, keep heating surfaces clean.
- Easy access to furnace and all return tubes. Refractory baffles and brickwork are not required.
- An engineered product, with every part designed for simultaneous operation and maximum efficiency with every other part.
- A single source of responsibility for the trouble-free performance of each component.
- Continental Boilers have received world wide acceptance by those who are most insistent in requiring silent operation, namely Hospitals, Schools, and Churches.
- A nationwide network of factory-trained servicemen, available through your local CONTINENTAL Boiler distributor.

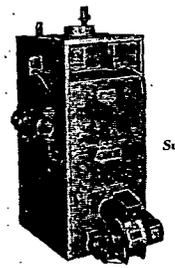
CRANE CO.

General Offices: 836 South Michigan Avenue, Chicago 5, Illinois

Nationwide Service Through Branches • Wholesalers, Plumbing and Heating Contractors

BOILERS, BASEBOARD, RADIATORS, AIR CONDITIONERS, VALVES, FITTINGS, PIPE, WATER AND STEAM SPECIALTIES, CONTROLS AND PLUMBING MATERIALS

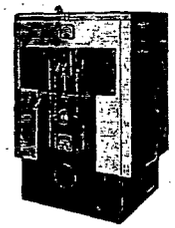
Crane—a single source for everything in heating



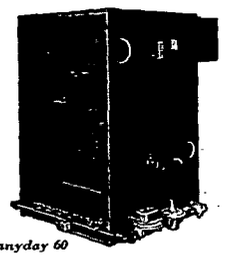
Sunnyday Seven



Sunnyday 26



Sunnyday 30



Sunnyday 60

CRANE CAST IRON BOILERS

A complete line available for the hydronics* field. (* Industry's new name for the science of heating and cooling with water.) Residential and commercial boilers. Cast iron sections of all Crane Sunnyday residential hot water boilers carry a 20-year guarantee.

RESIDENTIAL BOILERS

Crane Sunnyday Seven. All new oil fired boiler. Available in 2 sizes, prewired, pre-assembled, ready for installation. Available with tankless water heater. Net *I=B=R* ratings: 78,000 and 117,800 Btu/hr. (water)

Crane Sunnyday 15. Saves up to 15 percent on fuel. Four sizes, oil or gas fired. Two sizes available assembled and wired. Net *I=B=R* rating: 66,800 to 156,000 Btu/hr. (water, oil fired)

Crane Sunnysaver 15E. New replacement boiler for customers who already have good burners and controls. Compact, easy to install. Attractive, fully-insulated extended jacket. Four sizes, Net *I=B=R* ratings: 66,800 to 156,000 Btu/hr. (water, oil fired)

Crane Sunnyday 2WG. Only 33 inches high. Four sizes, designed especially for gas. Two sizes available fully assembled and wired. *A.G.A.* input: 70,000 to 210,000 Btu/hr. Gross *A.G.A.* output: 56,000 to 168,000 Btu/hr. Net rating: 42,000 to 126,000 Btu/hr. (water)

Crane Sunnyday 20 "All Purpose". All-fuel boiler for home or small commercial steam or hot water systems. Net *I=B=R* rating: 100,500 to 245,300 Btu/hr. (water, oil fired)

Crane Sunnyday 26. Gas fired hot water or steam boiler. Features tankless water heater on hot water models. New efficient ribbon burner. Equal ratings for all gases. *A.G.A.* input: 100,000 to 450,000 Btu/hr. Gross *A.G.A.* output: 80,000 to 360,000 Btu/hr. Net rating: 60,000 to 270,100 Btu/hr. (water)

Crane Sunnysaver 26. New low-cost factory-assembled gas fired hot water boiler. Available in 3 and 4-section sizes. Tankless water heater available in 4-section model. *A.G.A.* approved for all gases. Gross *A.G.A.* output: 80,000 and 120,000 Btu/hr. Net rating: 60,000 and 90,000 Btu/hr.

COMMERCIAL BOILERS

Crane Sunnyday 30. Medium size all-fuel boiler for small commercial steam or hot water systems (or large homes). 14-inch and 21-inch base for oil fired models. Net *I=B=R* rating: 225,100 to 843,200 Btu/hr. (water, oil fired)

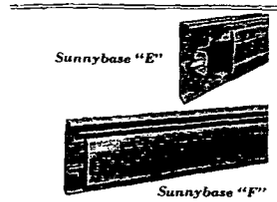
Crane Sunnyday 40. Gas fired boiler for large homes or commercial use. Patented staggered heat travel. For hot water or steam systems. *A.G.A.* input: 350,000 to 2,380,000 Btu/hr. Gross *A.G.A.* output: 280,000 to 1,904,000 Btu/hr. Net rating: 210,100 to 1,478,000 Btu/hr. (water)

Crane 41. All-fuel boiler for commercial, industrial, or institutional installations. Precision ground cast iron sections easily assembled. Net *I=B=R* rating. 621,900 to 1,823,800 Btu/hr. (water, oil fired)

Crane Sunnyday 60. Big 6-inch gas boiler. Steam, hot water, or large volume indirect water heating. *A.G.A.* input: 625,000 to 5,000,000 Btu/hr. Gross *A.G.A.* output: 500,000 to 4,000,000 Btu/hr. Net rating: 374,100 to 3,108,000 Btu/hr. (water)

VALVES AND FITTINGS

A complete line. All the necessary piping items for any hydronic heating system.

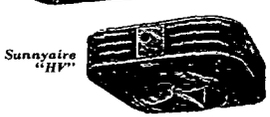


Sunnybase "E"

Sunnybase "F"



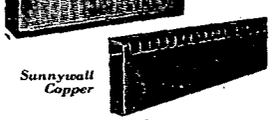
Sunnyaire Gas Unit Heater



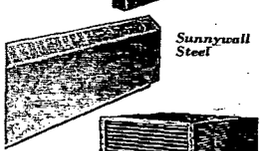
Sunnyaire "HV"



Sunnyaire "CA"



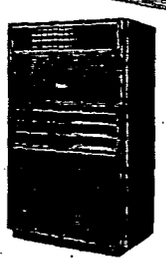
Sunnywall Copper



Sunnywall Steel



Crane Stowaway Air Conditioner



Crane Packaged Air Conditioner

CRANE SUNNYBASE PANELS

Complete line of finned-tube residential radiation. Takes the place of woodwork. Saves space, leaves entire floor free for furniture.

Sunnybase Type E. Copper tube, aluminum fins. Fins are specially corrugated for rigidity and heat transfer efficiency. Free expansion of element is provided by electro-galvanized Silent-Glide Slide Shoes. Finished installation measures 2²³⁄₃₂ x 8½ in. 3, 4, 5, and 8-ft lengths, complete accessories available.

Sunnybase Type F. Copper tube, aluminum fins. Corrugated construction gives rigidity and increased heating surface (over 4 square feet per lineal foot of baseboard). Sized-to-mate tube ends, jointed without couplings, minimize brazed joints and save fittings. Finished installation measures 2¾ x 9⁹⁄₁₆ in. Available in 4, 5, 6, and 8-ft lengths, with complete accessories.

CRANE SUNNYAIRE UNIT HEATERS

Completely automatic, safe, economical. Easy to install and maintain. Gas, hot water and steam models available in full range of sizes.

Sunnyaire Gas Unit Heater. 10 basic sizes. *A.G.A.* approved. Designed for natural, manufactured, mixed, or LP gases. 25,000 to 200,000 Btu/hr. (input).

Sunnyaire Hot Water and Steam Unit Heaters. 58 models. Type HV—for both horizontal and vertical applications. Features wrap-around heating element. Capacities from 34,000 to 684,000 Btu/hr. Type H—for horizontal applications. Features continuous tube heating element. Capacities from 15,000 to 335,000 Btu/hr.

CRANE SUNNYAIRE CABINET HEATERS

Sunnyaire Hot Water and Steam Cabinet Heaters. A complete line of blow-through and draw-through types for all practical installations. 21 hot water models, 14 steam models with air capacities from 265 to 1005 cfm. Heating capacities: 71 to 313 Sq. ft. EDR for steam, 9.1 to 65 MBH for hot water.

CRANE SUNNYWALL COMMERCIAL RADIATION—I=B=R RATED

Two types of high-capacity, superior-quality commercial radiation: Copper tube with aluminum fins, steel tube and steel fins. Full line of accessories and optional equipment.

Sunnywall copper tube with aluminum fins. Ten types of heating elements. 700 to 3060 Btu per ft. (1 lb steam) 5 enclosure styles.

Sunnywall steel tube with steel fins. Five types of heating elements. 890 to 3280 Btu per ft. (1 lb steam) 3 enclosure styles.

CRANE AIR CONDITIONING

Crane Stowaway Air Conditioner. 2 and 3½-hp units. Compressor, condenser, blowers and evaporator all in one cabinet. Available with prefabricated Fiberglas ducts.

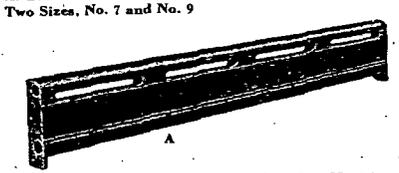
Crane Packaged Air Conditioners. A new line for offices, restaurants, banks, stores and small factories. Includes 3, 5, 7½, 10 and 15 ton units with front grille or top duct discharge. Water condensing units are factory assembled, wired, charged and tested. Air condensing units have receiver ready for addition of remote air-cooled condenser. All models fully warranted.

Burnham Corporation

HEATING & COOLING DIVISION
IRVINGTON, NEW YORK

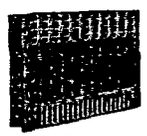
There's a Burnham for Every Purpose—Write for Descriptive Catalogs

A. Burnham Cast-Iron BASE-RAY® Radiant Baseboard
Two Sizes, No. 7 and No. 9



A

I=B=R Rating per sq ft for No. 7 is 2.35 and for No. 9 is 3.45 per sq ft Tappings ¾ in. top and bottom. Sections come in 12, 18 and 24 in. lengths. New matching sheet metal baseboard extensions and accessories available.



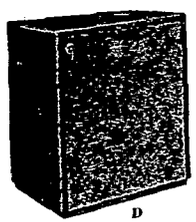
B



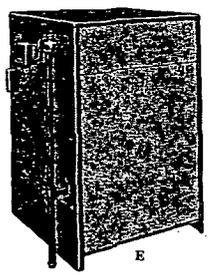
C

B. BURNHAM RADIANT RADIATORS—Two heights 20 and 23 in. Ratings 2.25 sq ft per section and 3.40 sq ft per section respectively.

C. BURNHAM SLENDERIZED RADIATORS—Three to six tubes in four heights 19 in. to 32 in. Ratings are from 1.6 sq ft per section to 3.7 sq ft per section.



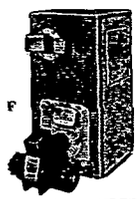
D



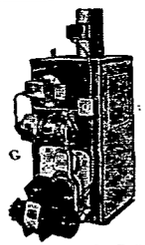
E

D. HOLIDAY Cast-Iron Boiler #60 is *A.G.A.* approved for manufactured, natural, mixed or LP gases and *I=B=R* approved and rated. Capacities for water from 300 to 1100 sq ft and for steam 190 to 690 sq ft. Can be equipped with built-in year 'round tankless hot water coil. Five sizes as HOLIDAY-PAK Factory assembled.

E. New HOLIDAY Cast-Iron Boiler #70 is *A.G.A.* approved for manufactured, natural or mixed gases and is *I=B=R* approved and rated. Capacities are for water 600 to 2700 sq ft and for steam 375 to 1690 sq ft. Can be equipped with tankless built-in hot water coil.



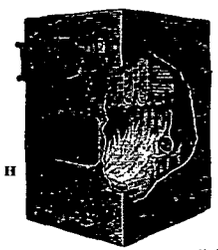
F



G

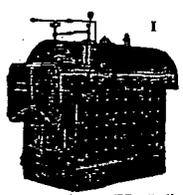
F. BURNHAM JUBILEE® Wet Base Cast-iron Oil Boiler—Vertical Flue Travel. Available in 7 sizes for steam, 320 to 955 sq ft and 8 sizes for water, 445 to 1525 sq ft. An overall range of 9 sizes. Flush and extended Jacket models. Provided with built-in hot water coils if desired.

G. BURNHAM JUBILEE PAK—Available in 5 sizes, 445 sq ft to 1010 sq ft water only. Shown with controls and wiring factory installed. Delivered crated ready for easy installation.

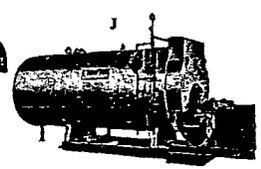


H

H. PACE-KING® BOILER—Exclusively oil fired for larger buildings. Vertical flue design . . . low stack temperatures. Tankless hot water heaters with capacities from 6 to 15 gal. 14 models from 1205 to 4500 sq ft for steam and 1925 to 7200 sq ft for water. Built according to *ASME* code and *I=B=R* rating and testing code.



I



J

I. WELDED STEEL Boiler—Compact type—Capacities from 2600 to 35000 sq ft for steam and 4680 to 56000 sq ft for water, EDR. Net *SBI* ratings. Furnished for coal, oil or stoker firing.

J. SCOTCH TYPE Steel Boiler—completely packaged with combustion unit or unpackaged. Net *SBI* capacities from 2600 to 35000 sq ft EDR steam and from 4680 to 56000 sq ft EDR water. Also we build these up to 4000 sq ft of heating surface.

National-U. S. Radiator
CORPORATION

HEATING AND AIR CONDITIONING DIVISION
Johnstown, Pennsylvania

Cast Iron and Steel Boilers • Baseboard • Convectors • Radiators
Fintube Unit Heaters • Furnaces • Oil and Gas Burners •
Air Conditioning • Accessories

Nationwide distribution through Branches, Wholesalers, Plumbing and Heating Contractors

NATIONAL-U. S. CAST IRON BOILERS AND UNITS (FOR ALL FUELS)



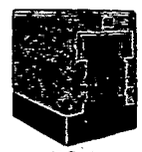
Sunray IV



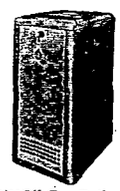
Comfortline



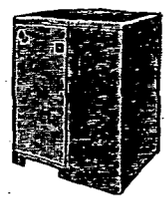
Premier



47 Series



"11B" Gas Boiler



"16C" Gas Boiler



"22" Gas Boiler



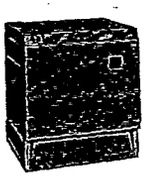
"66" Gas Boiler

CAST IRON BOILERS AND UNITS—RATINGS, APPLICATIONS, FUELS

BOILER OR UNIT	NET *I=B=R* RATING BTU/HR		NO. OF SIZES	APPLICATION	FUEL
	STEAM	WATER			
Sunray IV Packet	66,800	158,300	4	Small to medium homes	Oil
Sunray IV Oil Heating Boiler & Unit	66,800	219,100	6	Small to large size residential installations	Oil
Comfortline Boiler and Oil Heating Unit	75,000	85,500	2	Small to medium residential	Oil
Premier Oil Heating Boiler ("25B" Series)	203,300	565,600	6	Large residential to small commercial installations	Oil
"27" Series All-Fuels Boiler	109,500	234,100	6	Medium to large size installations	All Fuels
"27" Series Oil Heating Boiler and Unit	109,500	234,100	6	Medium to large size residences	Oil
"37" Series All-Fuels Boiler	232,600	636,200	9	Large residential and small commercial installations	All Fuels
"37" Series Oil Heating Unit	232,600	636,200	9	Large residential and small commercial installations	Oil
"47" Series All Fuels Boiler	622,700	1,823,000	10	Institutional, commercial and industrial	All Fuels
"47" Series Oil Heating Unit	622,700	1,537,300	8	Institutional, commercial and industrial	Oil
"11B" Series Gas Boiler*	27,000	108,000	7	Small to medium homes	All Gases
"16C" Series Gas Boiler*	87,000	565,600	11	Residential to commercial	All Gases
"22" Series Gas Boiler*	38,000	107,300	6	Small to medium homes	All Gases
"33" Series Gas Boiler*	97,500	259,600	6	Medium to large size homes	All Gases
"44" Series Gas Boiler*	121,500	486,100	9	Large size installations	All Gases
"66" Series Gas Boiler*	450,100	3,416,100	20	Institutional, commercial and industrial installations	All Gases

* *A.G.A. Approved.*

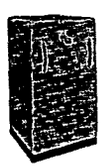
NATIONAL-U. S. STEEL BOILERS AND UNITS—(ALL FUELS)



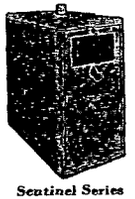
Mark II Packet



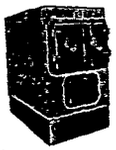
Mark III Packet



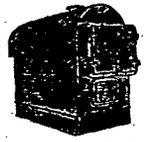
Model U Packet



Sentinel Series



Commander Series



Commercial Steel Boiler



Scotch Type "Wet Back" Boiler (Type SN & SF)



"King of Scots" Boiler-Burner Unit

STEEL BOILERS AND UNITS—RATINGS, APPLICATIONS, FUELS

BOILER OR UNIT	SBI NET RATING		NO. OF SIZES	APPLICATION	FUEL
	STEAM	WATER			
Mark II Packet (OP-15-WE)		74,000	1	Small to medium homes	Oil
Mark II Packet (GP-15-WE)		69,000	1	Small to medium homes	Gas
Mark III Packet	87,000	189,000	4	Small to medium size residential installations	Oil
Model U Packet (OU-19-W)		87,000	1	Small to medium size residential installations	Oil
Model U Packet (GU-19-W)		78,000	1	Small to medium size residential installations	Gas
Sentinel Steel Boilers and Units (17", & 24", Series)	87,000	189,000	4	Small to medium size residential installations	Oil
Commander Steel Boilers (26", 29", 39, & 41", Series)	243,000	1,350,000	18	Large residential to small commercial installations	Oil, Gas or, Stoker
Commander Steel Boilers and Units (26", 29", 39", & 41", Series)	243,000	1,350,000	13	Large residential to small commercial installations	Oil or Gas
Commercial Steel Boilers	813,000	9,450,000	16	Institutional, commercial and industrial	Gas, Oil Stoker or hand
Scotch Type Boilers (SN-Natural Draft)	1,217,000	9,950,000	13	Institutional, commercial and industrial	Oil or Gas
*Scotch Type Boilers (SF-Forced Draft)	1,969,000	15,300,000	13	Institutional, commercial and industrial	Oil, Gas, Gas-Oil
*Scotch Type, Packaged Boiler-Burner Units	1,969,000	15,300,000	13	Institutional, commercial and industrial	Oil, Gas, Gas-Oil

* *Manufacturers Certified Ratings.*

National-U. S. Heat Distributors

Baseboard—three types available: Sunray (cast iron), Art Comfortline (standard capacity, non-ferrous) and Floorline (high capacity, non-ferrous).

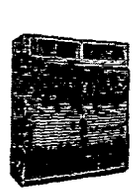
Convectors—two types: Aero Convectors (cast iron) and Art Convectors (non-ferrous). For hot water and steam systems. Numerous types of enclosures.

Radiators—three types available: Thin-tube are compact and slender; Sunray with solid front and Wall for versatility.

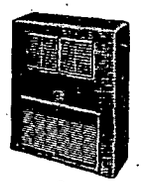
Unit Heaters—for hot water and steam also for gas firing—both types in a wide range of sizes

Fintube Radiation—available in two types: steel tubing with steel fins and copper tubing with aluminum fins. Variety of covers can be furnished.

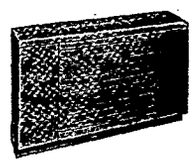
Conversion Burners and Heating Accessories—oil and gas conversion burners for boilers and furnaces. Complete line of heating accessories for convenient one-source service.



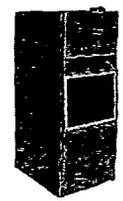
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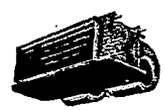
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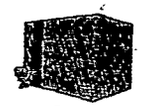
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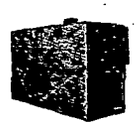
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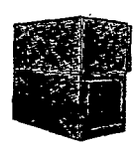
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7



8



9



10



11

CAPITOLAIRE AIR CONDITIONING AND WARM AIR FURNACES

1. Residential and Commercial Packaged Air Conditioners . . . water or air cooled . . . 3 to 20 tons.

2. Residential Fan-Coil Unit, Model VRS . . . for heating and cooling . . . in 150 and 300 CFM sizes . . . available for recessed or free standing installations.

3. Vertical Fan-Coil Unit, Model VER . . . for heating and cooling . . . in sizes of 200, 300, 400 and 600 CFM . . . for concealed or free standing installations.

4. Gas or Oil Fired Steel Furnaces . . . residential and small commercial . . . ranging in size from 64,000 to 200,000 Btu/hr output.

5. Horizontal Fan-Coil Unit, Model HER . . . in sizes of 200, 400 and 600 CFM . . . for concealed installations with and without duct work.

6. Horizontal Fan-Coil Unit, Model HRZ . . . in 300 to 1750 CFM . . . for surface or concealed installations.

7. Hi-Cap Furnace . . . for gas, oil or coal firing . . for commercial and industrial installations . . . output up to 1,000,000 Btu/hr . . . suspended models also available.

8. Model CEN-IE . . . self-contained packaged air conditioner with built-in evaporative condenser . . . ranging from 7½ to 60 ton capacity.

9. Model CEN-IW . . . self-contained packaged air conditioner with water-cooled condenser . . . ranging from 7½ to 60 ton capacity.

10. Model CHL-IW . . . packaged water chiller with water-cooled condenser . . . ranging from 7½ to 75 ton capacity.

11. Model CHL-IE . . . packaged water chiller with built-in evaporative condenser . . . ranging from 7½ to 75 ton capacity.

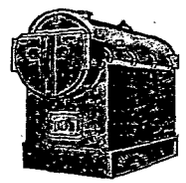
Pacific Steel Boiler DIVISION

National-U.S. Radiator Corporation

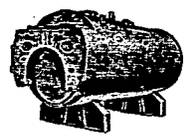
JOHNSTOWN, PENNSYLVANIA

MEMBER SBI

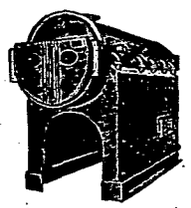
Sales Offices in All Principal Cities



Pacific Standard Firebox Boilers



Pacific Scotch Type Boilers



Pacific "LoSet" Boilers



Pacific Split Firebox Boilers

SBI TABLE 1 BOILERS

A Complete Line of Low-Pressure Steel Heating Boilers For Commercial Application

All Pacific Commercial Boilers are built in conformity with the *ASME* Code for low pressure heating boilers and are rated in accordance with *The Steel Boiler Institute* Rating Code with the exception of the Scotch Type Boilers for forced draft firing which have Pacific certified ratings.

PACIFIC STANDARD FIREBOX BOILERS

Pacific Standard Firebox Boilers for Stoker, oil or gas firing are built in capacities of *SBI* rating from 2,840 to 64,280 sq ft for steam and from 4,540 to 102,850 sq ft for water. Design is of two types: Low Water Line and High Firebox. Pacific Direct Draft and Smokeless Boilers for coal firing are built in capacities of 2,200 to 35,000 sq ft for steam and in corresponding capacities for water.

PACIFIC SCOTCH TYPE BOILERS

Pacific Scotch Type Boilers are available for natural or forced draft firing with oil or gas. The Natural Draft Scotch Type Boiler ratings range from an *SBI* rating of 5,800 to 64,280 sq ft steam and 9,280 to 102,850 sq ft water. The Forced Draft Scotch Type Boiler ratings are manufacturers' certified rating and range from 1,969,000 to 21,775,000 Btu/hr. Both series applicable for steam and water heating.

PACIFIC "LOSET" BOILERS

Pacific "LoSet" Boilers for oil firing are available in 18 sizes with capacities of *SBI* rating from 4,500 to 64,280 sq ft steam and 7,200 to 102,850 sq ft water. Unique "LoSet" design requires minimum headroom without sacrificing boiler efficiency.

PACIFIC SPLIT-FIREBOX BOILERS

Pacific Low Water Line, High Firebox and LoSet Boilers are available in three sections—shell, firebox and base—and can be moved through minimum building openings. Where necessary, Pacific fireboxes can be split (as illustrated) to allow the boiler to be taken into the building in four pieces, through most doors, windows and hallways. No cutting or welding is required in assembling Pacific Boilers of this type.

Descriptive bulletins on Pacific Commercial Boilers will be mailed on request

Pacific Steel Boiler DIVISION

National-U.S. Radiator Corporation

JOHNSTOWN, PENNSYLVANIA

MEMBER SBI

Sales Offices in All Principal Cities

SBI TABLE 2 BOILERS

A Complete Line of Residential Steel Boilers for Coal, Stoker, Oil or Gas Firing

All Pacific Residential Boilers are built in conformity with the *ASME* Code for low pressure heating boilers and are rated in accordance with *The Steel Boiler Institute* Rating Code.



Pacific "Plate Flue" Boilers

PACIFIC "PLATE FLUE" BOILERS

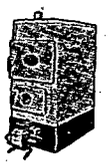
Pacific "Plate Flue" Boilers are designed for oil or gas firing and are built in 4 sizes. *SBI* net ratings range from 108,000 to 243,000 Btu/hr for steam and water heating. They are available with either flush or extended jackets.



Pacific "Plate Flue" Oil Heating Units

PACIFIC "PLATE FLUE" OIL HEATING UNITS

Pacific "Plate Flue" Oil Heating Units are boiler-burner units designed for steam or hot water heating. Furnished with pre-cast combustion chamber, matched oil burner and automatic controls. Ratings are the same as for "Plate Flue" Boilers. Available with flush or extended jackets.



Boiler-Burner Units



Automatically Fired Boilers

PACIFIC SERIES "700" BOILERS (3-pass Design)

Pacific Series "700" Boilers of the 3-pass design (firebox and 2 passes of tubes) available for automatic and hand firing, also as oil or gas boiler-burner units. Designed for large residential installations. *SBI* Net Ratings range from 180,000 to 595,000 Btu/hr in 7 sizes for hand firing and from 297,000 to 810,000 Btu/hr in 7 sizes for automatic firing for steam and water heating. Available with flush jacket.



Boiler-Burner Units



Automatically Fired Boilers

PACIFIC SERIES "700" BOILERS (2-Pass Design)

Pacific "700" Series Boilers of the 2-pass design (firebox and 1 pass of tubes) available for automatic and hand firing, also as oil or gas boiler-burner units. Designed for large residential and small commercial installations. *SBI* Net Ratings range from 945,000 to 1,350,000 Btu/hr in 4 sizes for steam and water heating and for automatic firing. Hand fired *SBI* Net Ratings range from 690,000 to 990,000 Btu/hr. Available with flush jacket.

Descriptive bulletins on Pacific Residential Boilers will be mailed on request

THE PEERLESS HEATER COMPANY

Boyertown, Pennsylvania

Boilers Cast Iron

Hot Water
Steam
Vapor

PEERLESS
OF BOYERTOWN PENNSYLVANIA
SINCE 1908

Furnaces

Automatic
Winter Air
Conditioners

Manufacturing Division and General Offices

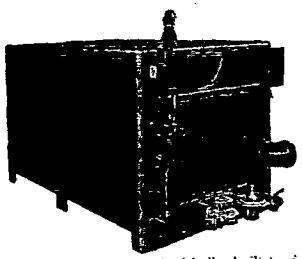
55 GAS FIRED SIZES

Rated 70,000 Btu to 5,400,000 Btu input/hr.

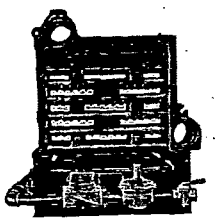
For close adherence to specifications.

Cast Iron Boiler **Gas Fired**

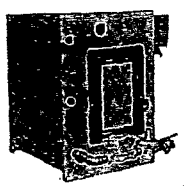
BIG SERIES 150



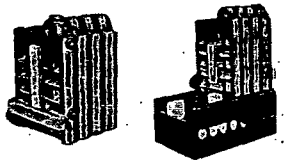
SERIES 150 High capacity gas fired boiler built to rigid specifications, designed for industrial use with Natural, Manufactured and Mixed gases, LP and LP Gas-Air Mixtures. Incorporates multiple pilot line, raised ports, cast iron burners, horizontal to vertical flue collector plus additional quality features. 33 Sizes. *A.G.A.* rated 600,000 to 5,400,000 Btu/hr input. Hot Water Steam and Vapor.



Peerless "SUPER" 13 gas passage, cast iron water tube section "H" block construction with 6 in. ports and tappings. Maximum strength and efficiency are attained.



SERIES 50 Recommended for industrial and large residential installations. Incorporates the same quality features of the 150 SERIES with less capacity and more compactness to enable installations where limited space is a requisite. The latest advancements and design are the features in the SERIES 50. 7 Sizes. *A.G.A.* rated 350,000 to 650,000 Btu/hr input. Hot Water, Steam and Vapor.

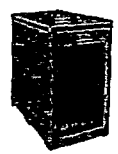


All Series 50 Cast Iron Boiler sections are preassembled in blocks of 3, 4 and 5 sections to effect a savings and assure positive installations. Section blocks are factory hydrostatically tested at pressures up to 200 lb.

The PEERLESS Series 50 and 150 introduces important advances in engineering research, modern design, new materials, manufacturing methods and boiler setting-up procedures. PEERLESS'S world famous master foundry workers and engineers have introduced sensationally new concepts in CAST IRON BOILERS, the lifetime metal. With the introduction of the Series 50 and 150, PEERLESS now offers 55 sizes of Gas Fired Boilers to meet every installation requirement from 70,000 to 5,400,000 Btu input.

A.G.A. A.S.M.E. I.B.R.

CAST IRON GAS AND OIL FIRED BOILERS



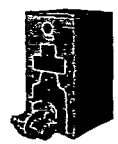
Series "TG"



Series "G"



Series "TO"



Series "TOU"

SERIES "TG" Gas fired cast iron boilers meet most residential requirements, including instantaneous domestic copper water heater. Available in deluxe flush and deluxe extended type jackets.

SERIES "G" Gas fired boilers incorporate the same features as the "TG" Series but do not have domestic hot water heater. Deluxe flush or deluxe extended jackets are available. Rugged cast iron construction for long life. Factory assembled block sections for fast set-up. Deluxe two-tone green enameled steel jackets. SERIES "TG" & "G" 6 Sizes—105,000 to 291,000 Btu Input hr. *A.G.A.* rated. Steam Water and Vapor.

SERIES "TO" Oil fired boilers with all Peerless water tube features: Factory assembled sections, domestic hot water heater optional. "O" Series available without domestic water heater.

SERIES "TOU" Oil fired boiler is the same as Series "TO" but includes Peerless Deluxe oil burner. Deluxe flush and extended jackets available. Peerless oil fired heavy duty residential heating boilers are made for homes of all sizes. 7 Sizes. Gross $I = B = R$ Output Btu/hr 93,000 to 234,000. Net Water Rating: 465 sq ft to 1170 sq ft. Net Steam Rating: 250 sq ft to 650 sq ft.

PACKAGED OIL AND GAS FIRED BOILERS



Series "TOUP" Crated



Series "TOUP"



Series "MM" Crated



Series "MM"

SERIES "TOUP" Oil fired boilers are delivered completely crated with all controls. Factory assembled and tested to assure time saving installations. 2 Sizes. "TOUP"-135A-W-Net Water Rating: 555 sq ft. "TOUP"-145A-W. Net Water Rating: 735 sq ft.

DESCRIPTIVE CATALOGS OF SPECIFICATIONS, RATINGS AND DIMENSIONAL DRAWINGS OF PEERLESS BOILERS WILL BE MAILED UPON REQUEST.

SERIES "MM" (Mighty Midget) boiler with compact design for installations in limited space areas. White baked enamel. Completely assembled and crated. "MM" 70B-W, *A.G.A.* input 70,000 Btu/hr, 24 in. high. "MM" 95B-W, *A.G.A.* Input 95,000 Btu, 30 in. High. "MM" 120B-W, *A.G.A.* Input 120,000 Btu, 36 in. High.



Oil Model



Gas Model

PEERLESS GAS AND OIL FIRED FURNACES

Peerless fully automatic Oil and Gas fired furnaces represent advancements in Engineering, Performance and Design.

A full line consisting of 9 Series with 22 models provide installations for every type of home.

Gas fired furnaces are rated 70,000–150,000 Btu/hr Input. Oil fired furnaces are rated 72,000–140,000 Btu/hr Output.

The 9 Series of Peerless gas and oil fired furnaces include 10 Models in the gas fired Series and 12 Models in the oil fired Series. Series include HI-BOY, LO-BOY, GRAVITY, COUNTERFLOW and HORIZONTAL TYPES.

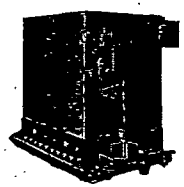
Peerless warm air furnaces are completely assembled, wired and packaged at the factory for ease of installation. Specification sheets are available by writing The Peerless Heater Company, Boyertown, Pennsylvania.

PENNSYLVANIA FURNACE & IRON COMPANY

Warren, Pennsylvania
316 North Pine Street

Pennco Cast Iron Gas Fired Boilers, Pennsylvania Forced Air and Gravity Furnaces, Floor Furnaces, Conversion Burners, Multitube Burners, and Cooling Equipment

SERIES 6 PENNCO GAS FIRED BOILERS



Thirty sizes, all gases, 400,000 to 4,400,000 input.
Steam, water, vapor and volume hot water heating.
ASME and *A.G.A.* approved. Twice tested to 200 lb hydrostatic for working pressures of 15 lb steam or 80 lb maximum water pressure.
Base is of welded steel, shipped assembled complete with manifold and safety pilot in place.

Horizontal draft diverters save head room. Vertical diverters can be furnished upon request.
Front casing panel removable for convenience in cleaning.
Draw bolts for easy assembly of section. Holding bolts keep sections metal to metal.
Individual gas burner cocks for each section permit shutting off one or more burners and still maintain operation of the boiler.
Boilers are equipped with gas actuated controls to permit operation in event of electric failure. Electric controls are also available.

SERIES 3 PENNCO GAS FIRED BOILERS



Eight sizes for all gases. Inputs from 96,000 to 432,000 Btu.
Steam, water, vapor and volume water heating.
ASME and *A.G.A.* approved. Two tests of cast sections at 200 lb hydrostatic. Water boiler approved for 80 lb working pressure.
Factory assembly of battery of sections.
Water leg sections provide increased efficiency.
Hollow projections on sections increase area of heating surface for faster transfer.

Boiler casing Standard type with controls exposed or DeLuxe type enclosing controls. Two-tone blue baked enamel finish.
Controls are gas actuated or electric types; factory assembled and fire tested. Color coding removes any guesswork during erection.

SERIES 2 PENNCO GAS FIRED BOILERS



Four sizes for all gases. Water operation only. Inputs 44,000 to 87,625 Btu.
ASME and *A.G.A.* approved. Sections tested at 200 lb hydraulic pressure.
Approved for 80 lb working pressure.
Cast sections shipped assembled.
Adapted to heating of apartments, small homes, stores, and offices.
Wet base and diamond shaped hollow projections increase efficiency.
Choice of Standard casings for basement use or DeLuxe casing for installation in kitchens and other finished rooms.

SPECIAL

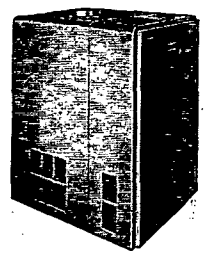
Stand-by controls to supplement natural or artificial gas with LP gas.
Automatic operation of dual gas controls. By predetermined setting based on outside temperature operation will swing from natural gas to LP gas and return as weather changes. *Ask for special bulletins.*

The H. B. SMITH COMPANY, INC.

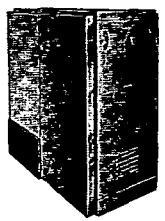
Westfield, Mass.

Branch Offices and Sales Representatives in Principal Cities

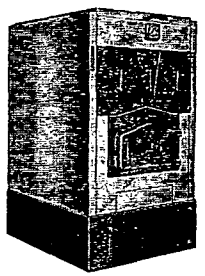
A complete line of modern cast iron sectional boilers for residential, commercial and industrial heating and for domestic hot water supply.



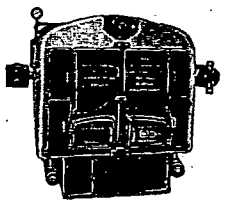
SMITH "CENTURY" GAS BOILER, *A.G.A.* approved, for steam or hot water heating. Input ratings from 138,000 to 745,500 Btu. Combines latest features of gas boiler design. Available with five sizes of built-in tank domestic hot water heaters, capacities to 168 gal. and eight sizes of tankless heaters, capacities to 8 gpm. Entire burner assembly mounted on wrought iron frame with casters for easy removal.



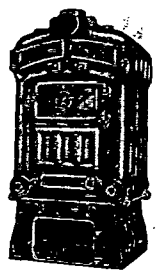
"100" & "2000" OIL BOILER-BURNER steam or water units for homes and small commercial installations. Furnished with line of tankless heaters with capacities up to 8 gal. of hot water per minute. Net radiation to 1275 sq. ft. steam; 2210 sq. ft., water.



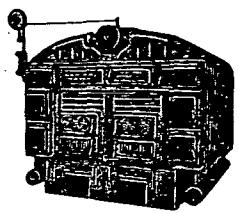
SMITH-MILLS BOILERS, Series 15-20-25; capacities 200 to 2275 sq. ft. steam radiation. This complete line of modern push nipple boilers is available in models for gas, oil, stoker or hand firing. Have provisions for built-in domestic hot water heaters and controls.



MILLS WATER TUBE BOILERS, Series 24-34-44; capacities 900 to 13,380 sq. ft. steam radiation. Independent header type construction. Tens of thousands are installed in schools, hospitals, apartment houses, stores and other commercial and public buildings. Models for hand and all types of automatic firing.



SMITH HY-TEST BOILERS for hot water supply are furnished in several models, and many sizes for tank capacities to 20,000 gal. Constructed of the finest quality grey iron castings, these Hy-Test units are carefully tested at extremely high pressures before shipment.



"42" & "60" SMITH BOILERS, may be used in batteries for heating loads up to and over 100,000 sq. ft. steam radiation. Many of these large units installed in industrial plants furnish steam for processing requirements as well as for heating and domestic hot water.

SPENCER HEATER

A PRODUCT OF AVCO MANUFACTURING CORP.

WILLIAMSPORT • PENNSYLVANIA

SBI

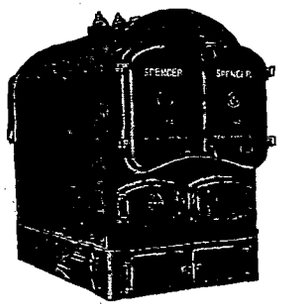
Sales Representatives

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Spokane, Wash.
St. Petersburg, Fla.
Toledo, Ohio
Washington, D. C.
Portland, Ore.
Blenheim, Ont., Canada

STEEL AND CAST-IRON HEATING BOILERS FOR EVERY BUILDING . . . FOR EVERY FUEL

"LW", "A", "CP", "R", and Suburban Series fully approved by Steel Boiler Institute

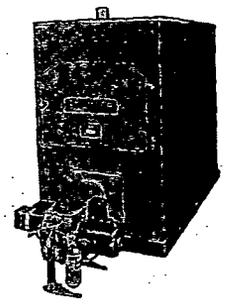
All products manufactured in strict accordance with the ASME code and carry the code seal. Every Spencer meets rated specifications, is easy to install; assured economical operation.



"A" SERIES

CAPACITIES 3,500 to 42,500 sq ft

STEAM *SBI* RATING—A rugged, tough commercial and industrial heating boiler *designed for fast steaming* and efficient operation. All sizes available with front head cuts for low-set installations and with storage or instantaneous service water coils.

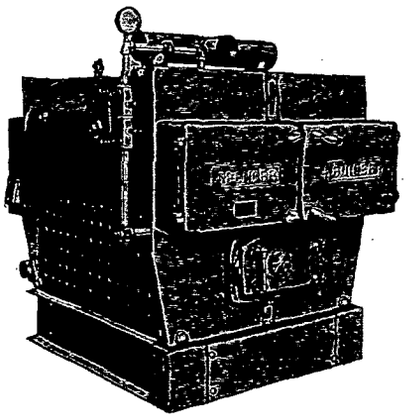


"CP" SERIES (OIL- OR GAS-FIRED)

CAPACITIES 1,300 to 5,000 sq ft

STEAM NET *SBI* RATINGS

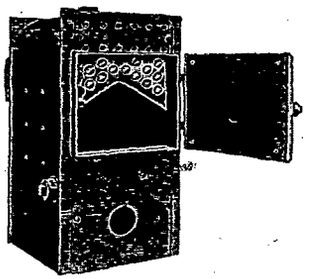
468,000 to 1,800,000 Btu GROSS OUTPUT—Incorporating a revolutionary refractory slab base and with either oil- or gas-burning equipment. Easy and quick to install. Low waterline for problem installations.



"LW" SERIES

CAPACITIES 2,200 to 42,500 sq ft

STEAM *SBI* RATINGS—A patented, divided low-waterline boiler *made in two watertight sections* that need no welding on job. Designed to overcome problems in low head room, excavation and entry. Particularly adapted for replacement installations. Available *with front head cut for extra low-set installations.*

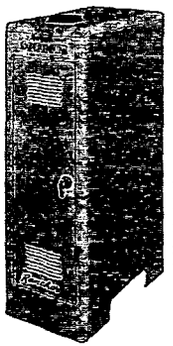


"R" SERIES

RESIDENTIAL STEEL BOILER

CAPACITIES 320 to 1,100 sq ft

STEAM, NET LOAD—*SBI* RATINGS—A ruggedly *constructed*, high-quality boiler for oil, gas, or stoker firing. Available with storage or instantaneous type service water coils in a wide range of capacities. Particularly suited for engineers and contractors who prefer to build their own design for heating installations. Available with both flush and extended model jackets.

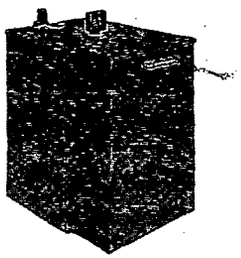


"RANCHER" CAST-IRON GAS BOILER CAPACITIES:

1G series: 40,000 to 128,000 Btu

2G series: 80,000 to 232,000 Btu

A clean, neat compact design. Especially quiet in operation, using cast-iron burners with raised drilled ports. Also available with year-round domestic hot water coils.



"SUBURBAN" GAS-AND OIL-FIRED PACKAGE UNITS

CAPACITIES: 87,000, 108,000, 149,000 and 189,000 Btu—NET *SBI* WATER RATING—A compact, quality package unit designed for installation in kitchens, utility or rumpus rooms. Baked white enamel jacket. Completely wired and ready *for installation* . . . with volumes of year-round domestic hot water supply.

SPENCER HEATER

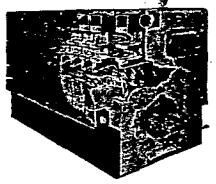
A PRODUCT OF AVCO MANUFACTURING CORP.

WILLIAMSPORT • PENNSYLVANIA

WEIL-McLAIN COMPANY

General Sales Office: MICHIGAN CITY, INDIANA

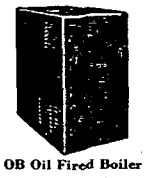
Distributors in principal cities



HR Heavy Duty Boiler for Horizontal Rotary Burners

Special steel front plate and section simplify rotary burner installation. Large fire-box volume develops better combustion conditions—high base eliminates high brick foundation. *ASME* constructed.

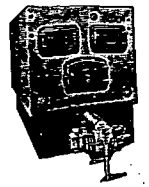
Load range: Steam and Water 885, 200–2,942,500 Btu/hr.



OB Oil Fired Boiler

Cast iron boiler designed specifically to burn oil with great economy. Can be installed on any type of floor—special base plate allows free air circulation beneath boiler. Shipped completely assembled including specially designed refractory combustion chamber. *ASME* constructed.

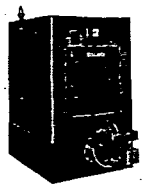
Load range: Net *I=B=R* rating, Steam 68,000–88,000 Btu/hr.; Water 79,000–101,000 Btu/hr.



B-HO-40 and 44 Heavy Duty Oil Fired Boilers

For dependable, efficient operation in school, commercial and industrial installations. The burner and other components are engineered to function as a unit. High base eliminates brick foundation or pitting. New burner design assures smooth, quiet starts. *ASME* constructed.

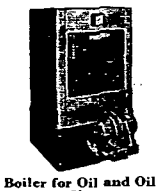
Load range: Net *I=B=R* rating, Steam and Water 667,700–1,897,500 Btu/hr.



No. 82 Oil Fired Boiler and Oil Heating Unit

Available as "B" unit, including burner and combustion chamber; "A" unit with combustion chamber only; "O" unit, without burner and combustion chamber. An electronic safety control proves the flame in boilers burning 7 GPH and up. *ASME* constructed. Provision for single or dual domestic water heaters.

Load range: Net *I=B=R* rating, Steam and Water 339,100–793,700 Btu/hr.



No. 72 Boiler for Oil and Oil Heating Unit

Exceptional fuel economy due to self-regulating, turn-around flue gas travel. Extra large fire box; wide, finned crown sheet. Available as "B" unit, including burner and combustion chamber; "A" unit with combustion chamber only; "O" unit without burner and combustion chamber. *ASME* constructed.

Load range: Net *I=B=R* rating, Steam 161,000–305,000 Btu/hr.; Water 180,000–331,000 Btu/hr.



Type J Gas Boiler for Hot Water or Steam

Cast iron construction for long life. Large water volume for longer steaming period, less tendency for flashing. Factory assembled controls. Completely insulated. *A.G.A.* approved for natural, mixed, manufactured or LP gases; *ASME* constructed.

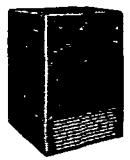
Load range: Net *I=B=R* rating, Water and Steam 375,100–3,105,600 Btu/hr.



Type G Gas Boiler

Large capacity hot water boiler engineered into new, low space-saving design—*A.G.A.* approved for closet and alcove installation. Sections and base factory-assembled for easy installation. Corrosion-resistant cast iron construction. *ASME* constructed.

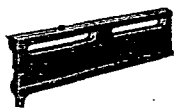
Load range: Net I=B=R rating, Water 45,000–135,000 Btu/hr.



Type H Gas Boiler for Hot Water and Steam

Medium sized unit designed for efficient combustion and thorough heat absorption. Built to *ASME* code. *A.G.A.* approved. Burns all gases including LP and LP gas-air mixtures. Safety pilot, convenient observation ports. Fully enclosing, insulated jacket.

Load range: Net I=B=R rating, Water and Steam 72,000–324,000 Btu/hr.

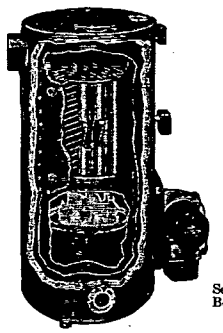


Cast-Iron Baseboards

Provide balanced combination of radiant and convected heat to assure uniform temperatures from floor to ceiling. Two heights—No. 7 and No. 9. Complete line of metal finishing accessories. Also Solray Cabinet Radiators, Raydiant Recessed Radiators, Junior Radiators.

Aldrich Company

121 E. Williams St., Wyoming, Illinois



Series "D" Boiler Burner

Boiler-Burner Units, Hot Water Heaters, Domestic and Commercial Oil Burners

The Aldrich line of vertical firetube steel boiler-burner units are *ASME* constructed and stamped, *SBI* rated and listed in rating book of *Mechanical Contractors Association*. The Series "D" units are available in ten sizes for hot water heating, steam heating and hot water supply—oil or gas fired. Boiler shipped with round jacket, insulation, flue baffles, combustion chamber and tankless coil (if required) factory installed. Burner, trim and controls packed in carton. The Aldrich Series "C"—Gulf Stream boilers are available in five sizes from 495 to 1260 sq ft water for hot water heating and hot water supply—oil or gas fired. The packaged units are furnished completely erected, prewired with equipment installed. Units are compact, efficient and economical in operation.

Aldrich ASME boilers furnished as high delivery hot water heaters are ideal for large residences, motels, institutions, swimming pools, barracks and commercial applications where large quantities of hot water are required. Available in two types—direct heating galvanized storage (HSG) and indirect submerged coil (ID) with same ratings and either oil or gas fired. Available in wide range of sizes with outputs from 118 to 950 gph at 100 deg rise.

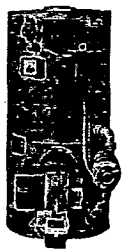
The Model E oil burner (0.5–4.0 gph) is available for domestic use and Model DX (4.5–10.0 gph) and JU (8.0–19.0 gph) for commercial applications.

SERIES "D" BOILER-BURNER SPECIFICATIONS

Boiler No.	D-120	D-160	D-220	D-280	D-360	D-450	D-530	D-610	D-720	D-890
Net Rating *SBI* Steam Sq Ft	350	450	620	790	1010	1240	1460	1690	2020	2480
Net Rating *SBI* Water Sq Ft	580	720	995	1260	1620	1980	2340	2700	3240	3960
Net Rating *SBI* Steam Btu/hr	87000	108000	149000	189000	243000	297000	351000	405000	486000	594000
Net Rating *SBI* Water Btu/hr	87000	108000	149000	189000	243000	297000	351000	405000	486000	594000
Gross Output—Btu/hr	115000	144000	198000	252000	324000	396000	468000	540000	648000	792000
Gross Input—Gas—Btu/hr	144000	180000	248000	315000	405000	495000	585000	675000	810000	990000
Firing Rate—Oil—gph	1.1	1.35	2.0	2.5	3.0	3.7	4.3	5.0	6.0	7.3
Tankless Coil Output—gpm	3½	4	4	4	5	5	6	6	8	8
Hot Water Heater (HSG) Output GPH @ 100° Rise (40°-140°F)	135	170	235	300	385	475	560	645	775	950
Storage Capacity—Boiler Gal	28	41	65	73	94	98	110	125	150	170
Heating Surface—Sq Ft	19	24	32	41	53	65	77	88	106	129
Number of Flue Tubes	16	20	24	30	34	42	52	60	71	92

SERIES "C"—GULF STREAM SPECIFICATIONS

Boiler No.	C-10	C-12	C-14	C-20	C-25
Net Rating *SBI* Water Sq Ft	495	580	720	995	1260
Net Rating *SBI* Btu/hr	74000	87000	108000	149000	189000
Gross Output Btu/hr	99000	115000	144000	198000	252000
Gross Input—Gas—Btu/hr	120000	132000	165000	250000	320000
Firing Rate—Oil—gph	1.0	1.1	1.35	2.0	2.5
Tankless Coil Output—gpm	3	3	3	4	4
Hot Water Heater (HSG) Output GPH @ 100° Rise (40°-140°F)	118	135	170	235	300
Storage Capacity—Boiler—Gallons	19	23	29	31	39
Heating Surface—Sq Ft	16	19	24	32	41
Number of Flue Tubes	20	20	20	32	32



Gulf Stream Package

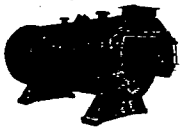
AMERICAN-Standard

KEWANEE BOILER DIVISION

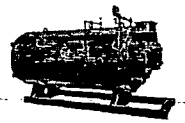
101 FRANKLIN STREET, KEWANEE, ILLINOIS

KEWANEE STEEL BOILERS ARE CONSTRUCTED IN STRICT ACCORDANCE WITH *ASME* CODE.

With a history of over 90 successful years, Kewanee Boiler Division today provides a complete line of steel fire tube boilers, both high pressure and low pressure, for heat, power and process steam in installations from coast to coast and abroad. Both Scotch type and firebox boilers are manufactured, with a total of 36 sizes of packaged boiler-burner units for oil, gas and gas-oil combination firing. *Your Kewanee Man* is always available to provide complete literature and specifications . . . solve installation problems . . . answer every question. Call him anytime.



M-800 SERIES—Compact, efficient, economical to operate. High pressure HM Series in 14 sizes, 55 to 651 hp . . . corrugated furnace as shown. Low pressure LW Series in 16 sizes, 1,372 to 21,855 MBh . . . wet-back design provides additional primary heating surface. Both available as packaged units.



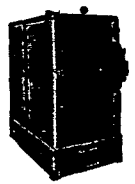
WELDED SCOTTIE JR.—Ideal for small industry, 8 sizes of welded Kewanee Scottie Jr. Boilers range in certified ratings from 18 to 92 hp in the high pressure models . . . 606 to 3,091 MBh in low pressure. Available as packaged unit with integrated burner for oil, gas or dual fuel firing.



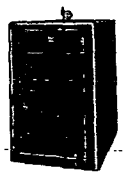
"500" SERIES—For heavy duty heating, power or process requirements of hospitals, schools, industries or apartment buildings, the steel riveted "500" Series is available in 15 sizes, 32 to 321 hp, for 125 and 150 lb wp . . . and for high-temperature hot water systems above 250 F. Two-pass tubular, firebox type.



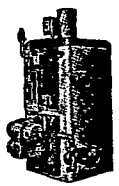
TYPE "C" SERIES—Compact boiler . . . all-welded . . . requires a minimum of floor space. Produces ample steam for heating large buildings at low cost. 15 psi steam or 30 psi water up to 250 F in 14 capacities 1,236 to 10,800 MBh. Corrugated crown sheet provides additional heating surface next to the fire. Has large firebox. Steel base.



SQUARE-HEAT TYPE "R"—Excellent source of heat for any medium size building, this one-piece, all-welded boiler fires efficiently from front or rear with any fuel. Twelve sizes range from 243 to 1,350 MBh 15 psi steam or 30 psi water up to 250° F. Available with attractive insulated jacket and packaged for oil, gas or combination oil-gas.



ROUND TYPE "R"—A rugged boiler with the capacity to serve today's larger, more extended homes. Its heavy steel plate construction assures many extra years of service. There are four Round "R" sizes from 108 to 243 MBh steam or water. Available with square or round insulated jacket.



OW AND GW SERIES—Packaged with oil or gas burner and all controls . . . for forced hot water heating of small to medium size homes. Compact design permits installation in any convenient location. Rated to produce 74 to 189 MBh. Also available as a boiler unit only, for oil or gas firing.



STORAGE WATER—Where large quantities of hot water are used at irregular intervals . . . and where exhaust or live steam is available—such as hotels, apartments, schools, hospitals, laundries, industrial buildings, etc. Capacities range from 92 to 2,240 gph. Equipped with supporting saddles.

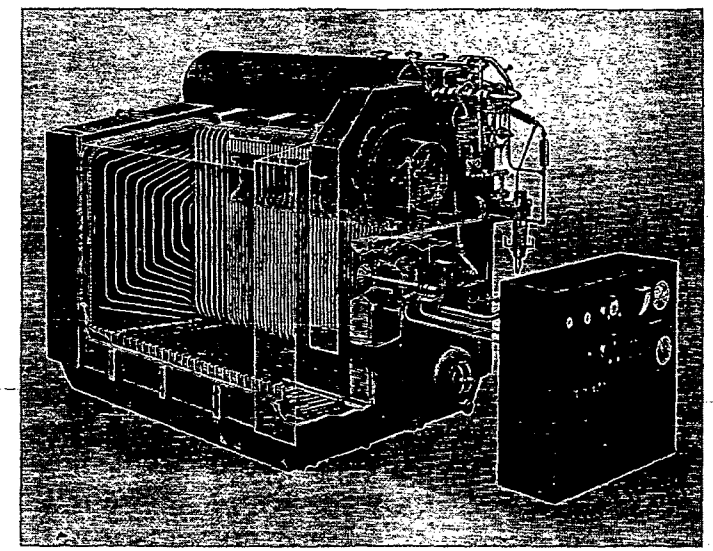
BRANCHES IN PRINCIPAL CITIES . . . STAFFED BY QUALIFIED HEATING SPECIALISTS

The Babcock & Wilcox Co.

161 East 42nd Street, New York 17, N. Y.

Offices in All Principal Cities

Water-tube Boilers for Stationary Power Plants, and Marine Services . . . Superheaters . . . Economizers . . . Air Heaters . . . Pulverized-Coal Equipment . . . Chain Grate Stokers . . . Oil, Gas, and Multifuel Burners . . . Seamless and Welded Tubing and Pipe . . . Refractories . . . Process Equipment.



B&W INTEGRAL-FURNACE BOILER, TYPE FM

Shop-Assembled Steam Generator

B&W Type FM Boiler combines the benefits of package steam with cost-saving big-boiler advantages. This compact self-contained unit has been widely applied to heating services in hospitals, schools, colleges, and other buildings. It has also gained wide acceptance throughout a broad range of industries—supplying steam for combination services.

The Type FM Boiler is particularly suitable for small- and medium-sized plants where excessive fuel consumption and costly maintenance are often serious problems. And, even for some larger steam requirements, multiple-unit Type FM installations have proved more flexible in operation and more economical than one or more large boilers requiring costly field erection and close operating supervision. A large part of the total FM capacity now in service or on order consists of multiple-unit installations.

The Type FM Boiler is fast steaming and features automatic operation with special safety provisions. It quickly responds to load changes and operates at high efficiency over the load range. This versatile unit burns gas and/or oil, separately or in combination, with quick and easy changeover accomplished at the control panel.

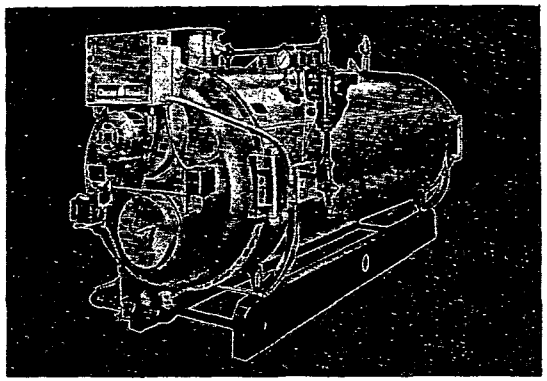
The Type FM Boiler is available in standard sizes for steam requirements to 40,000 lb per hr and for operating pressures to 235 psi. Many have been installed for higher pressures and with a moderate degree of superheat. *Details are given in Bulletin G-76—copies sent on request.*

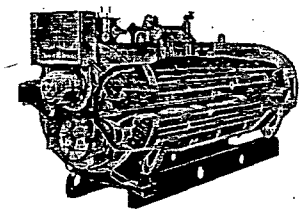
CLEAVER-BROOKS COMPANY

498 E. Keefe Avenue, Milwaukee 12, Wisconsin

CLEAVER-BROOKS Packaged Boilers for high- or low- pressure steam, or hot water . . . for heating or processing

All Cleaver-Brooks boilers meet the following design standards for improved performance and operating safety. (1) Four-pass five tube construction. Higher gas velocities extract more heat from flame, assure better heat transfer with lower fuel costs. (2) Heating surface of 5 sq ft per hp. (3) Forced draft controls air for combustion and assures proper air-to-fuel ratio for top efficiency, eliminates expensive chimney. (4) Updraft construction with low furnace keeps hottest combustion gases deep within the boiler for greater safety.





19 sizes; 130 models—oil, gas or combination oil-gas firing. Model CB boiler (shown) incorporates control panel and other controls on front for convenience; available in sizes 15–350 hp, 15–250 psi. Model LR (not shown) is available in sizes 400–600 hp, 15–250 psi.

Cleaver-Brooks BOILERS—Steam or hot water, for heating and processing 15 to 600 hp, 15 to 250 psi. Oil, gas and combination oil & gas fired models

COMPLETE SELF-CONTAINED STANDARD SIZES

HP	15	20	30	40	50	60	70	80	100
Rated Capacity—lb steam per hour (212 F)	520	690	1035	1380	1725	2070	2415	2760	3450
HP	125	150	200	250	300	350	400	500	600
Rated Capacity—lb steam per hour (212 F)	4310	5180	6900	8600	10500	12110	13800	17250	20700

CLEAVER-BROOKS BOILER ADVANTAGES

Guaranteed 80 percent efficiency—Minimum 80 percent overall efficiency at 30 to 100 percent of full rating at working pressures.

Quiet operation—Even at peak load, boilers more than meet the low sound levels required for schools, hospitals and institutions.

Nation-wide starting service safeguards your boiler investment. Cleaver-Brooks' technician starts boiler, adjusts it to optimum efficiency and trains boiler operator.

Full factory tested including complete check of control system on every boiler. Your boiler is proved ready for efficient performance on arrival at your plant.

No costly chimney—Combustion gases are carried off through a small vent, eliminating high-cost chimneys.

One complete package—A complete unit from a single source—one responsibility. Ready for operation and service connections on arrival.

Compact—Maximum steam generating capacity in limited space. Requires little floor space . . . fits easily into low head room.

Automatic, safe operation—Controls centralized and conveniently located. Modern electronic combustion controls are standard equipment.

Fast, easy maintenance—Front and rear doors are hinged or davited for fast inspection, cleaning and servicing.

Fuel flexibility—Burns heavy or light oil, gas or combination oil/gas—simple, quick changeover.

Clean—Oil or gas firing keeps boiler room and plant clean, eliminates the problem of ash removal.

Totally enclosed control panel—Control panel conveniently located, contains burner switch, relays, blower motor starter and control switches. Wiring is in conformity with the National Electric Code.

Fast installation—No special foundations needed; existing concrete floors will suffice.

Complete unit approval—Labeled by *Underwriters' Laboratories* and *Canadian Standard Association*. Conforms to ASME codes; factory tested before shipment.



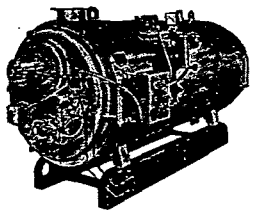
Write for free descriptive literature and specifications. Or consult the Cleaver-Brooks representative listed in your classified telephone book under "BOILERS—MANUFACTURERS & DISTRIBUTORS"

Cleaver Brooks®

One of the originators and largest producers of packaged boilers.

CLEAVER-BROOKS COMPANY

498 E. Keefe Ave., Milwaukee 12, Wisconsin



Cleaver Brooks
Progress
Packaged Heating Boiler

For commercial heating applications: apartment buildings, hotels, hospitals, stores, commercial and public buildings. Seven sizes up to 2,010,000 Btu output. Steam or hot water, gas or oil fired or combination oil/gas fired.

PROGRESS BOILER

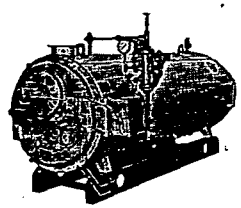
MODEL	GROSS Btu	EDR Steam GROSS	EDR Water GROSS
P-15	500,000	2085	3355
P-20	670,000	2790	4460
P-25	835,000	3480	5570
P-30	1,000,000	4170	6700
P-40	1,340,000	5580	8930
P-50	1,670,000	6970	11,150
P-60	2,010,000	8370	13,400

MONITOR BOILER

MODEL	HORSE POWER	CAPACITY LB. Steam/hr	GROSS Btu	HEATING SURFACE sq ft
M-15	15	520	500	75
M-20	20	690	670	100
M-25	25	860	835	125
M-30	30	1035	1000	150
M-40	40	1380	1340	200
M-50	50	1725	1670	250
M-60	60	2070	2010	300

Cleaver Brooks
Monitor
Packaged Boiler

Low-cost dry steam for dairies, laundries, dry cleaners, canneries, food processing and industrial applications. Seven sizes . . . 15–60 hp . . . 150 psi. Available in oil, gas or combination oil/gas models with quick fuel change over.



Cleaver-Brooks Progress and Monitor Boiler Advantages:

- **A complete package**—From one company, one source. Factory-assembled, tested and shipped as a fully self-contained unit—ready for installation, and requiring only service connections.
- **Four-pass, forced-draft design** is an efficient combination to transmit a greater percentage of heat to boiler water.
- **Field starting service**—final check and adjustment by authorized service representative after installation, plus operator training—at no extra cost.
- **Quiet-running blower** supplies combustion air—eliminates need for expensive chimney.
- **Electronic combustion-safety controls** assure safe shut-down. Electric ignition assures smooth, noiseless light-off of main flame.

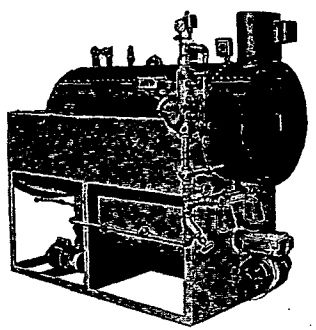
For complete details write direct or contact your Cleaver-Brooks representative listed under "BOILERS—MANUFACTURERS & DISTRIBUTORS" in your classified phone book.

COLUMBIA Boiler Company of Pottstown

Pottstown, Pa.

Makers of steel power and heating boilers

COLUMBIA'S H.R.T.* OIL OR GAS FIRED BOILER & STEAM GENERATOR



***ASME* CODE CONSTRUCTED**

Available for pressures of 15 to 250 psi.
A complete, automatic power plant wherever steam is used for processing

Columbia h.r.t. boilers are fully automatic, compact, complete steam generators. Available in sizes ranging from 2 to 100 horsepower. Designed for firing with gas, light or heavy oils. A special Columbia unit makes possible the inter-changeable use of gas or oil.

Overfiring Permitted: Conservative ratings plus large combustion volume permit overfiring. This frequently eliminates the need for additional boiler capacity, allows for plant expansion.

High Steam Output: Columbia's h.r.t. boilers have an unusually large heating surface to produce more steam at higher efficiencies. More heat is converted into steam since it is proportioned to make maximum use of hot gases of combustion for the production of steam.

Horizontal Design for longer life: The complete submersion of tubes protects them from water line corrosion.

Dry Steam Assured: Large steam space and greater liberation area virtually eliminate waste producing water carryover.

Over-All Efficiency: Factory installed, specially designed combustion chamber provides rapid heating refractory surfaces for peak combustion efficiency. Insulation, 7 inches thick in the combustion chamber proper, and 4 inches thick in the gas passage, keeps heat loss at a minimum. Special refractory brick heats up to great intensity to complete the combustion of previously unignited fuel particles. This complete combustion not only increases efficiency but makes for clean combustion. Soot accumulation on heating surfaces is minimized. Boiler tubes require less cleaning.

Safety Assured: Two pass updraft design lowers direct loss and reduces possibility of furnace pressure, pulsation or excessive combustion noises.

Accessibility and easy Maintenance: Simplified construction provides easy front and back access to the single bank of tubes. Entire unit can be gotten to for inspection of boiler and chamber by removing the covers.

SPECIFICATIONS AND DATA
COLUMBIA HRT BOILERS FOR GAS, LIGHT OIL OR DUAL FUEL
HORIZONTAL RETURN TUBULAR TYPE

	BOILER SIZE									
	6	8	10	15	20	30	40	60	80	100
Hp Rating	6	8	10	15	20	30	40	60	80	100
Length O.A. in.	58	69	69	84	84	100	100	102	124	124
Width O.A. in.	29	33	33	39	39	45	45	52	58	58
Height O.A. in.	44	50	50	64	64	68½	68½	80	96	96
Water Heating Surface sq ft	30	56	72	92	120	200	290	415	548	668
Oil Firing Rate gal. per hr	2.00	3.00	3.5	5.00	7.00	10.00	12.00	18.00	24.00	30.00
Gas Input Btu per hr	280,000	400,000	490,000	700,000	900,000	1,400,000	1,700,000	2,600,000	3,300,000	4,200,000
Approximate Output Btu per hr	200,000	300,000	350,000	500,000	700,000	1,000,000	1,200,000	1,800,000	2,600,000	3,400,000

* *Horizontal return tubular type boiler.*

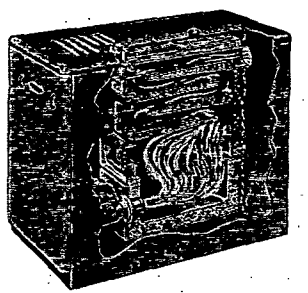
COLUMBIA Boiler Company of Pottstown

Pottstown, Pa.

Makers of steel power and heating boilers

STEEL, TUBELESS, OIL-FIRED, WATER TUBE GAS-FIRED AND COAL-FIRED HEATING UNITS

***ASME* CODE CONSTRUCTED**
***SBI* RATED**
***A.G.A.* APPROVED**



Available with highly efficient, custom engineered gas or oil burners. Columbia steel boilers, gas or oil fired, have self contained large copper instantaneous tankless coils, supplying a continuous flow of year 'round domestic hot water.

Maximum operating efficiencies assured by the balanced design of the boiler and burner; by the arrangement of the baffles which force the hot gases to travel four or five times the full length of the boiler.

One piece electric welded construction: Columbia offers a heating unit with full electric welded construction of heavy copper bearing steel, to assure lower heating cost and longer operating life.

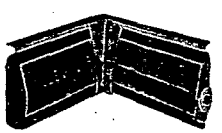
Quiet operation: Ample combustion space, the insulation and heavy steel jacket completely eliminate objectional noise.

Dual Firing: Columbia oil heating units may be easily converted for efficient gas firing.

Oil and gas burners for Columbia Boilers are designed and manufactured by Columbia to insure maximum performance of boiler-burner units and to provide undivided responsibility.

	MODEL NO.							
	L-16	L-18	L-20	L-22	L-24	*L-30	*L-32	*L-40
Ratings—Oil Fired								
Input—Gals per Hr.	1.00	1.20	1.35	1.75	2.25	3.00	4.00	7.00
SBI Net Rating								
Sq Ft Forced Water	580	720	990	1260	1620	1980	2700	4000
Sq Ft Gravity Water	510	640	880	1120	1440	1760	2400	3500
Sq Ft Steam	320	400	550	700	900	1100	1500	2200
Btu per hr Forced Water	87,000	108,000	149,000	189,000	243,000	297,000	405,000	600,000
Btu per hr Gravity Water	77,000	96,000	132,000	168,000	216,000	264,000	360,000	530,000
Btu per hr Steam	77,000	96,000	132,000	168,000	216,000	264,000	360,000	530,000

* Constructed Water Tube Only
• *Gas Fired Units A.G.A.* and *SUR* Approved for outputs from 72,000 to 172,000 Btu's per hour.
• *Coal Hand Fired* Conversion Units available in sizes L-18, L-22 and L-30



COLUMBIA BASEBOARD RADIATION

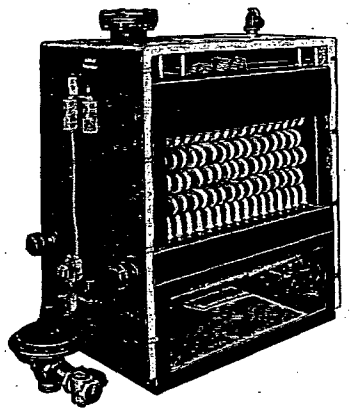
- Suspended Element—no noise!
- Tension Fastening—no rattling!
- Low Angle Heat—no drafts!
- Pre-Punched Holes—fast installation!

BRYAN STEAM CORPORATION

Chili Pike, Peru, Indiana

Steam Boilers, Hot Water Boilers and Indirect Water Heaters designed exclusively for oil or gas firing.

1. Bryan Copper Water Tube Boilers are engineered expressly for oil and gas firing. The Copper water tube construction offers these exclusive advantages:



(a) High heat conductivity of copper heating surface.

(b) Extremely rapid water circulation through the small diameter tubing.

(c) Tube arrangement causes products of combustion to be finely divided, reducing "surface film" to minimum.

(d) Relative corrosion resistance of copper.

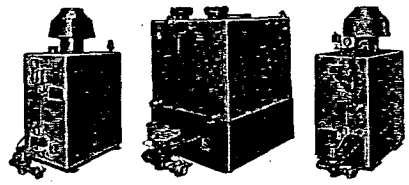
(e) Flexibility of tubes—able to withstand sudden expansion and contraction—preventing "thermal shock" damage.

(f) Tubes easily replaced—requiring no expensive and time consuming tube welding or rolling.

All Boilers are built in accordance with requirements of the *ASME* Boiler Code. Gas Fired Heating Boilers are approved by the *American Gas Association* (*A.G.A.*)

2. Domestic Heating Boilers

The Domestic Heating Boilers are available in 9 sizes, from 75,000 Btu to 600,000 Btu output. Oil and gas burners are installed at the factory as packaged units. Factory wiring is optional. Smaller models available as a completely packaged unit with all hot water circulation equipment installed.



3. Commercial Heating Boilers

The Commercial Heating Boilers (400 and 500 Series) are available in 8 sizes from 720,000 Btu to 1,800,000 Btu output, for steam and hot water radiation. These sizes are suitable for most schools, churches, small apartment houses and office buildings. Sizes up to 1,080,000 Btu are available as completely assembled and packaged units.

4. High Pressure Boilers

Bryan High Pressure Boilers are available in sizes up to 50 hp and up to 125 lb mwp. They are particularly noted for their ability to provide steam for applications requiring steam on short notice and at high efficiency. Hospitals, dry cleaning plants, laundries, dairies, tire repair shops, food processers and many others find them fine for this purpose.

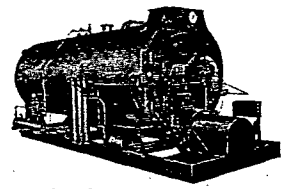
5. Indirect Water Heaters

The Bryan Indirect Water Heaters are available in sizes from 112,000 Btu to 1,800,000 Btu output. This heater combines all of the advantages of the Bryan Copper Tube Boiler with the known advantages of heating water by indirect means. Widely used for one or two temperature water in restaurants, soda fountains, hotels, apartments, laundries, industry and others. Also available as a swimming pool heater.

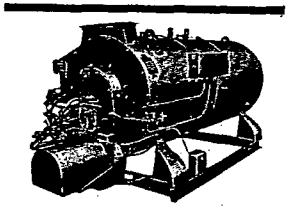
CYCLOTHERM DIVISION

NATIONAL-U. S. RADIATOR CORP.

Oswego, N. Y.



Cyclotherm Steam Generator



Cyclotherm Hot Water Generator

STANDARD BURNER ARRANGEMENTS

Package Steam Generators

Fuel	Cyclotherm Models
Light Oil, Nos. 1, 2, 3, or Diesel 28-44 API Gravity	C-500 through C-28,000
Heavy Oil, Nos. 4, 5, 6, 14-26 API Gravity	C-2,800 through C-28,000
Gas, mfg., mixed and natural	C-500 through C-28,000
Comb., light oil and gas	C-500 through C-28,000
Comb., heavy oil and gas	C-2,800 through C-28,000

Package Hot Water Generators

Fuel	Cyclotherm Models
Light Oil, Nos. 1, 2, 3, or Diesel 28-44 API Gravity	CW-7 through CW-70
Heavy Oil, Nos. 4, 5, 6, 14-26 API Gravity	CW-28 through CW-70
Gas, mfg., mixed and natural	CW-7 through CW-70
Comb., light oil and gas	CW-7 through CW-70
Comb., heavy oil and gas	CW-28 through CW-70

SPECIAL CONDITIONS

Specifications are descriptive and not a representation. Equipment described or the equivalent furnished for either steam or hot water. Cyclotherm Division National-U.S. Radiator Corp., reserves the right to alter, without notice, dimensions, designs and components of Cyclotherm units.

For complete information on how Cyclotherm gives you more steam in less space at less cost write to Cyclotherm Division, National-U. S. Radiator Corp., Oswego, N. Y. for free booklet, Cyclotherm Cyclonic Combustion, The Advanced Combustion Method that Sends Steam Costs Down.

CYCLOTHERM STEAM AND HOT WATER GENERATORS

Guaranteed Minimum 80 Percent Efficiency. With Cyclonic Combustion, a patented Cyclotherm combustion principle, 65 percent heat transfer is reached in the combustion chamber alone and 15 percent more in one pass of return tubes. 80 percent efficiency—or better—in two passes.

Compact. By eliminating superfluous tubes and passes, Cyclotherm produces maximum steam in minimum dimensions. Up to one-third smaller than other package boilers.

Low Maintenance Costs. Cyclotherm parts are easy to get at, easy to clean. Simplified construction eliminates many man-hours of maintenance.

Shipped Complete. A complete package, Cyclotherm is ready to operate when shipped. One manufacturing responsibility behind the entire job.

Automatic Controls. On models up to 60 hp, burner automatically shuts off when steam demand is satisfied, automatically turns on when more steam is needed. On larger models, electronic controls modulate firing rate so that equipment may be operated at from 30 percent to 100 percent of capacity without loss of efficiency.

No special foundation or excavation is needed. You can relocate your Cyclotherm if you change your plant layout.

No Costly Stack. Cyclotherm requires only a simple flue.

Quick Changeover. Burning oil and/or gas, Cyclotherm offers an easy and quick changeover. You burn whichever fuel is more economical.

Full power from a cold start is reached in 15 to 20 minutes.

Conforms to Codes. Cyclotherm conforms to *ASME* codes and carries *Underwriters' Laboratories* label.

Worldwide Service Facilities. Cyclotherm Sales Engineers are ready to serve you wherever you are located.

One Year Warranty. If there is any defect in materials or workmanship within one year of the date of adjustment, Cyclotherm will repair or replace the defective part without any cost to you.

CYCLOTHERM STEAM GENERATORS

Model No.	Output Btu Per hr	Steam Per hr Lbs	Equivalent Radiation Sq Ft	Overall Dimensions Height	Width Light Oil	Width Heavy Oil	Length
C-500	502,500	518	2090	3'5¼"	3'2¾"		6'0"
C-700	670,000	690	2792	4'1½"	3'9"		6'0"
C-1000	1,005,000	1035	4188	4'1½"	3'9"		7'8"
C-1400	1,340,000	1380	5653	4'6"	4'0"		8'11"
C-1725	1,675,000	1725	6980	5' 3¼"	4'2"		9'7"
C-2100	2,010,000	2070	8375	5' 3¼"	4'3"		10'6"
C-2800	2,680,000	2760	11167	5'6½"	4'8"	5'3"	12'8"
C-3500	3,350,000	3450	13958	6'1½"	5'8"	6'1"	13'8"
C-4400	4,187,500	4315	17448	6'3½"	5'10"	6'10"	14'5½"
C-5200	5,025,000	5175	20938	7'1¾"	6'6"	7'6"	16'5½"
C-7000	6,700,000	6900	27917	7'8¾"	7'1"	8'1"	18'8½"
C-8700	8,375,000	8625	34896	7'10¾"	7'1"	8'1"	20'11½"
C-10500	10,050,000	10350	41875	8'6½"	7'6"	8'6"	21'0"
C-12000	11,725,000	12075	48854	8'5¾"	7'6"	8'6"	21'0"
C-13800	13,400,000	13800	55833	8'6½"	7'6"	8'6"	21'0"
C-17500	16,750,000	17520	69792	9' ½"	7'10"	8'10"	22'1½"
C-21000	20,100,000	20700	83750	9' ½"	7'10"	8'10"	24'7½"
C-26000	25,125,000	25875	104688	10'7¼"	9'9"	10'7"	23'0"

CYCLOTHERM HOT WATER GENERATORS

Model No.	Output Btu per Hr	Overall Dimensions Height	Width Light Oil	Width Heavy Oil	Length
CW-7	670,000	46¾"	37"		79½"
CW-10	1,005,000	53½"	40"		84"
CW-14	1,340,000	58¼"	43"		92"
CW-17	1,675,000	58¼"	43"		102"
CW-21	2,010,000	58¼"	43"		115"
CW-28	2,680,000	70¾"	53"	58"	128"
CW-35	3,350,000	70¾"	53"	60"	150"
CW-44	4,187,500	80¾"	57"	70"	154¾"
CW-52	5,025,000	82¾"	57"	70"	178¾"
CW-70	6,700,000	87½"	72"	84"	183¼"

CYCLOTHERM® DIVISION NATIONAL-U. S. RADIATOR CORP.,
47 East First Street, Oswego, N. Y.

Federal BOILER COMPANY

Midland Park, New Jersey

Member SBI

Reg. U. S. Pat. Off.

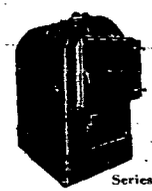
The Federal Line of Hydronic Heating Equipment is fully rated in accordance with provisions as set up by the *Steel Boiler Institute (S.B.I.) Rating Code*. Federal commercial steel boilers are constructed in conformance with rigid standards as set up by the *American Society of Mechanical Engineers*. Every boiler receives extensive testing after assembly. Each boiler also bears the *ASME* stamp of approval. Other items in the Federal Line, not illustrated here, are: Vertical steam and hot water steel residential boilers; oil and gas-fired cast iron boilers; packaged steel and cast-iron boilers; "FBC Ultrapower" baseboard radiation; and a full selection of heating accessories.



Series FD

SERIES FD—HORIZONTAL FIRE-TUBE RESIDENTIAL STEEL BOILERS:

Available in Stoker, Gas or Oil-Fired Models. 9 full range sizes to meet most residential heating needs . . . from 540 MBh Gross Output to 1800 MBh Gross Output (Net *SBI* Rating—1690 sq ft steam to 5620 sq ft steam). Features newly designed flue doors to prevent heat loss and infiltration of excess air. Rigid one piece steel base supplied . . . extra high bases for stoker fired models. *ASME* Coded and Stamped. Provides unlimited year round supply of domestic hot water.



Series FLR

SERIES FLR—INDUSTRIAL, INSTITUTIONAL AND COMMERCIAL HORIZONTAL FIRE-TUBE BOILERS:

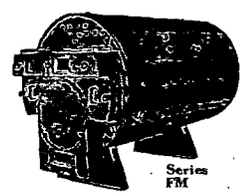
18 sizes for all ranges of heating needs . . . 792 MBh to 12,600 MBh Gross Output (Net *SBI* Rating—2490 sq ft steam to 39,370 sq ft steam). Features hi-fire box design in larger sizes to increase efficiency of boiler. Dry base supplied in conventional heights or extra high for stoker fired models. Available with oversized tankless coils for abundant year-round domestic hot water supply.



Series FLW

SERIES FLW—LOW WATER LINE BOILERS:

For industrial, institutional, or commercial applications. Oil or gas-fired. Provides lowest water line of all Federal commercial firebox boilers. New special wet leg construction completely surrounds combustion area with water walls. Reduces costly pitting and combustion chamber installations. Supplied with revolutionary windbox to provide controlled firing of modern fuels. 21 full range sizes from 540 MBh to 12,600 MBh Gross Output (Net *SBI* Rating—1690 sq ft steam to 39,370 sq ft steam). Designed for installation in limited head room areas . . . or where water, rock conditions require costly foundation installation.



Series FM

SERIES FM—SCOTCH MARINE BOILERS:

For industrial, institutional, and commercial applications. Features "wet back" design for more efficient heat transfer. Compact and rugged, FM boilers are designed for low head-room installation. Available in 14 full range sizes from 936 MBh to 7200 MBh Gross Output (Net *SBI* Rating—2930 sq ft steam to 22,510 sq ft steam). Oversize tankless coils available for year-round domestic hot water supply. Available in oil or gas-fired models, jacketed or unjacketed.

FITZGIBBONS BOILER COMPANY, INC.

New York Sales and Executive Offices 101 Park Avenue, New York 17, N. Y.

Field Sales Office and Plant, Oswego, New York

PRODUCTS—STEEL HEATING BOILERS AND BOILER UNITS for all fuels and all heating systems. Capacities to meet requirements of any building. Built and rated according to *ASME* and *SBI Codes* and *"Hartford"* inspected.



4000 SERIES RESIDENTIAL BOILERS

Four sizes
108,000 to 243,000 Btu/hr oil fired
Four sizes
99,000 to 195,000 Btu/hr gas fired



"PB" PACKAGED BOILER

4 sizes
1975 MBtu/hr to 3113 MBtu/hr
Oil only



7700 SERIES RESIDENTIAL BOILERS

Four sizes—oil fired
99,000 to 198,000 Btu/hr
77G1 and G2
Two sizes gas fired
72,000 and 93,000 Btu/hr



"P" PACKAGED BOILER

4 sizes
1975 MBtu/hr to 3113 MBtu/hr
Light oil—oil/gas—gas



"R-Z-U" JUNIOR BOILER

11 sizes
297 to 1350 MBtu/hr
Steam or water
Gas, oil or coal fired



"SCOTCH" TYPE BOILER

16 sizes
1392 MBtu/hr to 15,430 MBtu/hr
Steam or water
Gas or oil fired



"RW" HIGH PRESSURE BOILER

For high pressure forced hot water systems
16 sizes
1080 MBtu/hr to 15,430 MBtu/hr
Gas or oil fired



"SCOTCH" TYPE BOILER

Available also for burner attachment at installation or Fitzgibbons factory



"R" TYPE PACKAGED BOILER

13 sizes
1975 MBtu/hr to 15,530 MBtu/hr
Gas or oil fired



"D" TYPE BOILER

19 sizes
929 MBtu/hr to 15,430 MBtu/hr
Steam or hot water
Gas, oil or coal fired
Also available in low-set model for special installations

Descriptive catalogs on all these Fitzgibbons boilers are available upon request

HYDROTHERM INC

NORTHVALE, NEW JERSEY

PACKAGED AUTOMATIC GAS HEATING PLANTS

Providing Space or Volume Water Heating for Residential, Commercial and Industrial Uses

45,000 BTU/hr to 3,600,000 BTU/hr input ratings



HYDROTHERM

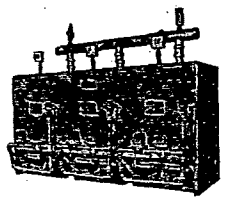
Model R-150
with Hammeroid finish jacket;
425 lbs, 16 in. x 31 in. x 27 in.

HYRDOTHERM boilers are of all cast iron construction and are exclusively designed for gas. The patented absorption unit, composed of deep-rib horisontal sections connected in sigzag, permits maximum heat transfer in minimum space with small water content. This "true" gas design results in low fuel consumption, quick heat response, uniform heat transfer, self cleaning flue passages, and small installation space. HYDROTHERM boilers are *A.G.A.*-approved for all gases and are factory assembled and wired with safety and gas controls, ready for quick installation. Boiler and controls are enclosed in an attractively finished insulated steel jacket.

All boilers are tested at 250 lb hydrostatic pressure and carry a 100 lb *ASME* rating which permits direct connection of boilers to city water lines as required for volume water heating applications.

MULTI-TEMP

Multi-Temp
Model R-900; 1800 lbs,
65 in. x 26 in. x 37 in.



MULTI-TEMP, a new concept in heavy duty gas-fired hot water heating, provides any desired capacity with a multiple of HYDROTHERM units, each equipped with its own controls. True input modulation is accomplished by making boiler units respond to the return water temperature through aquastats with differential settings in the return manifold. At a cold start, all units go on; as the return water temperature rises, one or more units drop out according to their own aquastat settings till the input is just enough to maintain the design water temperature, effecting a year-round fuel saving of up to 20 percent.

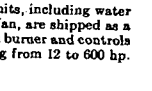
MULTI-TEMP, being composed of multiple units, offers extra protection against service interruption. Each boiler unit is shipped completely assembled to simplify field installation.

CAPACITY RANGE OF HYDROTHERMS

	Model No.	Space Heating *A.G.A.* Ratings						Volume Water Heating Gallons per Hour for Temperature Rise Shown									
		Natural, Mixed and Manufactured Gas			L. P. Gas (Propane)			Natural, Mixed and Manufactured Gas					L. P. Gas (Propane)				
		input 1000 Btu/hr	output 1000 Btu/hr	I-B-R installed radiation*	input 1000 Btu/hr	output 1000 Btu hr	I-B-R installed radiation*	60°	80°	100°	120°	140°	60°	80°	100°	120°	140°
SINGLE UNITS	M-45	45	36	180	42	33.6	168	73	54	43	36	30	67	50	40	33	28
	M-75	75	57.6	288	73	57.6	288	115	86	69	57	49	115	85	69	57	49
	M-85	85	68	340	85	68	340	135	102	81	68	58	135	102	81	68	58
	R-100	100	80	400	100	80	400	160	120	96	80	68	160	120	96	80	68
	R-120	120	96	480	110	85	440	193	144	115	96	82	178	133	105	88	75
	R-150	150	120	600	145	116	580	240	180	144	120	103	223	174	139	116	99
	R-180	180	144	720	180	144	720	289	215	173	145	124	289	215	173	145	124
	R-210	210	168	840	210	168	840	337	253	203	169	145	337	253	203	169	145
	R-250	250	200	1,000	250	200	1,000	400	300	240	202	173	400	300	240	203	173
	R-300	300	240	1,200	290	232	1,160	480	360	288	240	206	464	348	278	232	198
MULTI-TEMP SINGLE JACKET	R-360	360	288	1,440	360	288	1,440	580	430	350	290	250	580	430	350	290	250
	R-420	420	336	1,680	420	336	1,680	680	510	410	340	290	670	510	400	340	290
	R-500	500	400	2,000	500	400	2,000	800	600	480	400	340	800	600	480	400	340
	R-600	600	480	2,400	580	464	2,320	960	720	580	480	410	930	700	560	470	400
	R-750	750	600	3,000	750	600	3,000	1210	900	720	600	520	1200	910	730	600	520
	R-900	900	720	3,600	870	696	3,480	1450	1080	870	720	620	1400	1050	840	700	600
MULTI-TEMP MULTIPLE JACKET	R-1000	1,000	800	4,000	1,000	800	4,000	1610	1210	960	800	690	1610	1200	960	800	690
	R-1200	1,200	960	4,800	1,160	928	4,640	1930	1450	1160	960	830	1860	1400	1120	930	800
	R-1500	1,500	1,200	6,000	1,450	1,160	5,800	2410	1810	1450	1210	1030	2330	1750	1400	1160	1000
	R-1800	1,800	1,440	7,200	1,740	1,392	6,960	2890	2170	1740	1450	1240	2800	2100	1680	1400	1200
	R-2100	2,100	1,680	8,400	2,030	1,624	8,120	3370	2530	2020	1690	1450	3260	2450	1960	1630	1400
	R-2400	2,400	1,920	9,600	2,320	1,856	9,280	3850	2890	2310	1930	1650	3720	2800	2240	1860	1600
	R-2700	2,700	2,160	10,800	2,610	2,088	10,440	4340	3250	2600	2170	1860	4190	3140	2520	2100	1800
	R-3000	3,000	2,400	12,000	2,900	2,320	11,600	4820	3610	2890	2410	2070	4660	3490	2800	2330	2000
	R-3300	3,300	2,640	13,200	3,190	2,552	12,760	5300	3980	3180	2650	2270	5130	3840	3070	2560	2200
	R-3600	3,600	2,880	14,400	3,480	2,784	13,920	5780	4340	3470	2890	2480	5590	4190	3350	2800	2400

* These ratings are based on piping and pickup factors established by "*The Institute of Boiler Manufacturers.*" When piping loses are negligible, output may be increased 12 per cent, as in panel and baseboard heating, etc.

OTHER FAMOUS HYDROTHERM PRODUCTS



Hydro-Fin



Hydro-Chiller

Hydro-Vector

THE INTERNATIONAL BOILER WORKS CO.

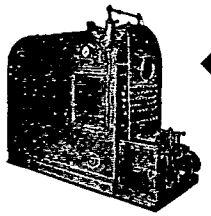
500 Birch Street · East Stroudsburg, Pa.

A Complete Line of WATER TUBE Heating and Power Boilers

International water tube design assures high heat absorption and exceptionally low operating cost with all fuels. Boiler water circulation is positive and directed, delivering a constant payload of steam or hot water to the boiler nozzle. Every International Boiler is built to *ASME* code requirements, and is tested and approved by a nationally recognized insurance company before shipment. Full access is provided for cleaning BOTH the fireside and waterside of all tubes without dismantling the boiler.

PACKAGE BOILERS

...fully automatic and factory coordinated.

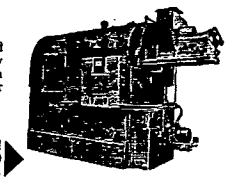


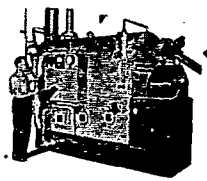
the COMPAK - Forced Draft Series

Forced draft operation eliminates the need for an induced draft fan or separate controls. Available as Type FPL for low pressure steam or hot water heating and as Type FPH for high pressure steam generation in sizes from 12 to 750 hp. Write for *Bulletin 1400.*

the COMPAK - Induced Draft Series

. . . noted for high thermal efficiency, quiet operation and convertibility to coal firing. Type PL units (15 lb steam, 30 lb water) . . . and Type PH units (130, 160 or 200 psig) are built in in sizes from 12 to 600 hp. Write for *Bulletin 600.*



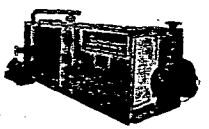


the COAL-PAK AUTOMATIC - Bituminous Coal Fired

First FULLY AUTOMATIC package unit complete with combustion controls, automatic coal feed and ash removal system for burning LOW COST BITUMINOUS COAL cleanly and efficiently. Type PSL (15 lbs. steam, 30-160 lb water) and Type PSH (130, 160 or 200 psig) in sizes from 71.6 to 300 hp. Write for *Bulletin 1100.*

the THERMOJET-Oil/Gas Fired

This LOW HEADROOM forced recirculation package unit features famous LaMONT boiler design specifically adapted for SINGLE BUILDING low or high temperature hot water heating systems. Type TJW for operating pressures from 30 to 600 psig in standard sizes from 250,000 to 14,000,000 Btu/hr. Write for *Bulletin 1000.*



HEATING BOILERS—15 lb Steam—30 lb Water

TYPE C—built in *SBI* rated sizes from 2,190 to 51,000 sq ft steam. Can be equipped with an immersion coil of instantaneous or storage tank type for hot water service requirements. Write for *Bulletin 800.*

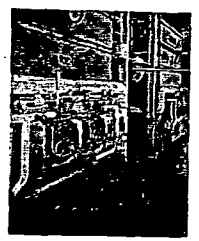
TYPE IDL & BBL Boiler-Burner Units are built in capacities from 1,820 to 84,000 sq ft steam. Write for *Bulletins 500 and 900.*

HIGH PRESSURE BOILERS—130—160—200 psig

TYPE CH Of all welded construction these boilers are available in standard sizes from 8 to 363 hp—oil, gas or coal fired. Write for *Bulletin 1500.*

TYPE IDH & BBH Boiler-Burner Units, including water tube boiler, steel base and induced draft fan, are shipped as a unit ready for installation of oil and/or gas burner and controls at job site. Built in standard sizes ranging from 12 to 600 hp. Write for *Bulletins 400 and 900.*

INTERNATIONAL-LAMONT . . . .



HIGH TEMPERATURE WATER GENERATORS

Discharge Temperatures to 430 F. Design Pressures to 600 psig

BUILT-IN ECONOMIZER permits safe use of temperature differentials to 200 F with no thermal shock. LOW HEADROOM reduces cost of new boiler house construction and permits installation in many existing structures. TYPE LFW available in standard sizes from 3 to 200 million Btu/hr. Completely packaged oil/gas fired units to 60 million Btu/hr. Write for *Bulletin 700.*

THERMAL LIQUID HEATERS

Safe, low pressure industrial process heat to 750° F. . . . plus temperature control to ±2 F. Ideal for use with Aroclor, Dowtherm, Mobiltherm and other thermal liquids.

TYPE TJH and LFH available in standard sizes from 150,000 to 50 million Btu/hr. Completely packaged units in sizes to 20 million Btu/hr. Write for *Bulletin 1300.*



See Your Local INTERNATIONAL DISTRICT REPRESENTATIVE or write for Bulletins above

Johnston Brothers, Inc.

Established 1864

Ferrysburg, Michigan

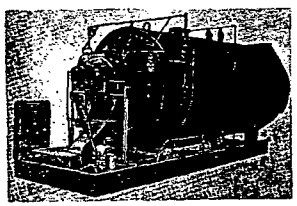
Telephone — Grand Haven 3125

Member SBI

Reg. U. S. Pat. Off.

PACKAGED BOILER UNITS
FULLY AUTOMATIC
OIL OR GAS OR COMBINATION
OIL AND GAS

FORCED DRAFT
THREE PASS AND TWO PASS
COMPLETELY ASSEMBLED AND
FIRE TESTED AT FACTORY

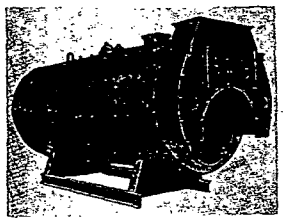


HIGH PRESSURE TYPE from 75 to 750 hp and pressures of 125, 150 and 200 lb psi (250 lb available as a special).

LOW PRESSURE TYPE for heating; 15 lb pressure, and EDR rating from 2190 sq ft to 42,500 sq ft steam.

ASME Code construction. *Underwriters' Laboratory* and *Factory Mutual* Approved. Specification Forms in detail to exactly cover the requirements of any particular job will be furnished upon request and without any obligation whatever. Ask for *Bulletins 509-B, 9600.*



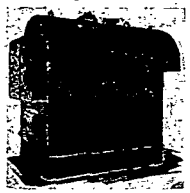


POWER, PROCESS, HEATING

15 to 400 hp and pressures 15 lb to 200 lb psi. For mechanical firing with Coal, Gas or Oil. Overloads up to 200 percent of rating readily developed. Oil or Gas firing equipment and controls can be installed for completely automatic operation and in emergency or fuel scarcity, can be readily converted to Coal firing.

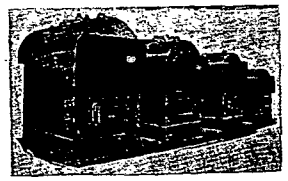
Ask for *Bulletin 4000 and 9500*





Firebox Heating Boiler, Compact Type. Oil, Gas, Stoker or Hand Firing. Three-Pass, 15 lb pressure, *ASME* Code. Capacities 2190 to 42,500 sq ft, *SBI* rating. Ask for *Bulletin 1500.*

Also built for High Pressure (125 psi). *Ask for Bulletin 702.*



WATERBACK DESIGN—Special attention is directed to this valuable feature. The rear combustion chamber is entirely submerged within the boiler water, eliminating use of firebrick, tile, or rear end refractories, thus avoiding the necessity of frequent repair and replacement costs.

SERVICE—Located in each sales area is a factory trained Service Engineer, capable of checking all phases of operation—adjusting controls and equipment to insure continuous maximum performance. Johnston service is prompt, liberal and cooperative—it is available to all customers.

ORR & SEMBOWER, INC.

Builders of Dependable Boilers Since 1885

Reading, Pa.

World-wide Factory-trained Sales and Service Representatives

Complete ONE-SOURCE Responsibility

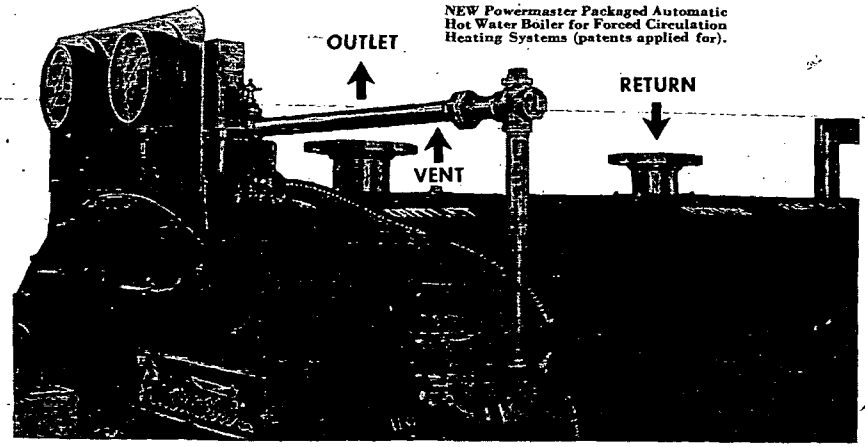
Powermaster and POWER-PAK Packaged Automatic Boilers

Fuel Burning Systems—oil, gas and combination oil/gas

Powermaster boilers are modern, completely factory-assembled, automatic units with a world-wide reputation for highly dependable and efficient performance in public buildings, hospitals, schools, apartments, industrial plants and other installations. Three standard models are available in sizes of 20 hp to 600 hp for steam generation to 250 psi, and for hot water to 30 psi. All models are designed and built to *ASME* Boiler Code requirements, approved by *Underwriters' Laboratories, Inc.*, and fully guaranteed by Orr & Sembower, Inc. All units are factory fire-tested and afford these advantages:

- Simple cost-saving installation
- High efficiency at all loads
- Instant response to load swings
- 3-pass boiler with 5 sq ft of heating surface per bhp
- Forced draft firing
- Quick fuel change-over
- Clean, quiet operation
- Fully automatic operating and safety controls
- Convenient accessibility for inspection and servicing
- Nation-wide factory-trained service

NEW Powermaster Packaged Automatic Hot Water Boiler for Forced Circulation Heating Systems (patents applied for).



Powermaster Hot Water Boiler—Powermaster is one of the first hot water heating boilers developed and constructed specifically for modern institutional, commercial and industrial forced circulation heating applications.

The *Powermaster* hot water boiler design and construction utilizes a dip-tube outlet located at the top of the boiler shell, designed to eliminate entry of air and vapor. The return connection, located at the center on the top of the boiler, is baffled to direct return water from the system down around the outer surface of the boiler. An integral vent connection is provided on the boiler.

The O & S *Powermaster* hot water boiler is not simply a steam boiler converted to hot water service. Its provisions for insuring efficient and effective internal circulation are the result of two years of concentrated research and development. These four advantages of the Powermaster packaged automatic hot water boiler emphasize its importance for every forced circulation hot water system:

1. **Powermaster hot water boiler offers more protection against thermal shock.**
2. **Powermaster internal circulation path eliminates short circuiting.**
3. **Powermaster internal circulation system eliminates stratification.**
4. **The hot water Powermaster insures uniform temperature gradients throughout the boiler.**

Write for information on the Powermaster packaged automatic boiler to meet your specific needs.

Raypak Company, Inc.

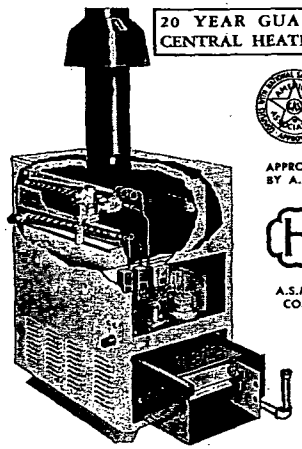
2416 Chico Avenue, El Monte, California

THE ADVANCED LINE OF GAS FIRED 100% COPPER AND BRONZE WATER TUBE HOT WATER BOILERS.

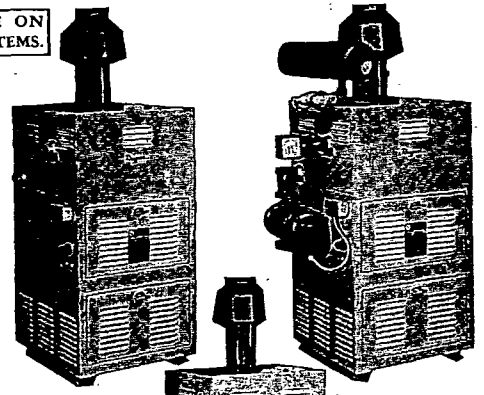
20 YEAR GUARANTEE ON CENTRAL HEATING SYSTEMS.

APPROVED BY A.G.A.

A.S.M.E. CODE



Raypak boiler cutaway with HWTG controls and Unatherm Governor.(PAT. PENDING)



160-T thru 1200-T extended jacket standard.



Models — TP typical pre-assembled package for central heating.

80-T and 120-T with flush jacket.

T MODELS cover standard boilers for central fluid heating, volume hot water supply and sanitizing 180F hot water applications. They feature a readily cleanable heat exchanger with individual bronze plugs for each tube, extended jacket, and gas modulation governed by water temperature. Temperature ranges to 240F.

TP MODELS cover the "plug-in" central heating package which Raypak developed and for which Raypak is so well known. They feature, in addition to T Model specifications, a high head circulator, heavy duty expansion tank with air charger, air vent, automatic make-up facilities, tridicator gage, relief valve, transformer relay (where applicable)—all wired and piped. The 80-TP Special, not a plug-in package, is an 80-T with pump, tank and accessories shipped loose.

HWTG MODELS include fully automatic swimming pool heaters. They feature Raypak's Unatherm Governor which prevents damaging condensate and virtually eliminates scaling—automatically—by maintaining 105-110F. boiler outlet temperatures regardless of flow rate changes from the filter system. In addition they feature a built-in pool aquastat, a self-contained independent electrical system and a low water cut-off device; no outside wiring—indefinite heat exchanger life, permitting no cost do-it-yourself scale removal.

All models feature extended and integral copper fin tubes, tapered bronze distribution headers, large ID inter-connecting bronze header in the combustion chamber, monolithic cast lightweight refractory, heavy 18 gage stressed-skin jacket, light weight (3 times the BTU's per pound of weight), compactness, A.G.A. approval on combustible flooring, and noiseless stainless steel ported burners with drawer-type mounting pan.

RAYPAK TRI-TEMP SYSTEM (patents pending) is a package for the food serving industry providing three temperatures of water from one boiler and includes glass-lined storage tank and heater coil.

NOTES

NEW! Interconnecting bronze manifold design prevents air cavitation in heat-exchanger.

NEW! Heat exchanger jacket panel permits boiler water tubes to be inspected and cleaned without breaking pipe connections.

Raypak stainless steel burners permit the same propane ratings as for natural gas on every size boiler.

160 lb. A.S.M.E. working pressure on all models.

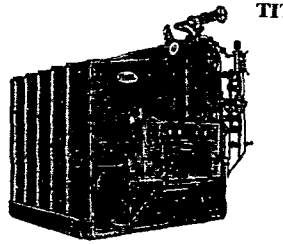
Standard heat exchangers tested at 1000 psi Hydrostatic pressure on request.

Model	MBH Input	MBH Output	EDR Sq. Ft. Gross	Approx. Wt. #
80-T	80	64	425	150
120-T	120	96	642	165
160-T	160	128	853	220
225-T	225	180	1200	250
310-T	310	248	1600	325
400-T	400	320	2133	400
600-T	600	480	3200	500
800-T	800	640	4270	700
1200-T	1200	960	6400	1000

TITUSVILLE IRON WORKS

DIVISION OF STRUTHERS WELLS CORPORATION

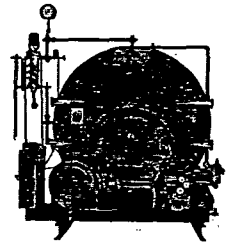
Titusville, Pennsylvania



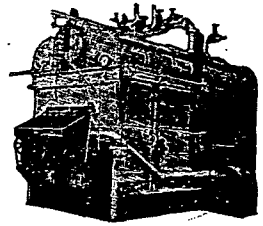
TITUSVILLE TYPE WTP PACKAGED STEAM GENERATOR

This compact, shop-assembled water-tube steam generator features simplicity and economy of operation, from 10,000 to 60,000 lbs of steam per hour. It is shipped as a package complete with firing equipment, controls and all other components installed ready for service connections in the field. Enclosed and protected by all-welded, pressure-tight steel casing, the unit is easy to move by lifting or skidding, and erection costs are reduced to a very low minimum. Forced draft, pressurized furnace operation eliminates need for stack or induced draft fan. The Titusville WTP Steam Generator is designed to use heavy oil, heavy oil and gas combination; light oil; light oil and gas combination; natural gas; manufactured gas; coke oven gas. Fully automatic welding is employed, welds are X-rayed, and drums stress-relieved. *Bulletin B-3255.*

TITUSVILLE 3-PASS Scotch Marine Boilers



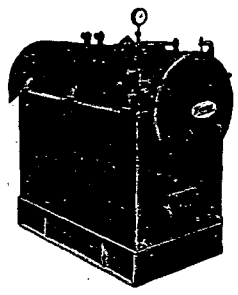
The Titusville 3-Pass Scotch Marine Boilers for power (S-3P) and heat (S-3H) are compact to the limits of practical size while maintaining excellent thermal qualities. Featuring complete wet-back construction, eliminating refractory problems, these modern boilers have water-cooled rear combustion chamber, large primary heating surface area, 3-pass tubes for maximum efficiency in heat extraction. Heat release rates of less than 88,000 BTU/cu ft of furnace volume at maximum continuous output ratings assure easy firing by all leading types of forced draft packaged burners. Pleasing appearance joins with practical design for ample access to all sections of boiler for inspection and maintenance. *ASME* power and heating boiler codes followed throughout in design and construction. *Bulletin B-3333.*



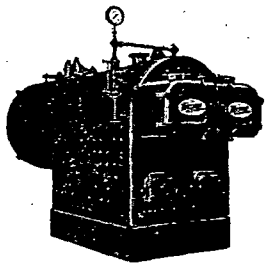
TITUSVILLE TICOTHERM STEAM GENERATORS

These large, highly efficient water tube steam generators are economical in operation and compact in design. They are constructed to meet or exceed a safety factor of 5, and drums are fusion welded, X-ray tested and stress relieved to meet or exceed ASME requirements as well as all local and state codes. Excellent results are achieved with any type of fuel. Ticotherm Steam Generators are completely insulated—top, sides and rear—with a minimum of 7 inches of refractory and insulation at side walls; boilers are completely steel encased. Four passes of gas travel are provided by vertical baffling, resulting in a minimum of draft loss and maximum heat absorption. Soot blowers are furnished as standard equipment on all boilers except gas-fired models. *Bulletin B-3250-A.*

TITUSVILLE ALL-WELDED FIREBOX BOILERS High and Low Pressures



The Titusville Compact Steel Heating Boiler (right) is compact in design, requiring less boiler room floor space than any other type. Excellent operating economies are obtainable, due to the unit's modern 3-pass construction feature. Built for use with all fuels and all types of firing—ample combustion space is provided to economically burn oil, gas, or coal. These all-steel boilers are manufactured in 19 different sizes, rated in accordance with latest rating code of The Steel Boiler Institute, constructed using ASME code rules. *Bulletin No. B-3000-B.*



Type WP Titusville All-Welded Firebox Boiler (left) is all-welded, utilizing advanced welding methods, quality steels and modern fabricating techniques. A streamlined boiler of exceptional strength, it has no riveted seams or screwed stays, remains tight under all operating conditions. Self-cleaning crown sheet and ample clean-out facilities are included in the design. These boilers combine high efficiency and low maintenance in service, and are readily adaptable for burning all sizes of bituminous and anthracite coal, or, in the WPO Model, for using any recognized oil, gas or combination gas-oil burners. *Bulletin No. B-3050.*

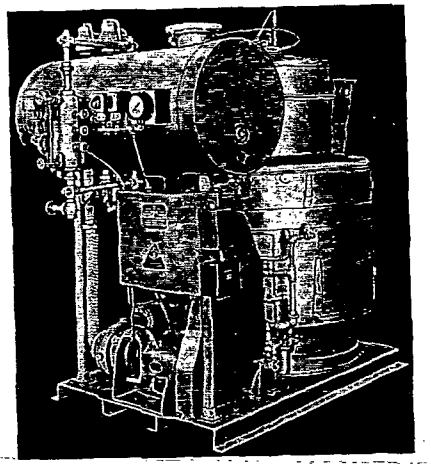
Write for the descriptive bulletins in which you are interested—let us arrange to have our nearest representative call to consult on your heating or power boiler requirements—without obligation of any kind.

VAPOR HEATING CORPORATION

80 East Jackson Boulevard · Chicago 4, Illinois

New York · St. Paul · Denver · Washington · Philadelphia · Atlanta
San Francisco · Houston · Richmond · Los Angeles · St. Louis

VAPOR MODULĀTIC WATER-TUBE BOILERS

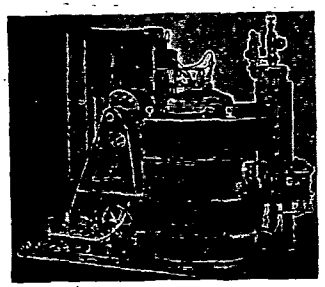


DRUM TYPE

Dependable, compact, Vapor Drum Modulātics are designed to furnish high or low-pressure steam, with high or low temperature hot water for heating, power, processing, countless production applications. ¼ the size and weight of conventional boilers of comparable ratings—Modulātics provide 4 bhp per sq ft of floor space. Largest size takes only 40 sq ft, fits easily in unused corners or aisles. Lowest installation cost—Modulātics are delivered as complete units factory assembled, wired and job-tested, ready to connect and fire up. Largest unit fits through plant doors. No special foundations or chimneys required.

Operation is fast, positive, automatic. Full steam in 5 minutes, or instant hot water, from cold start. Tracks fluctuating steam demand up or down with speed and accuracy.

Multiple-unit Modulātic installations can have a single, automatic coordinated control that cuts whole units in or out as steam requirements vary, and modulates between these steps. Several smaller units always operating at design capacity minimize the inherent inefficiency of idling larger boilers as steam demand slackens.



Modulātic boilers *never require replacement* because design permits easy, simple replacement of water-tube coils, or other parts, for lifetime peak efficiency. Built-in automatic safeguards against no water, flame failure, excessive pressure. All Modulātics conform to *ASME* Power-Boiler Code and *ASME* Heating-Boiler Code.

DRUM-TYPE SPECIFICATIONS IN BRIEF

Six Sizes	20 to 200 hp
Pressures	0 to 15 psi 5 to 150 psi
Fuel	Oil, Gas, or Combination
Floor Area (200 hp)	5 ft x 8 ft
Height (200 hp)	7 ft 8 in.
Floor Load (max.)	150 lb/sq ft
Operation	Automatic

INDUSTRIAL TYPE

Heavy-duty Modulātic boilers are made in nine sizes to 150 bhp and pressures to 1000 psi. These industrial boilers have all Modulātic advantages of compactness, quick steaming, flexibility, and high operating efficiency with reliable, automatic performance and minimum maintenance. Thousands of Modulātics have proved themselves in tough, demanding service throughout the world. Literature is available upon request for both drum type and industrial type Modulātics.

INDUSTRIAL-TYPE SPECIFICATIONS IN BRIEF

Eight Sizes	10 to 150 hp
Pressures	2 to 1000 psi
Fuel	Oil, Gas, or Combination
Floor Area (150 hp)	5 ft x 8 ft
Height (150 hp)	6 ft, 10 in.
Floor Load (max)	150 lb/sq ft
Operation	Automatic

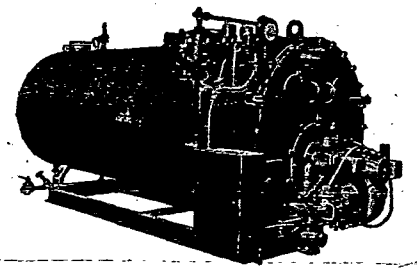
WESTERN BOILER COMPANY

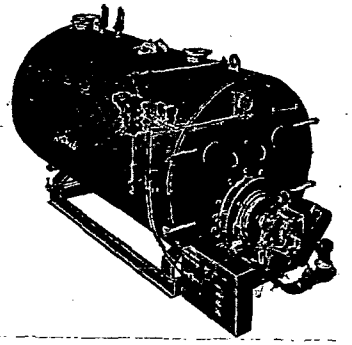
1600 North Indiana Street, Los Angeles 63, California
1 Vanderbilt Avenue, New York 17, New York

Detailed on this page are a few of the many features of Webco-Ray packaged heating, power and process steam boilers. Webco-Ray 3-pass boilers combine in a *single guaranteed package, the skills and experience* of two leaders in heating and power equipment. All units are assembled, pre-wired and test fired at Western Boiler *Company prior to shipment*, making possible immediate hookup at the job site.

Component parts are integrated for balanced, economical performance and dependability. Forced Draft design eliminates unsightly, costly, tall stacks, an important advantage in todays modern buildings.

Sizes from 10 to 600 boiler horsepower, 335,000 to 20,085,000 Btu/hr for oil, gas and combination oil-gas operation.





PACKAGED PRESSURE FORCED DRAFT: These ruggedly constructed Webco-Ray packaged boilers are ideally suited for fully automatic forced draft operation in the 10 to 100 *boiler hp*, 335,000 to 3,348,000 Btu/hr range. Units are available for light oil, natural gas or combination gas-oil operation. All are guaranteed to provide quiet, efficient and economical performance. With **Webco-Ray Pressure Forced Draft** units, a single motor provides complete power for both the fan and oil pump with resultant savings in electricity. The single fan furnishes all air necessary for combustion and to overcome draft loss through the boiler. Air is supplied under constant pressure unaffected by atmospheric conditions. These forced draft features eliminate the need for tall stacks *or other auxiliary draft equipment*. Only a small vent is needed to carry off spent gases. Burners meet all requirements of the *National Electric Code* and *Underwriters' Laboratories*.

PACKAGED ROTARY FORCED DRAFT: *This highly efficient* **Webco-Ray** packaged boiler is *fully automatic*, incorporating the **Webco-Ray Forced Draft System.** It is designed and built to comply with the requirements of the *ASME Code* and displays the appropriate Code symbols. Sized from 40 to 600 boiler hp, 1,339,000 to 20,085,000 Btu/hr, units are available to achieve efficient combustion of light and heavy fuel oils—high or low pressure gas. Combination models fire either oil or gas. Webco-Ray boilers feature large furnace volumes for high radiant heat transfer, quiet operation, longer life and minimum maintenance.

Forced Draft—The Webco-Ray forced draft system enables this unit to operate efficiently on all fuels regardless of external atmospheric conditions. Forced draft firing eliminates the need for costly, unsightly tall stacks or overfire draft controls. Only a small vent *is required to carry off spent gases.* The belt-driven forced draft fan can be easily adjusted to compensate for varying altitude factors.

Size	BOILER OUTPUT EDR Steam Sq Ft	EDR Water Sq Ft	Heat Output X 1000 Btu/hr	Oil Firing Rate GPH	Gas Firing Rate CFH	Size	BOILER OUTPUT EDR Steam Sq Ft	EDR Water Sq Ft	Heat Output X 1000 Btu/hr	Oil Firing Rate GPH	Gas Firing Rate CFH
10	1390	2230	335	2.8	418	115	16000	25600	3845	32.0	4810
15	2090	3350	502	4.2	628	125	17500	27900	4184	34.9	5230
20	2790	4460	669	5.6	837	150	21000	33500	5021	41.8	6276
25	3480	5570	836	7.0	1046	175	24500	39100	5858	48.8	7323
30	4180	6690	1002	8.4	1255	200	28000	44600	6695	55.8	8369
40	5580	8920	1339	11.2	1674	225	31400	50200	7525	62.7	9400
50	7000	11200	1674	14.0	2092	250	35000	55800	8369	69.7	10461
60	8400	13400	2009	16.8	2511	300	42000	67000	10043	83.7	12553
70	9800	15600	2343	19.5	2929	350	49000	78100	11716	97.6	14645
80	11200	17900	2678	22.3	3348	400	56000	89300	13390	111.6	16737
90	12600	20100	3013	25.1	3766	500	70000	111600	16738	139.5	20922
100	14000	22300	3348	27.9	4184	600	84000	133900	20085	167.4	25106

Capacities based upon Fuel Oil at 150,000 Btu/gal, gas at 1000 Btu/cu ft, and boiler efficiency of 80 percent

ARMSTRONG MACHINE WORKS

851 Maple St., Three Rivers, Mich.

GUARANTEED: SATISFACTION OR YOUR MONEY BACK

Traps Available from Local Stocks of 45 Factory Representatives 135 Jobbers, or Through the Supplier of Your Choice, or Direct from Factory

Open Float and Thermostatic Traps for Low Pressure Heating Service

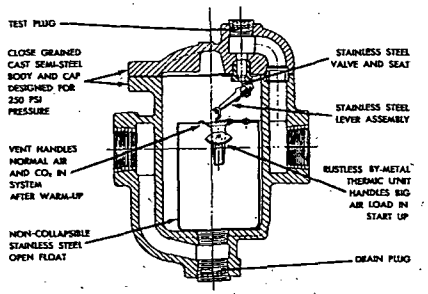
Armstrong O. F. & T. traps answer the specific needs of low-pressure intermittent service where large amounts of air accumulate when the steam is off—especially good for unit heaters, pipe coils and standing radiation.

Design is open-float, single valve, with bi-metal controlled air vent. Conventional float and thermostatic connections—¾ in. to 2 in.

Mechanisms are of the same design, materials and workmanship as those used in Armstrong forged steel inverted-bucket traps for high-pressure, high-temperature service. Bodies and caps are close-grained cast semi-steel, designed for pressures to 250 psig.

Further information in *Bulletin No. 775.*





FEATURES

1. Low First Cost—on the basis of actual capacity for discharging hot condensate. No. 860 (¾ in.) has a capacity of 640 lb/hr at 15 psi, lists at $12. Other capacities to 6800 lb/hr.

2. Easy Installation—horizontal straight-through pipe connections minimize fittings and labor. Also, traps can be equipped with internal check valves.

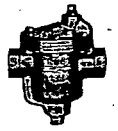
3. Long Life and Low Maintenance—Armstrong O. F. & T. traps are non-air-binding and self-scrubbing. Working parts are corrosion-proof. Open-float is non-collapsible. Inadvertent admission of high pressure steam will not harm trap. Traps hydraulically tested at 500 psig.

4. Automatic Air Elimination—during warm-up, thermostatic vent is wide open to discharge large amounts of air rapidly. Regular vent handles normal air and/or CO_2 in system after heat-up, venting it at steam temperature.

5. Fast Condensate Removal—condensate is discharged at steam temperature—as fast as it enters trap. No cooling leg needed.

6. No Steam Loss—valve parts are always water sealed, and cannot be nicked by dirt or scale. Excellent for vacuum service.

ARMSTRONG INVERTED-BUCKET STEAM TRAPS



Side Inlet, Side Outlet



Bottom Inlet, Top Outlet



Side Inlet, Bottom Outlet



Integral Strainer

For draining steam mains and all types of institutional and industrial steam-heated equipment—including hot water heaters, and kitchen, hospital and laundry equipment. High capacity for size due to time-tested two-phase leverage system. Sizes for every capacity need. Cast semi-steel models may be used for pressures to 250 psi. Forged steel models available for higher pressures.

Features: basically the same as those outlined above for O. F. & T. traps except for the extra large air-handling capacity—air is discharged through the bucket vent.

Choice of body styles and built-in accessories minimize fittings and installation labor. Each type available with thermic vent for handling faster heat-up and/or internal check valves. Integral strainer traps are available which cost less to buy and install than standard trap plus separate strainer. Following table lists types, connections, and maximum capacities of hot condensate:

Type	Connections in.	Max. Cap. (lb/hr)
Side Inlet-Side Outlet	½ to 1¼	7500
Bottom Inlet-Top Outlet	½ to 2	19000
Side Inlet-Bottom Outlet	½ or ¾	570
Integral Strainer	½ to 1¼	3500

Other Armstrong traps available with capacities up to 300,000 lb/hr.

Complete details in *Catalog K.*

Y-TYPE STRAINERS

Semi-steel: screwed, ¼ in.–2 in.; flanged, 2 in.–6 in.

Carbon steel: screwed or socket weld, ½ in.–3 in.; flanged, 2½ in.–6 in.

Chrome moly: screwed or socket weld, ½ in.–2 in.

Bronze: screwed or silbras, ½ in.–2 in.

Type 316 stainless: screwed or socket weld, ½ in.–2 in.



Y-Type Strainer



Air Relief Trap

AIR RELIEF TRAPS

Ball-float traps for venting air from high points in hot-water systems. Corrosion-proof stainless-steel mechanisms. Smallest size is No. 21 (shown above)—½ in. or ¾ in. connections, 6¼-in. diameter, capacity of 10 cfm at 30 psi, list price $16.00. No. 21G, same capacity as 21 but fits between studs.

MUELLER STEAM SPECIALTY CO., INC.

29 Meserole Avenue, Brooklyn 22, New York

MUELLER

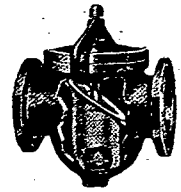
STRAINERS FOR ALL SERVICES



No. 251

251 SELF CLEANING "Y" TYPE

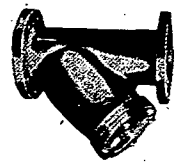
¼ in. through 3 in. Cast Iron, Cast Steel, Bronze and Aluminum. Pressures up to 600 lb steam.



No. 155

751 FLANGED SELF CLEANING "Y" TYPE

Sizes 1-½ in. through 16 in. Larger sizes available on application. Available in Cast Iron, Cast Steel, Stainless Steel and Bronze. Pressures up to 600 lb steam.



No. 751

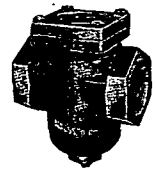
155 and 165 BASKET TYPE

Basket-Type Strainers available with screwed or flanged ends, in Cast Steel, Cast Iron, Stainless Steel and Bronze. Furnished with closed or open bottom baskets.

Baskets lined with mesh are available where very fine screening is required. Drain plug at bottom of strainer easily removed for blowdown of accumulated sediment.



No. 165



No. 175

145 BASKET TYPE

Sizes ½ in. through 3 in. Iron Body and Bronze Body screwed end removable Basket-Type Strainers.

Screens and Baskets made from a variety of perforated metals and wire mesh available for all types of strainers.

Cast Carbon Steel and Stainless Steel Basket Strainers are available for gasoline and jet fueling applications . . . In full accordance with U.S. Government Specifications.

Every Mueller Strainer is individually tested and fully guaranteed

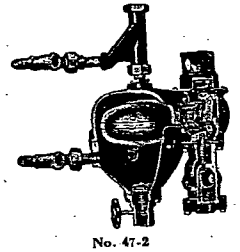
PRESSURE DROP CHARTS ON MUELLER STRAINERS AND 1959 CATALOG ARE AVAILABLE ON REQUEST.

McDONNELL & MILLER, Inc.

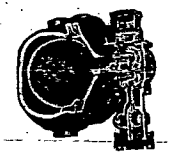
3500 N. Spaulding Ave., Chicago 18, Ill.

Doing One Thing Well

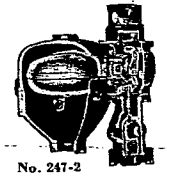
McDonnell Feeder Cut-off Combinations



No. 47-2



No. 51-2



No. 247-2

No. 47-2 is for steam boilers used in closed heating systems, up to 5000 sq ft capacity. Feeder *adds water* automatically to boiler as needed, to maintain a safe level at all times. Cut-off switch stops burner if water level drops into emergency zone. Features include *cool feed valve*, deep sediment chamber, self-closing blow-off valve, Quick Hook-up, straight thrust stainless steel valve, packless construction. Maximum steam pressure, 25 lb. Maximum water supply pressure, 150 lb. Shipping wt., 30 lb.

No. 47 Boiler Water Feeder. Same as No. 47-2 but without No. 2 Cut-off Switch, *for hand-fired boilers*. Shipping wt, 29 lb.

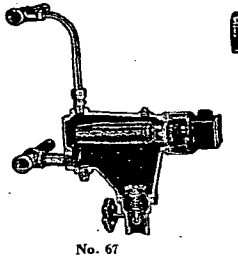
No. 51-2 is for steam boilers above 5000 sq ft capacity. Same basic features as No. 47-2 above, but has larger capacity and is installed with 1 in. equalizing pipes. Maximum steam pressure, 35 lb. Maximum water supply pressure, 150 lb. Shipping weight, 39 lb.

No. 51 Boiler Water Feeder. Same as No. 51-2 but without No. 2 Cut-off switch, for hand-fired boilers. Shipping weight, 38 lb.

No. 247-2 is for hot water space heating boilers. Hot water boilers can run into low water difficulties as a result of leakage, prolonged discharge by the *ASME* relief valve, or even failure to fill system properly. The No. 247-2 takes care of such situations by adding water as needed, and by stopping the burner should the water level drop into the emergency zone. Same as No. 47-2, but specially adapted for hot water service. Installed with 1 in. equalizing pipes. Maximum boiler pressure, 30 lb. Maximum water supply pressure, 150 lb. Shipping weight, 24 lb.

No. 247 Boiler Water Feeder. Same as No. 247-2, but without No. 2 Cut-off switch, for hand-fired boilers. Shipping weight, 23 lb.

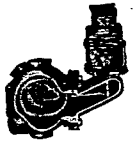
McDonnell Low Water Fuel Cut-offs



No. 67



No. 69

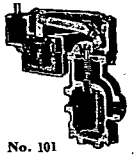


No. 63

No. 67 is for steam boilers of any size; maximum steam pressure, 20 lb. *Dependably stops* burner when water level falls into danger zone. Has deep sediment chamber, packless construction, Quick-Hook-up, and extra switch which closes on small drop without stopping burner to operate alarm or control No. 101 Electric Water Feeder. Shipping wt, 11 lb.

69 Series "Built-ins" fit 2½ in. tappings provided in most modern jacketed boilers. Individual models in the series—No. 69, 169, 269, 369, 469, 569 and others—vary principally in length of mounting barrel, are specified by boiler make and model number. Same operating mechanism as No. 67. Shipping wts. vary from 4½ to 9 lb.

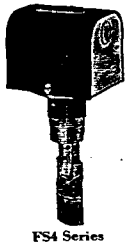
No. 63 is designed especially for use on hot water space heating boilers. Handles boilers of any size, pressures to 50 lb. Stops burner and prevents damage to boiler if low water condition occurs. (Can also be used on steam boilers.) Installed with 1 in. equalizing pipes. Shipping wt, 14 lb.



No. 101

McDonnell No. 101 Electric Water Feeder

Used with No. 67 and "Built-in" cut-offs—adds the convenience of automatic water feeding. For boilers to 5000 sq ft. Maximum steam pressure, 25 lb; water supply pressure, 150 lb. Water tappings, ½ in. Shipping wt, 8 lb. For gas boilers with low voltage control circuits order No. 101-24V, complete with transformer. Shipping wt, 12 lb.



FS4 Series

McDonnell FS4 Series Flow Switch

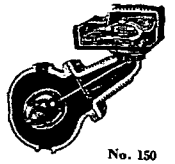
Dependable, moderately priced device that makes or breaks electrical circuit when flow of liquid in a pipe starts or stops. Used to turn on signal or alarm, or start a motor, burner or metering device. Parts contacting liquid are copper, brass or phosphor bronze. Segmented paddle easily adapted for 1 in. or larger pipe. Models available to make or break circuit with flow. Shipping wt, 3 lb.

McDONNELL & MILLER, INC.

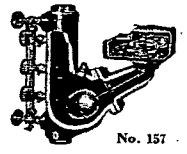
3500 N. Spaulding Ave., Chicago 18, Ill.

Doing One Thing Well

McDonnell Pump Controllers and Low Water Cut-offs



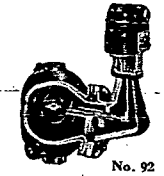
No. 150



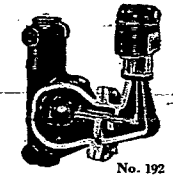
No. 157

No. 150 is for steam boilers of any size. Maximum steam pressure, 150 lb. Most widely known and used control of its type. All operating parts are away from heat. No stuffing box to bind or leak. Has *two-switch* construction. One starts and stops pump as boiler calls for water; second operates on greater drop of water line to stop burner and/or complete alarm circuit. Has 1 in. equalizing tappings. Shipping weight, 31 lb.

No. 157 is same as the No. 150 above, but with water column type float chamber that greatly simplifies installation on boilers having separate water columns. Has tappings for ¾ in. water glass cocks, 1 inch equalizing pipes and ½ in. tri-cocks. Shipping weight, 50 lb.



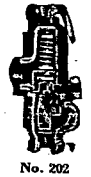
No. 92



No. 192

No. 92 *is for boilers of any size.* Maximum boiler pressure, 250 lb. New control on the market. Introduces the principle of repulsion magnetic switching. Remarkable dissipation of heat permits use of 75 C wire in junction box, as now required by *Underwriters' Laboratories*. Installed with 1¼ in. equalizing pipes. Shipping weight, 64 lb.

No. 192 is same as the No. 92 above, but with water column type float chamber that greatly simplifies installation on steam boilers having separate water columns. Body includes tappings for ½ in. gage glass and tricocks, and for 1¼ in. equalizing pipes. Shipping weight, 69 lb.



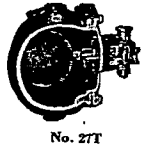
No. 202

McDonnell 202 Series Pressure-Temperature Relief Valve

For hot water tanks and heaters. Bears the *ASME* symbol of approval, with pressure relieving capacity Btu rated by *National Board*. Also listed by *A.G.A.* All connections are ¾ in. internal tappings. Features special connection for flow to fixtures for easier hook-up, better operation; if use of this is not feasible order NF models.

McDonnell Valve No.		Opening Pressure	Output Btu./hr.	Shipping Wt.
With Fixture Connection	Without			
202-75	202NF-75	75	415,000	4
202-100	202NF-100	100	532,000	4
202-125	202NF-125	125	650,000	4

McDonnell Make-up Water Feeders



No. 27T

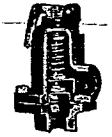


No. 21

No. 27T teams up with McDonnell pump controls to provide the one good way to control water level in steam boilers. Pump control calls for water as the boiler itself dictates. No. 27T makes sure there is always an adequate minimum supply in the receiving tank. Has large feeding capacity; stainless steel valve; all brass valve body. For external hook-up with 1 in. equalizing pipes on existing receivers. Feed water tappings, ¾ in. Shipping weight, 39 lb.

21 Series Teams up with McDonnell pump controls. Makes sure there is always an adequate minimum supply in receiver when pump control on boiler calls for water. Mounts right on receiver, and feeds directly into it. Two standard flange sizes fit most existing tank openings: *No. 21* has six bolt holes on 5¾ in. bolt circle, *No. 221* on 8½ in. circle. Feed water tappings, ¾ in. Shipping weights, 15 and 26 lb.

McDonnell 230 and 240 Series Pressure Relief Valves



No. 230-¾ in.-30

Comply with *ASME* Boiler Code in *every respect; tested and Btu rated by National Board of Boiler and Pressure Vessel Inspectors*. Basic construction of all valves is the same, except 240 Series has outlet one pipe size larger than inlet. All operating parts of non-*corrosive* materials. Independent-action testing levers cannot hamper normal operation.

For Hot Water Space Heating Boilers

McDonnell Valve No.	Boiler Size Btu./hr.	Inlet	Outlet	Opening Pressure	Shipping Weight
230-¾ IN.-30	303,000	¾	¾	30	3
240-¾ IN.-30	513,000	¾	1	30	3
230-1 IN.-30	743,400	1	1	30	4
240-1 IN.-30	910,000	1	1¼	30	5
230-1¼ IN.-30	1,025,100	1¼	1¼	30	6
240-1¼ IN.-30	1,560,000	1¼	2	30	7¼
240M-3	2,313,000	2	1 & 2	30 & 25	15
240M-4	2,710,000	2	1¼ & 2	30 & 25	16
240-2 IN.-30*	2,180,000	2	2½	30	26
240-2 IN.-25*	2,550,000	2	2½	25	25

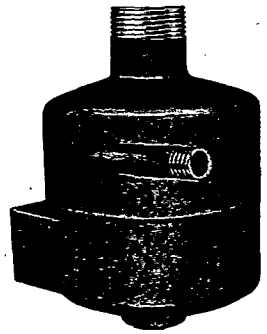
* Recommended: 25 lb. opening valve, with smaller 30-lb. valve for thermal expansion.

For Hot Water Tanks and Heaters

McDonnell Valve No.	Opening Pressure	Capacity Btu./hr.	Inlet	Outlet	Shipping Weight
230-¾ IN.-75	75	502,000	¾	¾	3
230-¾ IN.-100	100	640,000	¾	¾	3
230-¾ IN.-125	125	783,000	¾	¾	3
230-¾ IN.-150	150	925,000	¾	¾	3
240-1 IN.-75	75	1,320,000	1	1¼	5
240-1 IN.-100	100	1,810,000	1	1¼	5
240-1 IN.-125	125	1,960,000	1	1¼	5
240-1 IN.-150	150	2,320,000	1	1¼	5
2240-75	75	2,320,000	2	1¼ (2)	15
2240-100	100	3,220,000	2	1¼ (2)	15
2240-75	75	3,780,000	2	1¼ (2)	19
2240-100	100	4,830,000	2	1¼ (2)	19

WILSON ENGINEERING CORPORATION

6 North Michigan Avenue, Dept. HG59, Chicago 2, Illinois



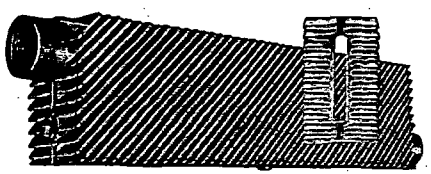
WILSON
CENTRIFUGAL BLOWDOWN SEPARATOR

SOLVES BOILER BLOWDOWN DISPOSAL PROBLEMS

Here is the safe, economical way to handle boiler blowdown . . . at atmospheric pressure. Steam flash is instantaneously cleared through the large top vent pipe. Water and sludge fall by gravity through bottom drain. No objectionable noise. Install where convenient, inside or out. One Wilson Separator can handle two or more boilers in battery, if desired. Special models with flanged or butt welding connections available.

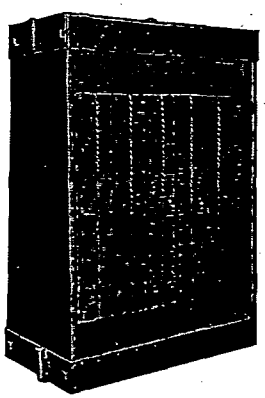
• MEETS REQUIREMENTS OF A.S.M.E. UNFIRED PRESSURE VESSEL CODE

• SIMPLE TO INSTALL • REQUIRES NO MAINTENANCE • THOUSANDS IN SERVICE



WILSON ALL-CAST ALUMINUM RADIATION ELEMENTS

Wilson All-Cast Aluminum Directional-finned Radiation Elements are readily adapted to many applications. The patented 45° directional fin feature and the chamber shape provide for compact installation and high thermal efficiency in heat transfer problems. Extremely rugged in construction. High pressure hydrostatic tested.



WILSON UNIT HEATERS

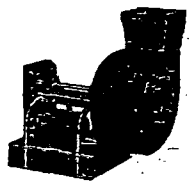
Provides large volume of warm air over wide floor areas. Completely dependable in service where corrosive atmospheres and maintenance costs are prime factors. Ask for specifications.

WHITTY COMPANY, INC.

86 Western Avenue — Boston 34, Mass.

MANUFACTURERS OF INDUCED DRAFT FANS, DAMPERS, AND DRAFT CONTROLS

INDUCED DRAFT FANS



A, B, D Chassis

Used in lieu of tall chimneys; used to improve the combustion efficiency of automatic burners; used to correct inadequate draft.

Furnished in 22 models with capacities to 24,000,000 Btu boiler output. A size to match every boiler and heater in the Commercial-Industrial range.

Fan housing is mounted on a rigid structural chassis in either a clockwise or counter-clockwise rotation; with a choice of vertical, horizontal, or angular discharge.

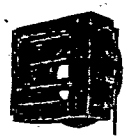
See "Time-Saver" chart below for application data.



K Chassis

MULTI-BLADE DAMPERS

Inlet boxes with factory installed Dampers are available for use with power-operated draft regulators. Add Suffix "T" to model number.

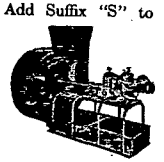


Multi-Blade Dampers

Optional apparatus for regulating over-fire draft is available for shipment with Fans. Power actuators may be ordered for mounting to dampers at the factory.

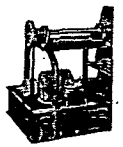
STEAM DRIVE

Steam-turbine Drive has built-in speed reducer to maintain the same slow speed feature as the motor-driven models. Add Suffix "S" to model number.



Steam Drive

MECHANICAL TURNDOWN



Mechanical Turndown

Turning handwheel varies Fan speed through an automatic pulley device while maintaining constant motor speed. Add Suffix "J" to model number.

DRAFT MONITOR

Prevents start-up of burner unless boiler draft is proved adequate and cuts off burner whenever the draft becomes inadequate. Specify DM-3.



Draft Monitor

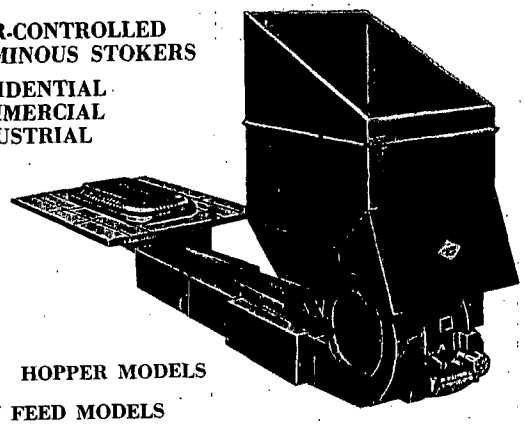
"Time-Saver" Chart for Induced Draft Fans

DRAFT FAN Whitty Model Number	BOILER RATING SBI Table One	SBI Table Two	ASME HP	Radiation Net Ft	BOILER OUTPUT Gross BTU 1000	FIRING DATA Coal Hr. Lb	Gas Hr. Ft	Oil Hr. Gal	FLUE GAS CFM 600 F	SP WG In.	SQUARE BREECHING OR CHIMNEY 1 FAN	2 FANS	3 FANS	SQUARE FRESH AIR INTAKE 1 FAN	2 FANS	3 FANS
6K12		88	15	1666	500	50	715	5	400	.50	8	10	12	6	8	10
7K16		153	24	2666	800	80	1140	8	640	.50	9	12	15	7	10	13
8A16	215	177	30	3333	1000	100	1430	10	800	.60	10	13	16	8	11	14
8A25	358	238	35	4000	1200	120	1716	12	960	.40	11	14	17	8	11	14
9A25	358	294	45	5000	1500	150	2140	15	1200	.40	12	15	18	9	12	15
9A33	429		54	6000	1800	180	2570	18	1440	.40	12	15	18	10	14	17
10B33	503		63	7000	2100	210	3000	21	1680	.40	13	16	19	11	15	15
10B50	503		75	8333	2500	250	3570	25	2000	.60	14	18	21	12	16	20
12B50	715		90	10000	3000	300	4285	30	2400	.60	15	20	24	13	18	22
12B75	833		111	12333	3700	370	5280	37	2960	.60	15	20	24	14	19	24
14B75	1073		135	15000	4500	450	6430	45	3600	.60	16	22	28	15	21	26
14B100	1250		156	17333	5200	520	7420	52	4160	.60	19	24	28	17	24	29
16B100			174	19333	5800	580	8280	58	4640	.60	19	25	30	18	25	31
16B150	1429		210	23333	7000	700	10000	70	5600	.60	21	28	32	19	27	33
18D150	1788		225	25333	7600	760	10860	76	6080	.70	22	28	32	21	29	36
18D200	2143		275	30666	9200	920	13140	92	7360	.70	23	29	35	22	31	37
20D200	2500		318	35333	10600	1060	15140	106	8480	.80	25	32	37	24	34	41
20D300	2857		366	40666	12200	1220	17420	122	9760	.80	26	34	41	27	39	44
22D300	3214		405	45000	13500	1350	19300	135	10800	.80	27	36	43	27	39	47
22D500	3571		500	55500	16700	1670	23800	167	13350	.80	30	39	47	29	41	50
24D500			570	64000	19200	1920	26300	192	15360	.90	32	43	50	32	45	56
24D750			700	77300	22200	2220	35000	222	17760	.90	34	45	54	34	53	65

Example: Model 16B150 indicates size 16 Fan with 16 in. round inlet and 16 × 16 in. square outlet, mounted on a B size chassis and driven by a 1½ hp motor. Dimensions in inches.

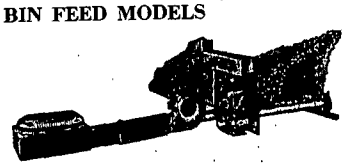
AIR-CONTROLLED BITUMINOUS STOKERS

- RESIDENTIAL
- COMMERCIAL
- INDUSTRIAL



HOPPER MODELS

BIN FEED MODELS



Hopper Models

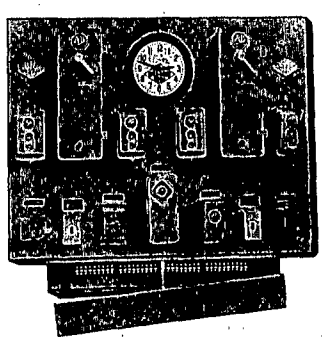
Patented automatic air control insures delivery of correct quantities of air to meet constantly changing fire-bed conditions, eliminates need for banked firing methods and prevents back-smoking. If overload occurs, shear pin and automatic switch protect stoker against damage. Strong, cast iron hopper base with heavy gage rust-resisting steel hopper. Will-Burt hopper type stokers are available in 20 to 750 lb per hr capacities.

Bin Feed Models

Like the hopper models, Will-Burt bin-feed stokers offer all the advantages of automatic air control and the patented protective shear pin and switch. Patented flange and set screw locking device prevent coal screw from disengaging from drive shaft. Precision machined power unit. Capacities from 20 to 750 lb per hr.

PRE-WIRED PANEL CONTROL BOARDS

Prewired, pretested control panel boards provide an interconnected control system for stokers to be operated by either a common operating and/or individual holdfire control, depending on the setting of the safety switch, off-on switch and timeswitch cycle. Mounting wall brackets are provided. Not illustrated, but available are control cabinets with controls mounted and wired on a subpanel and completely enclosed in heavy steel cabinet.



REFERENCE MANUAL . . . *yours to use!* Specifications, drawings, technical data and recommended applications. Write for free copy.

STOKER RATING TABLE

* Load values given are based on coal having a heat value of 12,000 Btu per lb. (Density 48 lb per cu ft) and an overall efficiency of boiler and stoker of 65 percent to and including 100 lb/hr and 70 percent for those stokers over 100 lb/hr. Full-time operation.

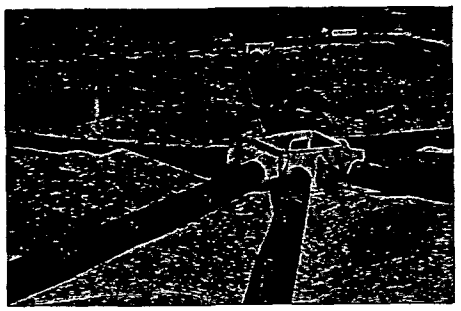
STOKER NUMBER	20	30	45	65	75	100	125	131	150	200	250	300	350	400	500	600	750
COAL BURNING RATE Approximate Lb Per Hr, Continuous Operation	10 15 20	15 22 30	22 33 45	28 45 65	30 50 75	40 70 100	50 88 125	60 105 150	60 105 150	80 140 200	100 175 250	120 210 300	140 245 350	160 280 400	200 350 500	240 420 600	300 525 750
GROSS OUTPUT 1000's Btu per hour at Boiler Nozzle or Furnace Bonnet*	156	234	351	507	585	780	1,050	1,260	1,260	1,680	2,100	2,520	2,940	3,360	4,200	5,040	6,300
MIN. COMB. VOL.—Cu Ft.	4	6	9	13	15	20	25	30	30	40	50	60	70	80	100	120	150
MIN. SETTING HEIGHT—In's. Hearth to Crown Steel Boilers	18	19¼	21	23	24	27	28	28½	28½	30	31½	33	34½	36	39	42	46½
Cast Iron Boilers	16	17	18½	20	21	23½	24	25	25	26¼	27½	29	30	31½	34	37	40½
MOTOR SIZE—Hp (Hopper Model)	¼	¼	¼	½	½	½	½	¾	¾	¾	1	1	1½	2	2	3	5
MOTOR SIZE—Hp Power Unit (Bin Feed Model)	¼	¼	¼	¼	¼	¼	¼	½	½	½	½	½	½	½	¾	¾	1
Fan	One Only	One Only	One Only	½	½	½	½	½	¾	¾	¾	¾	1	1½	1½	2	3
HOPPER CAPACITY (Approx. lb)	350	350	350	600	600	600	600	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,200
FAN—CAPACITY IN cfm No. 20, 30, 45 @ 1½ in. S.P. All others @ 2 in. S.P.	385	535	535	660	660	1,320	1,320	1,320	1,405	1,405	1,875	1,875	2,720	2,720	2,720	2,720	4,080
COAL SCREW—Std. Hopper Model, Overall Length (In.)	61	61	61	69	70	72	74	92	93	93	95	99	101	101	104	107	125
COAL SCREW—Dia. (In.)	2⅜	2⅜	2⅜	3 5/16	3 5/16	3 5/16	3 5/16	3 5/16	4⅞	4⅞	4⅞	4⅞	4⅞	4⅞	4⅞	4⅞	4⅞
COAL SCREW REMOVAL Std. Hopper Model—Min. Clearance Req'd. (In.)	16	16	16	25	26	28	30	13	15	15	15	18	20	20	23	26	44
WEIGHT CRATED (Approx. Lb) STD. HOPPER MODEL Refractory	560	560	560	760	810	850	900	1,020	1,470	1,470	1,500	1,680	1,750	1,750	1,810	1,840	2,005
Deadplate							1,040	1,160	1,760	1,760	1,805	2,000	2,210	2,210	2,380	2,440	2,670
STD. BIN FEED MODEL Refractory	625	625	625	800	850	900	950	985	1,590	1,590	1,650	1,830	1,900	1,900	1,960	2,010	2,070
Deadplate							1,090	1,125	1,880	1,880	1,955	2,150	2,360	2,360	2,530	2,610	2,735

Write for new 8-page Bulletin No. W-307 . . . contains installation drawings, size tables, photos and descriptions. Complete stoker reference manual sent on request.

SONOCO PRODUCTS COMPANY

CONSTRUCTION PRODUCTS DIVISION
Hartsville, South Carolina

LA PUENTE, CALIF. • MONTCLAIR, N. J. • AKRON, IND. • LONGVIEW, TEXAS • ATLANTA, GA. •
BRANTFORD, ONTARIO • MEXICO, D. F.



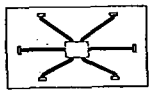
SONOCO SONOAIRDUCT® FIBRE DUCT

for slab floor perimeter heating or combination heating and cooling systems

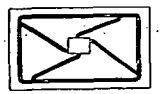
Sonoco Sonoairduct is a round, lightweight fibre duct that is aluminum foil lined and wrapped with black, asphalt filled, duplex kraft paper. It is especially designed for use as supply and return lines in gas and oil fired, slab floor perimeter heating or combination heating and cooling systems. It can also be adapted for use in spot cooling and ventilating systems. Sonoairduct meets and exceeds *F.H.A.* criteria and test requirements for products in this category.
Sonoairduct is aluminum foil lined for lower co-efficient of air friction. The air friction loss of 8 in. Sonoairduct is 0.151 in. of water per 100 ft at a velocity of 800 ft per minute and 70 F. Sonoairduct has a heat conductivity of 0.46 Btu per hr per sq ft per deg F per inch of thickness. This low heat loss prevents hot spots in the slab above the encased duct.
Sonoairduct possesses a sound-deadening quality which minimizes transmission of noise from furnace and from room to room. As the wall thickness increases with diameter size, Sonoairduct is highly resistant to crushing.
Sonoairduct Fibre Duct is available in 23 sizes—2 in. to 36 in. I.D. Unless other lengths are specified, shipments will be made in standard lengths of 18 ft. It is easy to cut to exact length with a hand or power saw on the job. Long lengths mean faster leveling and fewer joints to make and seal. Sonoairduct fits all standard metal bends, elbows, T's and registers. All installations should be made in accordance with Sonoairduct Manual No. 557, which is available upon request.

SONOAIRDUCT SIZES AND WEIGHTS

Inside Diameter	Wall Thickness	Approx. Weight Per 1,000 Feet	Approx. No. Ft. in 48 Ft. Cartoned
2"	.125"	373	103,033 ft.
3"	.125"	538	41,850 ft.
4"	.150"	934	23,056 ft.
5"	.150"	773	17,330 ft.
6"	.175"	1,068	11,898 ft.
7"	.200"	1,411	9,144 ft.
8"	.200"	1,606	8,994 ft.
9"	.225"	2,020	8,345 ft.
10"	.225"	2,227	4,500 ft.
11"	.225"	2,456	3,402 ft.
12"	.225"	2,676	3,034 ft.
14"	.250"	3,445	2,143 ft.
16"	.300"	4,688	1,620 ft.
18"	.300"	5,223	1,170 ft.
22"	.375"	7,249	954 ft.
23"	.375"	7,571	792 ft.
24"	.375"	8,685	723 ft.
28"	.375"	9,795	578 ft.
29"	.375"	10,108	414 ft.
30"	.400"	11,525	414 ft.
32"	.400"	13,253	324 ft.
34"	.400"	13,043	306 ft.
36"	.400"	14,135	306 ft.



For warm air perimeter loop systems



For warm air perimeter radial systems

CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION AND CAPACITY

Side Rectangular Duct	6	7	8	9	10	11	12	13	14	15	16	17	18	19
6	6.6													
8	7.5	8.2	8.6											
10	8.4	9.1	9.8	10.4	10.9									
12	9.1	9.9	10.7	11.3	11.9	12.5	13.1							
14	9.8	10.7	11.5	12.2	12.9	13.5	14.2	14.7	15.3					
16	10.4	11.4	12.2	13.0	13.7	14.4	15.1	15.7	16.3	16.9	17.5			
18	11.0	11.9	12.9	13.7	14.5	15.3	16.0	16.6	17.3	17.9	18.5	19.1	19.7	
20	11.5	12.5	13.5	14.4	15.2	15.9	16.8	17.5	18.2	18.8	19.5	20.1	20.7	21.3
24	12.4	13.6	14.6	15.6	16.6	17.5	18.3	19.1	19.8	20.6	21.3	21.9	22.5	23.2
28	13.2	14.5	15.6	16.7	17.7	18.7	19.6	20.5	21.3	22.1	22.9	23.6	24.4	25.0
30	13.6	14.9	16.1	17.2	18.3	19.3	20.2	21.1	22.0	22.9	23.7	24.4	25.2	25.9
40	15.3	16.8	18.2	19.4	20.7	21.9	23.0	24.0	25.1	26.0	27.0	27.9	28.8	29.7
50	16.8	18.4	19.9	21.3	22.7	24.0	25.2	26.4	27.6	28.7	29.8	30.8	31.8	32.8
60	18.1	19.8	21.4	23.0	24.5	25.8	27.3	28.7	29.8	31.0	32.2	33.4	34.5	35.5
70	19.3	21.0	22.8	24.5	26.1	27.6	29.1	30.4	31.8	33.1	34.3	35.6	36.8	37.9

Reprinted by permission from Heating Ventilating Air Conditioning Guide 1954, Chapter 23

1933

AMERIVENT

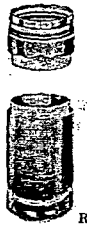
A Division of American Metal Products Company, Inc.
6100 Bandini Blvd., Los Angeles 22, Calif.

AMERIVENT double-wall GAS VENT PIPE and FITTINGS

For Gas-Fired Water Heaters, Boilers, Furnaces, Space or Wall Heaters

Basically, AMERIVENT is a pipe-within-a-pipe. Like the familiar thermos bottle, it utilizes reflective surfaces possessing low heat-radiating values and separated by an insulating air-space which prevents excessive heat loss—and draft loss—from the vent. AMERIVENT'S thin aluminum inner pipe is designed to heat immediately to a fully-primed, fully-operating temperature, thus preventing condensation and dangerous spillage of waste gases into the structure through the draft hood. The heavy-gage galvanized outer pipe stays cool to protect against fire hazards and heat-damage and to permit low-clearance installation.

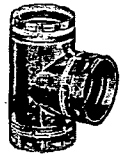
ROUND AMERIVENT



Round Amerivent



Round Americap



Tee



90° Elbow



45° Adjustable Elbow



Adjustable "L" Offset

PIPE SIZE:	AVAILABLE LENGTHS: 5'	3'	24"	18"	12"	6"
Style "E" 3", 4", 5" & 6"						
7" & 8"						
Style "R" 10" & 12" & Larger						

Shaded area indicates available sizes

Two types of Round Amerivent—Style "E" and Style "R"—are available, depending on the size desired. Both types carry full *U-L* approval, both employ a high-purity aluminum inner-pipe enclosed in a galvanized steel outer-pipe and both require only 1 in. minimum clearance from combustibles when installed. In addition, both pipes feature the improved Amerivent SNAP-LOCK joint. To assemble, simply press mating ends together and snap—a secure gas-tight joint is formed. A complete line of fittings is available.

OVAL AMERIVENT



Oval Amerivent



Oval Americap



Tee



45° Standard Elbow



45° Flat Elbow



Oval-to-Round and Round-to-Oval Adaptor

Size	AVAILABLE LENGTHS: 5'	4'	3'	18"	12"	6"
4"						
5"						

Shaded area indicates available sizes

Oval Amerivent is furnished in one model—Style "O"—in 4 in. and 5 in. sizes. It carries full *U-L* approval as a Type B gas vent requiring only 1 in. minimum clearance from combustibles. Style "O" Amerivent is also *UL*-approved as a Type B-W vent for use only with recessed (wall) gas heaters. In this application, 4 in. oval Amerivent can be installed in a 2 in. x 4 in. stud wall if used in conjunction with ceiling plate spacers and hold-down plate. A complete line of fittings is available, including adaptors for converting to or from Round Amerivent.

METALBESTOS DIVISION
WILLIAM WALLACE COMPANY · BELMONT, CALIF.

Representatives and Distributors in All Major Cities

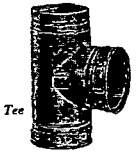
METALBESTOS DOUBLE-WALL GAS VENT PIPE AND FITTINGS



In Type B gas vent installations, the double-wall insulated design of Metalbestos gas vent pipe prevents condensation within the vent and spillage of combustion products at the appliance's draft hood. Rugged galvanized steel outer pipe is durable and strong. Inner aluminum pipe is corrosion-resistant, heats up quickly to create a strong venting draft. The built-in insulating air space prevents heat loss from flue gases as they pass through the vent to the out-of-doors.









RV METALBESTOS

Listed Sizes: 3 in., 4 in., 5 in., 6 in.

Standard Lengths: 6 in., 12 in., 18 in., 3 ft., 5 ft.

The convenient fittings of Metalbestos RV pipe permit complete venting for standard or special installations . . . eliminate job-site cutting and special hand-made parts. All RV round pipe features the quick-joining Rota-Lock Coupler.







QC METALBESTOS

Listed Sizes: 7 in. and 8 in.; Standard Lengths: 18 in., 3 ft., 5 ft.

Listed Sizes: 10 in. and 12 in.; Standard Lengths: 18 in. and 3 ft.

Larger-size QC Metalbestos is often required to furnish the common vent for two or more gas appliances. QC fittings are easily joined to QC standard lengths by means of the secure, die-formed QC coupler.







OVAL METALBESTOS

Listed Sizes: 4 in. (WV), 5 in. (Crimp)

Standard Lengths: 4-in. WV – 6 in., 12 in., 18 in., 3 ft., 4 ft., 5 ft.
5-in. Crimp – 3 ft. and 5 ft.

WV Metalbestos is designed for installation within 2 x 4 walls as Type B-W vent pipe for recessed wall heaters up to 65,000 Btu, and as Type B vent pipe for other installations. 5-in. Crimp-Joint Oval Metalbestos permits venting of larger-input appliances within 2 x 6 walls.

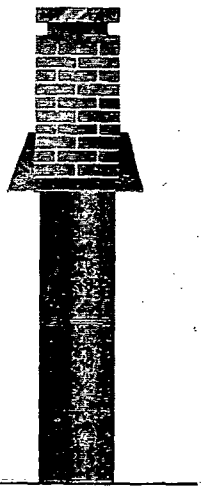
Metalbestos "Safety System" Gas Vent Tables Assure Correct Installation

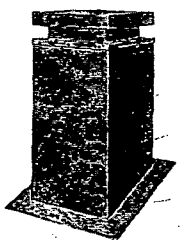
The Metalbestos "Safety System" Gas Vent Tables were developed by the Metalbestos Division of the William Wallace Company to give installers a simple, scientifically accurate guide to safe, correct gas venting. Comprehensive charts cover both individual and combined gas vents and provide the information that allows installers to solve the majority of their gas venting problems.

Easily read and understood, the illustrated "Safety System" Tables include appliance sizes and vent diameters most frequently encountered in gas vent installation. *Copies of the Metalbestos "Safety System" Gas Vent Tables may be obtained by writing to Dept. 9, William Wallace Co., Belmont, Calif.*

METALBESTOS DIVISION
WILLIAM WALLACE COMPANY · BELMONT, CALIF.

Representatives and Distributors in All Major Cities





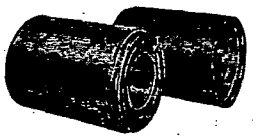
DELUXE AND STANDARD CHIMNEY HOUSINGS (16" x 16")

The four panels of the Standard Chimney Housing are simply assembled by means of slip-lock corners. Galvanized steel panels are easily painted in the same manner as galvanized gutter. Prepainted De Luxe Housing (shown in complete installation above) has the same construction as the Standard Housing and is available in attractive simulated red and buff brick. Heights: 3', 4', 5'.

THE METALBESTOS ALL-FUEL CHIMNEY FOR —

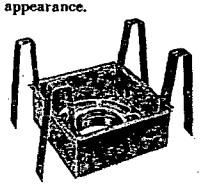
- Appliances Burning Oil . . . Gas . . . Coal . . . Wood
- Domestic Incinerators

The *light-weight, all-fuel* Metalbestos Chimney installs quickly and easily . . . reduces job-site construction time through simple, one-man installation. High-temperature-resistant mineral fiber and double-wall construction offer maximum performance and safety with any fuel. Exclusive Twist-Lock Coupling securely joins sections with a simple ⅛ turn . . . without cement, mastic, or screws. De Luxe Chimney Housing (shown at left) is one of four Metalbestos Chimney Tops designed to satisfy all requirements of cost or appearance.



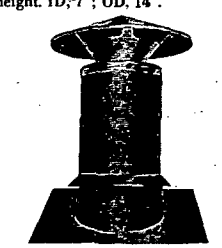
CHIMNEY PIPE

Between an outer pipe of galvanized steel and an inner pipe of stainless steel, mineral fiber is packed to assure safe, light-weight insulation. Combinations of easy-to-handle 18-in. and 30-in. lengths provide required Chimney height. ID, 7"; OD, 14".



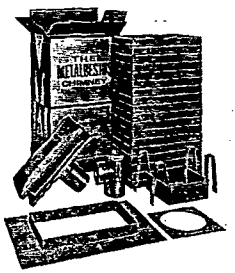
CHIMNEY SUPPORT ASSEMBLY

The sturdy Metalbestos Chimney Support Assembly when suspended from structural joists can easily support any Chimney height up to 53 feet.



ROUND CHIMNEY TOP ASSEMBLY

Contemporary in appearance, the Round Chimney Top handsomely complements modern building design. Complete assembly includes Chimney Top, Round Storm Collar, and Round Adjustable Roof Flashing. Tall Cone Roof Flashing is available for flat-roof installations.



CUSTOM CHIMNEY HOUSING

Custom Chimney Housing is pre-treated to permit direct paint application. "Shadow-Line" ridging adds extra reinforcement . . . simulates pattern of brick when painted. Custom Installation Package includes: Top, Housing, Chimney Flashing, Firestop Spacer, Support Assembly, Starter Tee.



This Metalbestos Chimney Catalog-Brochure gives you complete information on the all-fuel Metalbestos Chimney. Write to Dept. 9, William Wallace Co., Belmont, Calif.

DURA-VENT CORPORATION

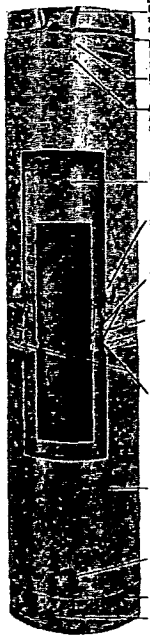
AFFILIATE OF PEERLESS MANUFACTURING DIV. OF DOVER CORP.

FACTORY & WESTERN SALES OFFICE
2525 El Camino Real
Redwood City, California

FACTORY & NATIONAL SALES OFFICE
1400 W. Ormsby Ave.
Louisville 1, Kentucky

DURA-VENT DOUBLE-WALL GAS VENT PIPE AND FITTINGS ASSURES SAFE GAS VENTING

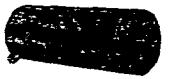
Dura-Vent Type B. Gas Vent Systems are made of 28-gage galvanized steel outer pipe and .016-inch aluminum inner pipe. The galvanized pipe complies with the U.L. requirements for fire stopping. The inner aluminum pipe is corrosion resistant to all flue gases, and it heats up quickly and maintains the proper temperature for a strong venting draft. The Dura-Vent design of continuous air space on all sizes of pipe and fittings keeps the outer pipe cooler than most other types of gas vent, and is preferred by most architects and specifying engineers.





ROUND PIPE

The 7-inch and 8-inch size pipe is made in the following lengths; 18-inch, 3-foot and 5-foot. The 10-inch and 12-inch size pipe is made in 18-inch and 3-foot lengths.
Dura-Vent round pipe and fittings are made in sizes from 3-inches to 12-inches in inside diameter. Pipe and fittings larger than 12-inches can be furnished on special order. The most commonly used sizes—3-inches through 6-inches—use the simple push-snap-twist lock. Pipe of larger sizes are locked together with screws.



ADJUSTABLE LENGTH

Adjustable lengths are 12-inches long and available for 3-inch size through 8-inch size round pipe.



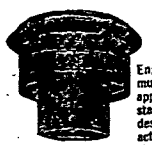
INCREASER

Increasers are used where two or more appliances are connected to the same vent. See Dura-Vent catalog for complete listing.



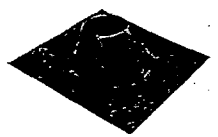
DRAFT-HOOD CONNECTOR

Draft hood appliance connectors are available in all sizes.



APPROVED DURA-VENT TOP

Engineered for maximum efficiency, smart appearance and easy installation. Scientifically designed to help draft action. Available in all sizes.



ADJUSTABLE ROOF FLASHING

Available in sizes 3-inch through 12-inch.



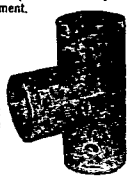
90° ADJUSTABLE ELBOW

Available in sizes 3-inch through 8-inch. A special feature of Dura-Vent is continuous air space the maintenance of in all positions of adjustment.



45° ADJUSTABLE ELBOW

Available in sizes 3-inch through 12-inch. A special feature of Dura-Vent is maintenance of continuous air space in all positions of adjustment.



TEE

Available in sizes 3-inch through 12-inch, plus a number of special reducing tees. Consult catalog for complete details.

OVAL VENT PIPE AND FITTINGS



STRAIGHT LENGTH

Available in 4-inch and 5-inch sizes. Consult catalog for details. Adjustable lengths are also available in 4-inch and 5-inch sizes.
Dura-Vent oval pipe and fittings are furnished in the 4-inch and 5-inch sizes. All oval pipe and fittings use the simple push-snap lock, which provides a rigid joint that can be easily taken apart if necessary.
The 4-inch and 5-inch size given for oval pipe refers to no measurable dimension, but signifies that the pipe is equal in venting capacity to 4-inch and 5-inch round pipe respectively.



45° ELBOW STANDARD

Standard 45° elbow illustrated. Available in 4-inch and 5-inch sizes. 45° flat elbow is also available in 4-inch and 5-inch sizes.



TEE

The oval tee available in 4-inch and 5-inch sizes.



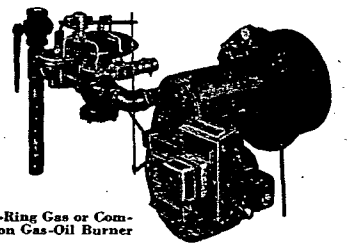
ADAPTER, OVAL TO ROUND

Oval to round adapter illustrated. Round to oval adapter also available. These adapters available in 4-inch and 5-inch sizes.

Gordon & Piatt, Inc. P.O. Box 914 Winfield, Kansas

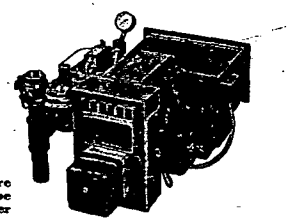
Combustion Engineering

TURBO-RING FORCED DRAFT GAS OR GAS-OIL BURNER provides all air for combustion under pressure adequate to fire Multiple Pass Scotch Marine Type Boilers without Stack or *ID* Fan. Complete package, with Combustion Safeguard, sequencing relays, indicating lights and switches prewired and tested. Natural Gas, No. 2 Oil, or Combination Gas-Oil in sizes to 200 hp. *See Cat. E-50.*



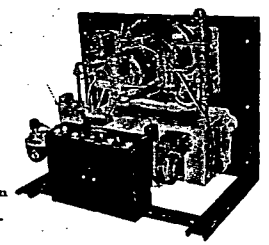
Turbo-Ring Gas or Combination Gas-Oil Burner

TURBO-FIRE POWER TYPE GAS BURNER refractory nozzles and distinctive air distribution give desired flame characteristics. Factory assembled with control panel prewired and tested. Sizes listed for 500,000 to 28,000,000 Btu/hr. *See Cat. C-10.*



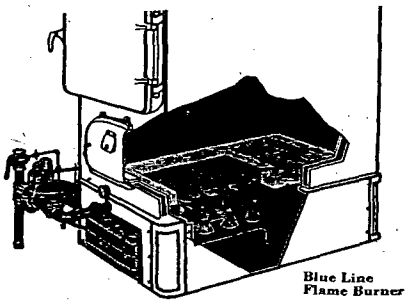
Turbo-Fire Power Type Gas Burner

COMBINATION GAS AND ROTARY CUP-OIL BURNER for larger installations requiring high performance operation on both gas and either light or heavy oil. Turbo-Fire Power Type Gas Burner and Rotary Cup-Oil Burner. Factory assembled, prewired and interlocked. *UL* listed to 30,000,000 Btu/hr. *See Cat. E-FGO-10.*



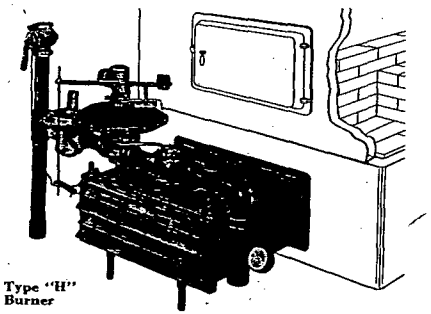
Combination Gas and Rotary Cup-Oil Burner

BLUE LINE FLAME BURNER atmospheric upshot type to convert large size heating plants and power boilers. Cast-iron burner heads with deep-slotted, machined ports suitable for natural gas or LP Gas under wide range of pressures. Welded manifolds to fit size and shape of fire box. On-Off, Throttling or Step-firing available. PBL Package Models to 960,000 in *Cat. A-11.* Larger sizes *Cat. A-10.*



Blue Line Flame Burner

TYPE "H" BURNER atmospheric inshot type for Scotch Marine Boilers and other applications where horizontal flame is indicated. Consists of removable cast-iron venturis with stainless steel flame retention nozzels. Natural or L.P. Gas. 33 sizes listed in *Cat. H-1A.* Others available on application.



Type "H" Burner

GORDON & PIATT are Combustion Engineers and Manufacturers, specializing in the design and manufacture of a complete line of Gas and Combination Gas-Oil Burners.

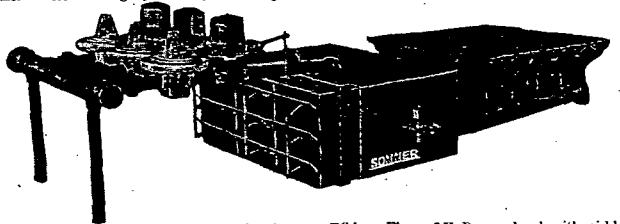
In addition to the burners illustrated, they manufacture Domestic Conversion Burners (7 Models listed by *A.G.A.* and *C.G.A.*). Control Panels, as cataloged, or custom-built to meet customers' requirements. Write for complete information.

The Sonner Burner Company

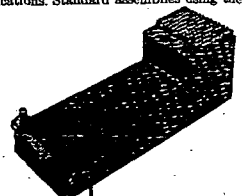
DESIGNERS AND MANUFACTURERS OF GAS CONVERSION BURNERS

OFFICES AND FACTORIES: WINFIELD KANSAS

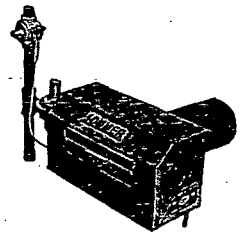
Sonner Gas Conversion Burners are a product of nearly a half century of specialization in design and manufacture. Many of the Sonner Conversion Units feature the Blue Ribbon Flame LV Burner Head with Airblend Design. Each ribbon of flame receives all the oxygen it needs to support complete effective combustion—from both sides along its full length—because the extraordinary Sonner Airblend Design provides just the right amount of secondary air in the right places. *Catalog 58-L-8* gives complete description of this efficient blue ribbon of flame. A wide range of designs provides a Sonner Burner for domestic, commercial, or industrial use. Sonner engineering service is always available for consultation; write for catalog containing complete line.



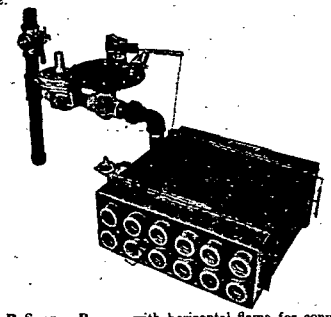
Type LVS Step-Fire Burner assemblies for large heating plants and power boilers where load varies to such extremes as to make this operation desirable. Three step control is the usual arrangement although it may be altered to meet engineers' specifications. Standard assemblies using the Sonner Blue Ribbon Flame LV Burner head with airblend design which provides just the right amount of secondary air to support complete and effective combustion, capacities for 1,440,000 to 7,200,000 Btu/hr are described in *Catalog 57-LVS-1*. Larger capacities available; consult Sonner engineering service.



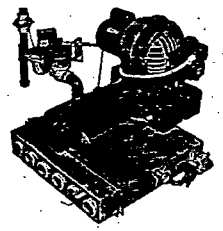
Type L-1 Heat Machine is designed specifically for the conversion of residential heating plants, either round warm air furnace or a rectangular boiler, to gas. Completely automatic burner, using the Blue Ribbon Flame LV Burner head with airblend design, is *A.G.A.* listed. Available in two sizes with capacities ranging from 75,000 to 400,000 Btu/hr. *See Catalog 58-L-8.*



Type WX-55 Sonner Burner specifically designed for replacing small gun type oil burners in domestic furnaces and boilers. *A.G.A.* listed with inputs ranging from 75,000 to 200,000 Btu/hr. *See Catalog WX-55-1.*



Type R Sonner Burner with horizontal flame for conversion of scotch marine boilers and similar equipment to gas and stand-by fuel applications. A multiple-venturi burner in factory built fully assembled form. Standard assemblies from 240,000 to 7,200,000 Btu/hr. *See Catalog 58-R-1*. Larger capacities available; consult Sonner engineering service.



Gas-Oil Unit for commercial and industrial boilers, two fuel burning units, each designed for one fuel only—natural gas and light fuel oils. For capacities from 280,000 to 2,800,000 Btu/hr input; *See Catalog 58-CU-1*. Larger capacities available; consult Sonner Engineering Service.

THE CARLIN COMPANY

912 SILAS DEANE HIGHWAY · WETHERSFIELD 9, CONN ·

MANUFACTURERS OF OIL BURNERS AND OIL-FIRED HOT WATER HEATERS

Pressure-atomizing, fully automatic, gun-type Oil Burners for residential and commercial installations.

The Carlin Company manufactures a complete line of quality oil burners in capacities from 0.65 to 20.00 gph for use of No. 2 grade fuel oil. Pedestal and flange mounted models are offered. Suspension mounting to popular makes of boilers and furnaces is made easy with a wide selection of flanges and mounting plates especially developed by U.S.-Carlin. Advanced design features include the "Shell Head," a significant improvement in oil burning which develops higher combustion temperatures, makes heat-wasting soot deposits negligible, conserves fuel. With either distillate or catalytic fuels, operation is quiet, clean, economical. Generous use of heat and corrosion resistant alloys guards against deterioration—makes cleaning easier.

U.S.-Carlin Oil Burners are available with Electronic Controls which comply fully with Underwriters' Laboratories latest safety switch timing requirements. Many types are factory mounted and wired to insure proper hook-up and save on-the-job wiring expense.

For complete technical literature including consulting engineers' manual, specification sheets, electronic control data, combustion chamber dimensions, etc., address The Carlin Co., 912 Silas Deane Highway, Wethersfield 9, Conn.

CONDENSED SPECIFICATIONS

MOTOR—Long-hour duty with safety overload control. Complies with *NEMA* mounting specifications. ⅙ hp. (Models 400, 150F, 400S, 150SF-2, 500S-35); ⅓ hp. (Models 650, 700S-35, 800S-4); ¼ hp. (Model 1200); ½ hp. (Models 2000S-5, 2000).

TRANSFORMER—Shielded to block radio and TV interference. 10,000 volt (12,000 volt on Models 2000S-5, 1200, 2000).

AIR TUBE—Steel with heat resistant air cone.

FAN—Precision balanced. Mounted directly on motor shaft.

NOZZLE—Stainless steel alloy. Hollow cone spray (Hollow cone or solid spray on Models 1200, 2000). Dual nozzles on Models 500S-35, 650, 700S-35, 800S-4, 2000S-5, 1200, 2000.

FUEL UNIT—Single-stage standard on Models 400, 150F, 400S, 150SF-2, 500S-35, 650. Two-stage on Models 700S-35, 800S-4, 2000S-5, 1200, 2000. Built-in strainer and pressure regulating valve. Capacity of strainer equals or exceeds maximum firing rate of burner.

SOLENOID VALVE—Combination delayed-opening and instantaneous cut-off oil valve standard on all "S" models, also 1200, 2000.

COMBUSTION HEAD—Models 150F, 400, and 650 have standard head with diffuser vanes formed at a fixed angle to establish air pattern. Model 150F (0.65 to 1.35 gph capacity) and Model 400 (0.65 to 1.00 gph capacity) have a series of adjustable openings in center of diffuser for regulating air delivery to shape the fire.
All "S" Models have "Shell Head" and finned collar for efficient mixture of air with oil. Models 1200, 2000 have High Temperature Combustion Head with finned collar for greater firing rate (Model 2000) and narrower fire.

ELECTRODES—Nichrome steel with full-glazed porcelain insulators counterbored for utmost safety. Flared, "Wide-arc" design permits heavier spark and assures positive protection during low voltage periods.

BUS BARS—Rigid heavy-gage brass (Models 400, 400S, 500S-35, 700S-35, 800S-4); phosphor bronze (Models 150F and 150SF-2); high tension leads (Models 650, 2000S-5, 1200, 2000).

HOUSING—Cast iron. Designed to assure efficient air delivery and quiet operation. Machined to rigid standards.

COUPLING—Universal type with neoprene center piece. (Flexible type on Models 2000S-5, 1200, 2000).

CONTROLS—Thermostat, Limit Control, and Stack Relay. Electronic Control set in place of Stack Relay on Models 800S-4, 2000S-5, 1200 and 2000.

ELECTRICAL CHARACTERISTICS—115V-60 cycle. 1725 rpm. All models except 150F and 150SF-2 available with 230V-60 cycle on special order. **Also a complete line of 110 and 220V-50 cycle models with capacities from 0.50 to 15.00 gph.**



MODEL 400
0.65 to 4.00 gph
Also available for flange mounting



MODEL 400S
0.75 to 3.00 gph
Also available for flange mounting



MODEL 150F
0.65 to 2.00 gph
Also available for *pedestal mounting*



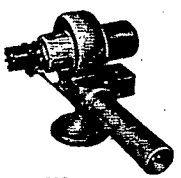
MODEL 150SF-2
0.65 to 1.35 gph
Also available for pedestal mounting



MODEL 500S-35
3.00 to 4.50 gph



MODEL 700S-35
3.50 to 7.00 gph



MODEL 650
3.50 to 6.50 gph



MODEL 800S-4
4.00 to 8.00 gph



MODEL 1200
6.00 to 12.00 gph



MODEL 2000S-5, 7.00 to 12.00 gph
MODEL 2000, 10.00 to 20.00 gph

U.S.-Carlin Oil Burners are listed by *Underwriters' Laboratories, Inc.* and are approved under *U. S. Government Commercial Standard C372-58*. Grade of fuel oil recommended is No. 2 *U. S. Government Commercial Standard CS18*.

Enterprise Engine & Machinery Co.

A SUBSIDIARY OF GENERAL METALS CORPORATION

18th & Florida Streets, San Francisco 10, California

Distributors in Principal Cities

ENTERPRISE BELT DRIVE HORIZONTAL ROTARY BURNERS

TYPE A OIL BURNER

A fully automatic burner equipped with either gas-electric or oil-electric ignition system. Firing rate controlled by either modulating or two position control meter, with low fire start.

TYPE I OIL BURNER

Manual—To be started, stopped and regulated by an attendant.

Semi-Automatic—To be started by an attendant. Firing rate controlled by either modulating or two position control motor.

COMBINATION GAS-OIL BURNERS

Three types of burners are available depending upon the method of starting the burners:

Type AY is furnished with gas-electric ignition and programming type safety combustion controls to provide full automatic operation. Requires manual re-set of control only after safety shut down.

Type AY-1 is furnished with manually operated gas-electric ignition and combustion safety control. Requires manual re-starting after shut down due to any cause.

Type AY-2 requires manual starting and torch ignition of main flame on oil or pilot flame on gas. Utilizes combustion safety control which requires manual starting after shut down due to any cause.

APPROVALS

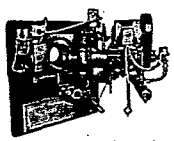
All Enterprise Burners are listed by *Underwriters' Laboratories, Inc.* and approved by *Canadian Standards Association* and many state and civic safety authorities.

FUEL

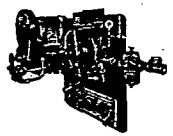
Burners are listed to burn oil not heavier than Commercial Standard Grade No. 5 without electric oil preheater and C.S.G. No. 6 when equipped with electric oil preheater; also natural, manufactured, or liquefied petroleum gases. Minimum required gas pressures of 2 to 5 in. water column, depending upon burner size.

DESCRIPTION

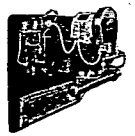
Burner utilizes standard electric motors with single- or multiple V-belt drive to provide full burner rating at any frequency of electric power. Available for either single or polyphase power, voltages of 110, 208, 220, 440 or 550 at 50 or 60 cycles. All listed types of electronic safety controls available. Burner may be furnished with integral fuel oil pump, with integral metering pump, or non-pump with separate pump set.



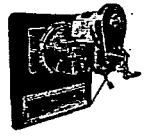
Type AY—Combination Gas-Oil Burner



Type A—Automatic Burner without Electric Oil Heater



Type A—Automatic Burner with Electric Oil Heater



Type I—Simple Manual Type Burner

Ratings are based upon the combustion of fuel having heat contents of 150,000 Btu per gal for oil and 1000 Btu per cu ft for gas, both at 75 percent efficiency. Outputs are based on the requirements of 970 Btu per lb of steam, 33.500 Btu per boiler hp, 240 Btu per sq ft of steam radiation and 150 Btu per sq ft of hot water radiation. When using lighter grades of fuel oil, burners will handle an increased oil input sufficient to maintain the same total heat output as for heavy oil.

BURNER SIZES AND BASIC RATINGS

Burner Size	Motor hp	INPUT Oil-Gal/Hr. Min.	Oil-Gal/Hr. Max.	Gas CF/Hr	OUTPUT Thousand Btu per Hr	Steam Lb/Hr	Steam KG/Hr	Boiler hp	Radiation-Sq Ft Steam	Radiation-Sq Ft Hot Water
AA3	½	1	4	600	450	464	210	12	1880	3000
A3	½	2	7	1050	787	812	368	22	3220	5250
C3	½	4	15	2250	1687	1740	790	50	7050	11250
E3	½	6	20	3000	2250	2320	1053	67	9400	15000
F3	¾	7	30	4500	3375	3480	1578	100	14100	22500
G3	1	9	35	5250	3937	4060	1840	118	16400	26300
H3	1	12	50	7500	5625	5800	2630	168	23400	37500
H3	1½	15	60	9000	6750	6960	3156	201	28200	45000
J3	2	19	75	11250	8437	8700	3945	250	35100	56250
K3	2	25	100	15000	11250	11600	5260	336	46900	75000
L3	3	30	135	20250	15187	15660	7110	453	63400	101000
M3	5	45	200	30000	22500	23200	10526	673	94000	150000

INDUSTRIAL COMBUSTION INC.

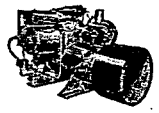
EXECUTIVE OFFICES: 4507 N. OAKLAND AVENUE, MILWAUKEE 11, WIS. • FACTORY: MONROE, WIS.

COMMERCIAL • INDUSTRIAL BURNERS ENGINEERED TO FIT YOUR EXACT NEEDS

HEV-E-OIL BURNERS
for No. 4, No. 5 and PS 300 OILS

COMPLETE SIZE RANGE
5 gph or 720,000 Btu GAS INPUT to 150 gph or 21,000,000 Btu GAS INPUT.
FOR BOILERS UP TO 480 hp

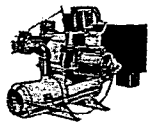
HEV-E-DUTY POWER GAS BURNERS
for NATURAL GAS or LPG . . . COMBINATION LIGHT or HEAVY OIL MODELS



HEV-E-OIL BURNERS
for No. 4, No. 5, and PS 300 OILS.
Larger models use No. 6 oil.

$	Burner Model	Capacity GPH Max.	Capacity GPH Min.	Boiler Max. HP	Maximum Net Standing Radiation Steam	Water
P	*AL2C — AM2CH — AM2CF	5	1	16	1,750	2,800
	AM3CH — AM3CF	11	1	35	3,800	6,000
E	AM4CH — AM4CF	17.5	3	56	6,100	9,800
	AM5CH — AM5CF	25	5	80	8,750	14,000
C	AM6CH — AM6CF	40	6	128	14,000	22,400
	AM7CH — AM7CF	60	8	192	21,000	33,600
$	*AM8-E	100	12	320	35,000	56,000
	*AM9-E	150	15	480	52,500	84,000

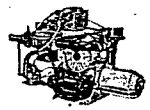
*AL2C uses Minneapolis-Honeywell RA117A stack control. Choice of M-H RA890C or Fireye FJ2 electronic controls on Models 2 through 7.
*Models 8 and 9 burn No. 6 oil, are factory-equipped with heaters, have FJ2 controls.



HEV-E-DUTY POWER GAS BURNERS
for NATURAL GAS or LPG

$	Burner Model	Gas Input Max. BTU	Boiler Max. HP	Maximum Net Standing Radiation Steam	Water
P	*G2	720,000	16	1,750	2,800
	*G3	1,580,000	35	3,800	6,000
E	*G4	2,500,000	56	6,100	9,800
	G5	3,600,000	80	8,750	14,000
C	G6	5,750,000	128	14,000	22,400
	G7	8,600,000	192	21,000	33,600
$	G8	14,300,000	320	35,000	56,000
	G9	21,000,000	480	52,500	84,000

*Choice of M-H RA890C or Fireye FJ2 controls on Models 2, 3 and 4. FJ2 controls standard on Models 5 through 9.



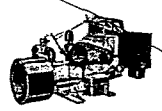
COMBINATION BURNERS
FOR LIGHT OIL and GAS

$	Burner Model	Gas Input Max. BTU	Oil Max. GPH	Boiler Max. HP	Maximum Net Standing Radiation Steam	Water
P	*GL2	720,000	5	16	1,750	2,800
	*GL3	1,580,000	11	35	3,800	6,000
E	*GL4	2,500,000	17.5	56	6,100	9,800
	GL5	3,600,000	25	80	8,750	14,000
C	GL6	5,750,000	40	128	14,000	22,400
	GL7	8,600,000	60	192	21,000	33,600
$	GL8	14,300,000	100	320	35,000	56,000
	GL9	21,000,000	150	480	52,500	84,000

*Choice of M-H RA890C or Fireye FJ2 controls on Models 2, 3 and 4. FJ2 controls standard on Models 5 through 9. Simple fuel change-over.

Industrial Combustion burners operate on the principle of low-pressure air-atomization and furnish all the air necessary for complete combustion. Uniform, high efficiency is obtained at all times regardless of weather.

Built to utilize the fuel lowest in cost and most readily available, these commercial-industrial burners are factory guaranteed . . . sold and serviced throughout the world. Fully approved, they meet all codes and *Underwriters Laboratories* standards.



COMBINATION BURNERS
for HEAVY OIL (No. 4, No. 5, PS 300) and GAS

$	Burner Model	Gas Input Max. BTU	Oil Max. GPH	Boiler Max. HP	Maximum Net Standing Radiation Steam	Water
P	AMG3	1,580,000	11	35	3,800	6,000
	AMG4	2,500,000	17.5	56	6,100	9,800
E	AMG5	3,600,000	25	80	8,750	14,000
	AMG6	5,750,000	40	128	14,000	22,400
C	AMG7	8,600,000	60	192	21,000	33,600
	*AMG8	14,300,000	100	320	35,000	56,000
$	*AMG9	21,000,000	150	480	52,500	84,000

Fireye FJ2 electronic controls standard on all burners. *Models 8 and 9 burn No. 6 oil. Simple fuel change-over.

- **COMPLETELY AUTOMATIC**
- **SAFE . . . LATEST ELECTRONIC CONTROLS**
- **AUTOMATIC ELECTRIC IGNITION**
- **LOW FIRE START**
- **EASILY MAINTAINED**
- **COMPLETE . . . READY FOR INSTALLATION**
- **FACTORY FIRE TESTED**
- **POSITIVE FUEL SAVINGS**
- **SUPER-SIMPLE INSTALLATION**

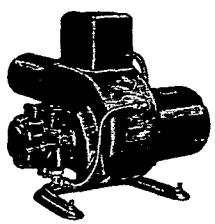
Write for free copy of
"ARCHITECTS and ENGINEERS HANDBOOK"
Descriptive BULLETINS 1 and 2 mailed on request.

S. T. JOHNSON CO.

940 Arlington Ave., Oakland 8, Calif.

Johnson Oil Burners

Church Road, Bridgeport, Pennsylvania



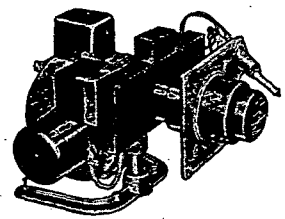
BANKHEAT BURNERS

Pressure atomizing type. Fully automatic electronic controls. Built with the heat saving air-shut-off feature which prevents cold air from passing through the burner, firebox and flue when burner is not in operation. *Pump-driven oil is forced* through orifice, pressure-atomized and mixed with whirling jet of fan-driven air, to burn with a soft clean flame while suspended in air.

Notable for exceptional efficiency, fuel-economy and trouble free performance. Adaptable to most domestic and commercial heating needs. Particularly suited to use for "conversions." Easy to install and service.

Water Heaters, Boiler-Burner Units, Hot Air Furnaces

Johnson Water Heaters, Boiler-Burner Units and Hot Air Furnaces powered by Bankheat Burners are available in a wide range of sizes and types. Descriptive literature on any item may be had on request.



BANKHEAT Combination OIL-and-GAS BURNERS

Will automatically burn either oil or gas with high efficiency. An ideal installation for areas where low-cost gas is available only on a part time basis. You can change from one fuel to the other with a flip of the changeover switch (either manual or, at small extra cost, automatically controlled by outside temperatures).

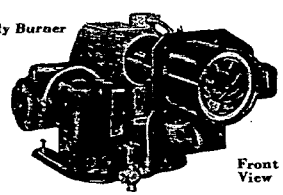
Sizes and Capacities. Bankheat Burners

SIZE	MAX FIRING RATE		MAX BOILER OUTPUT		
	Oil gph	Gas CFH 1000 Btu	Steam Sq Ft	Water Sq Ft	HP
0	2¼	315	900	1450	6½
0A	3	420	1300	2100	9
1A	5½	770	2300	3650	16
2	7½	1050	3100	4950	22
2A	10	1400	4150	6650	30
3	15	2100	6200	9900	45
3A	18	2520	7500	12000	54

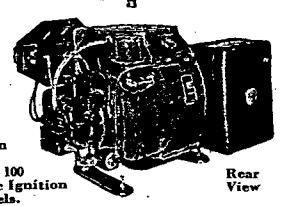
FORCED DRAFT BANKHEAT BURNERS

Fully automatic, for Firing on OIL only, GAS only, or COMBINATION OIL-and-GAS

Gas Only Burner Size 100



Front View



Combination Oil & Gas Burner Size 100 Gas-Electric Ignition on Both Fuels.

Rear View

Built to deliver all the air needed for combustion without the aid of an induced-draft fan or tall chimney. A special backward-curved fan rotating at twice ordinary speed drives this air with a minimum of pulsation in the air stream delivered to the combustion chamber. With Bankheat Forced Draft Burners there are few pulsation problems on "hard to fire" installations involving either positive or negative firebox pressures.

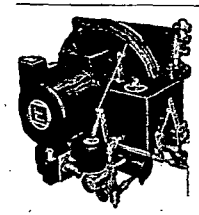
Made in 5 basic sizes: 30, 40, 60, 80 and 100 hp. These ratings are based on positive firebox pressures up to 0.75 in. wc. These units are easily wired, installed and serviced. Flange mounts are furnished. According to Type and Size, each burner is equipped with Straight-Electric or Gas-Electric ignition, with Electronic Controls to suit.

S. T. JOHNSON CO.

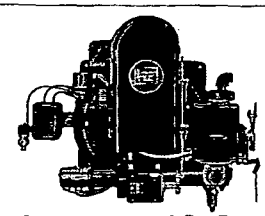
940 Arlington Ave., Oakland 8, Calif.

Johnson Oil Burners

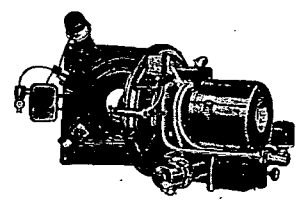
Church Road, Bridgeport, Pennsylvania



Model 53 Burner



Model 53 Burner with Belt Drive



Model 53 Combination Oil-and-Gas Burner

JOHNSON MODEL 53 METERING PUMP BURNERS

Fast Positive Starting. Smooth Operation. Greater Fuel Economy.

These rotary-type burners were *designed to produce sure*, automatic "starts" even when the oil in storage tank and lines is cold and hard to pump. This important performance feature was accomplished by using a positive-displacement Metering Pump, . . . a 3-Way Magnetic Oil Valve, . . . and a high *efficiency Suction Pump*, which enable Model 53 Burners to maintain, automatically, a fixed air-fuel-ratio regardless of variations in oil temperature and viscosity.

They are easy to set and to control. Firing Rate can be determined by reading the setting of the Metering Pump Quadrant. All working parts are quickly accessible for servicing or repair. The fuel economy and reliability of these burners has been thoroughly established in every type of heavy duty service throughout America and numerous foreign countries. Model 53 Burners are available with either conventional Direct Drive or with Belt Drive for use in installations using 50 cycle current to maintain 60 cycle speed, . . . and on installations where additional capacity is required.

They are available for firing on Oil Only, Gas Only, or as Combination Oil-and-Gas Burners. There are 9 sizes ranging from 28 to 560 hp and from 3980 to 78600 sq ft Steam Radiation.

SIZES AND CAPACITIES, MODEL 53

Burner Size*	Motor HP	Pre-Heater KW	Suction Pump Capacity GPH	METERING PUMP CAPACITY GPH		†BURNER RATING				
				‡Nominal Quad. Setting 1.0	‡Maximum Quad. Setting 1.25	Firing Rate GPH	Boiler HP	Pounds Steam Per Hr. at 212	Sq. Ft. Steam EDR	Sq. Ft. Water EDR	BTU Output (1000 Btu)
25	½	2	40	7.5	9.5	8.5	28	985	2980	6350	956
50	¾	2	40	15	18.75	17	57	1955	7850	17750	1912
75	¾	2	60	22.5	28.12	25	83	2900	11700	15750	2813
100	1	2	60	30	37.5	34	113	3930	15650	25500	3825
150	1½	2	125	45	56.25	50	167	5800	23400	37500	5625
200	2	5	125	60	75	67	225	7750	31400	50200	7538
300	3	5	190	90	112.5	100	333	11500	46800	75000	11250
400	4	5	190	120	150	133	450	15600	62350	101200	15188
500	7½	10	225	150	187.5	168	560	19400	78600	126000	18900

Capacities and ratings are based on 60 cycle current.

* Burner size designation is boiler horsepower developed at nominal firing rate obtained at metering pump setting of 1.0.

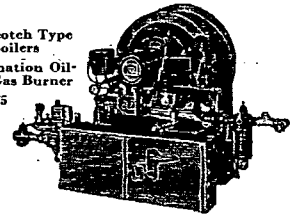
‡ *Maximum metering pump capacity setting 1.25 is maximum firing rate.*

† Burner rating is based on specific conditions of draft, combustion volume and combustion chamber design. Under favorable conditions, it may be permissible to use maximum firing rate. Under restricted conditions, it may be desirable to use reduced firing rates. Ratings based on 150,000 Btu/gal and 75 percent efficiency. For use with #2 to #6 C.S.G. Fuel Oil.

COMPACT FORCED DRAFT Package Unit BURNERS

For Scotch Type Boilers

Combination Oil-and-Gas Burner Size 75



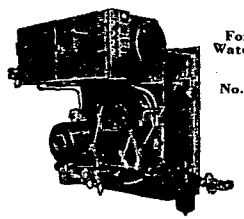
Built for firing on Oil Only, Gas Only or as Combination Oil-and-Gas Burners. Two types are available for use on Scotch Type or Firebox Type boilers. They are engineered to provide all the air needed for proper combustion regardless of stack conditions and firebox pressure variations.

For Firebox and Water Tube Boilers

Oil Only

No. 5 or No. 6 Oil

Size 150



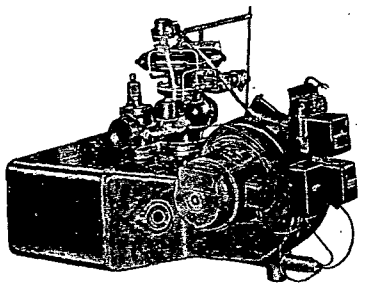
All these Forced Draft Package Units are factory-assembled, wired and tested, complete with all the controls mounted in dust type cabinet, ready for fast, inexpensive installation on any standard size and type of boiler. They are available in the sizes and capacities listed above for Johnson 53 Burners.

SHEETS GIVING CATALOG DATA AND SAMPLE SPECIFICATIONS ARE AVAILABLE ON REQUEST

THE METTLER CO., Inc.

1101 Buchanan St., Rockford, Ill.

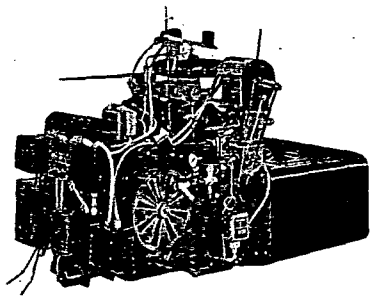
Division of Eclipse Fuel Engineering Company



Packaged burners for gas or combination gas-oil firing rated up to 800 bhp output have been designed and built by Mettler for more than 35 years.

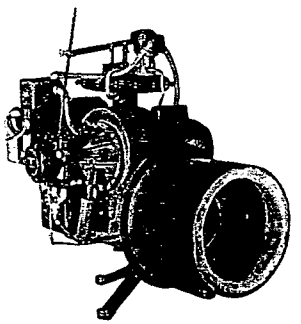
FAN-AIR GAS BURNERS

For forced, induced or natural draft operation in large and small power plants with capacities from 1,300,000 to 35,000,000 Btu input. Eight largest standard size burners are equipped with enclosed control cabinets. *Details in Bulletin G-200-B.*



FAN-AIR COMBINATION GAS-OIL BURNERS

With output ratings of 25 to 800 bhp, change fuel at the flick of a switch located either on burner or at remote station. Oil-gas internal assembly combines gas pilot, oil nozzles, spark rods, flame rod, photo-cell and stainless steel Turbo-Fins in a single compact unit for adjustment or cleaning. *Details in Bulletin GO-200-B.*



TURBO-FLAME BURNER

Is a single port nozzle mixing burner for gas, oil, or gas-oil combination firing. Designed for use on Scotch marine boilers where back pressure or positive pressure exists or is likely. Capacity ranges from 500,000 to 5,000,000 Btu/hr depending on back pressure. *Details in Bulletin GO-300-B.*

Other standard Mettler burner lines include the Series "C" entrained combustion gas burners and Mettler junior "Fan-Air" burners rated at 150,000 to 1,400,000 Btu input. Literature on both lines available.

RAY BURNER COMPANY

Since 1872

1301 San Jose Avenue; San Francisco 12, California

Atlantic Seaboard Div.; 629 Grove St., Jersey City 2, N. J. Distributors in All Principal Cities of the World



Scotch Marine application



Type for water tube boilers

RAY FORCED DRAFT PACKAGED BURNERS

A factory assembled unit readily applied to any type of boiler designed for forced draft operation. Fully automatic operation on oil, gas or combination gas-oil.

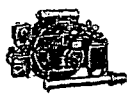
The combination gas-oil unit includes a rotary oil burner, integrated ring type gas burner, secondary air control with a forced draft fan, windbox, refractory support ring, completely wired control cabinet and pre-piping completely factory assembled for quick and easy installation to the boiler.

Oil components may be selected to burn any grade of oil with the secondary air control providing increased efficiency even with the difficult-to-burn catalytic residual #6 fuel oils. Sizes 1 to 10. From 40 to 750 developed output horsepower.

RAY COMBINATION OIL-GAS BURNERS

ARC-MODEL 550. Consists of a rotary type oil burner of Ray conventional design coupled with a gas housing for low pressure gas. The gas burner parts provide for the introduction of gas in a manner which fully utilizes the energy in the high pressure air supplied by the rotary burner fan. Gas and primary air are intimately combined, in two stages, before mixing with the secondary air which is introduced through a vaned annular opening concentric with the nozzle. Change-over requires less than one minute. Available to burn any grade of oil; 9 sizes with capacities to 31,500,000 Btu/hr. Gravity type to 48,000,000 Btu/hr.

TYPE PCP. Fully automatic combination gas-oil burner of the pressure atomising type. Available in three sizes with capacities from 560,000 to 4,200,000, Btu/hr. High speed 3450 rpm models available for forced draft operation. Also available for straight gas operation, as type PG.



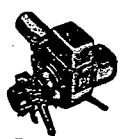
ARC-144 Model 550 for heavy oil and Low Pressure Gas



Type PCP for #2 oil or lighter and Low Pressure Gas

RAY DOMESTIC AND STEAM TURBINE BURNERS

RAY PRESSURE ATOMIZING OIL BURNERS. Fully automatic for #2 oil or lighter. 4 sizes with capacities from 1 to 30 gph. Pedestal or Flange mounting. All types have electric ignition interrupted or constant. AC or DC. Electronic controls are standard on all models above 7 gph capacity. High speed 3450 rpm models available for forced draft operation.



Fully Automatic, Type 1P, #2 or lighter oil



Steam Turbine Drive, Type TC, all grades oil. Tested and approved for U. S. Navy Service

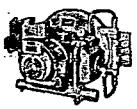
RAY INSHOT GAS BURNERS

Multi-jet atmospheric type gas burners with high flexibility in nozzle number, arrangement and capacity, permitting high input ratings in limited space. For commercial and industrial application with natural or induced draft. Available in 36 different jet arrangements with capacities from 99 to 7512 cfh.



Size 2, IC-29

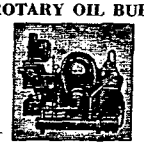
RAY HORIZONTAL ROTARY OIL BURNERS



Type AR-144 Size 10 for No. 6 oil, Available in 10 sizes from 2 to 210 gph. Fully automatic.



Fully Automatic Type AR-134, for No. 5 oil



Type BR-144, Belt Drive, Fully Automatic, Available in 6 sizes. Capacities from 2 to 110 gph.



Type ARIP, Fully Automatic for heavy oil where gas for ignition not available

Built in fully automatic, semi-automatic and manual types; in sizes from 7 to 1000 Boiler hp, 230 to 37,000 lbs/hr, to burn all grades of fuel oil.

Standard models include both direct and belt drives—the latter being recommended for use where other than 50 or 60 cycles AC, or only DC is available. Types for diesel electric or gas electric ignition; pump or gravity feeds. Direct drive types include a steam turbine driven model.

Most fully automatic types for heavy oil incorporate the Ray Dual Pump and Reservoir, with the Ray VISCOSITY Valve, a patented, exclusive feature which automatically meters the correct amount of fuel at all times, regardless of changes in viscosity of the oil due to temperature variations. All larger sizes employ dual ignition system, consisting of dual high-voltage transformers, dual electric ignitors and dual gas valves. Fully automatic burners include as standard equipment electronic pilot and flame failure control as required by code.

RAY BURNER HOURLY CAPACITY RATINGS

Burner Size	Motor H.P.	Oil Capacity U. S. Gallons		Equivalent Boiler HP		Equivalent Lbs. Steam Generated †		Heat Capacity Input Thou. Btu Oil or 1000 Btu Gas		Equivalent Sq. Ft Steam Radiation	
		Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
JP	¼	1	2	3	9	110	325	140	420	440	1350
1-P	¼	4	7	12	22	430	755	560	1000	1750	3060
2-P ††	½	7	18	23	55	755	1950	1000	2500	3060	7900
3-P	¾	14	30	45	95	1610	3340	1960	4200	6120	12140
0	¼*	2	5	7	16	230	580	300	750	938	2340
1	¼	4	11	13	35	460	1270	600	1650	1880	5180
2	¼**	6	15	17	50	580	1740	750	2250	2340	7020
3	½**	8	25	27	83	920	2900	1200	3750	3750	11720
5	1	10	35	33	116	1160	4050	1500	5250	4690	16400
6	1½	12	50	40	165	1400	5800	1800	7500	5620	23500
7	3	18	70	50	230	1740	8100	2250	10500	7030	32800
8	5	25	110	63	365	2900	12700	3750	16500	11700	51500
9	5	35	150	116	500	4050	17400	5250	22500	16400	78500
10	7½	50	210	155	700	5800	24400	7500	31500	23400	98300
12	7½	75	320	250	1000	8700	37000	11250	48000	35100	149200

NOTE: These ratings are predicated upon specific conditions of draft and furnace volume. It may be permissible, under desirable conditions, to operate at higher rates, or advisable under restricted conditions, to operate at reduced rates. Heating capacities are based upon 150,000 Btu per gal of oil for rotary burners and 140,000 Btu per gal of oil for pressure atomizing burners and upon an overall boiler efficiency of 75 percent.

† Steam capacity based on steam F and A 212 F. * ⅓ hp on belt driven models.

** ¾ hp on belt driven models. †† Motor hp may be increased for forced draft operation.

PETRO

3242 West 106th Street, Cleveland 11, Ohio

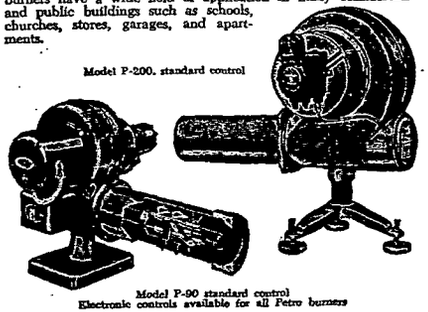
Quality Heating and Power Equipment since 1903

PETRO REG U S PAT OFF — INDUSTRIAL AND COMMERCIAL OIL AND GAS BURNING SYSTEMS FOR EVERY NEED

LARGE CAPACITY BURNERS, MODELS P-90, P-90E, P-200, P-200E:

In addition to domestic heating *in* large residences, these Petro burners have a wide field of application in many commercial and public buildings such *as* schools, churches, stores, garages, and apartments.

Model P-200, standard control



Model P-90 standard control
Electronic controls available for all Petro burners

CAPACITY OF PETRO PRESSURE ATOMIZING BURNERS

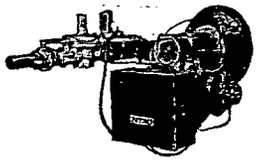
Burner Model	Nozzle Size Gals. per Hr.	Total Capacity* Steam Sq. Ft.	Total Capacity* Hot Water Sq. Ft.**	Approximate Shipping Weight
P-90	5 to 9	1750 to 3150	2800 to 5050	130
P-90E	5 to 9	1750 to 3150	2800 to 5050	130
P-200	9 to 20	3150 to 7000	5040 to 11200	180
P-200E	9 to 20	3150 to 7000	5040 to 11200	180
P-300***	20 to 30	7000 to 10500	11200 to 16800	260

*Total capacity equals standing radiation, plus piping loss, plus pickup, plus hot water.

**Rating based upon water at 170°F in radiators.

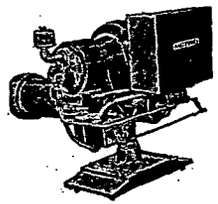
***Standard with electric controls giving two-second shut-down.

NEW COMMERCIAL FORCED DRAFT PACKAGE BURNER —FOR OIL, GAS OR DUAL-FUEL—P-240 GOSM



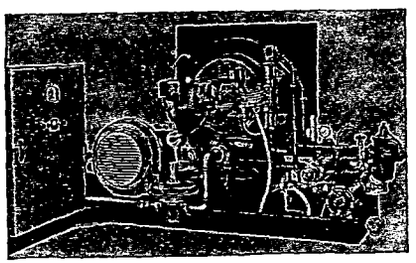
Here is a completely new commercial pressure burner which offers two distinct features: forced draft operation and freedom from pulsation. All necessary operating controls, including an electronic combustion control system, are integral components. The units are complete fuel burning systems—assembled, wired and tested at the factory. Air for combustion is supplied under pressure by a highly efficient blower. Burners are available in two types of construction: fan housing above or below the draft tube. Models with fan housing below the draft tube are particularly adaptable to Scotch-Marine type boilers, as no burner parts project above the draft tube to interfere with boiler cleanout doors. Installation is simple. Capacities range from 8 through 24 gph for oil and for gas from 1,200,000 through 3,600,000 Btu input.

"PACKAGED" COMMERCIAL OIL BURNERS—FOR FUEL OILS UP TO AND INCLUDING HEAVY NO. 5



Here at last is a reliable gun type low pressure burner that will fire No. 5 oils or any lighter grade, using the supercharger-type fuel atomizer. Atomization is so complete that electric ignition can be used in place of the gas flame required by other burner types. The ability to fire heavy or light oils, and to modulate over a wide range, makes this burner unique in its field. Capacities are from 3 to 25 gph.

PETRO COMPLETE "PACKAGED UNIT" Consisting of Burner (for oil or gas or both), Fuel System, Control Panel, and Preformed Refractory Combustion Throat.



This Petro package unit is much more than a conversion burner. It's a complete combustion system in which all elements are correctly balanced and integrated—a thoroughly engineered firing plant. Includes burner (for oil or gas or both), fuel system, forced draft air supply, control panel, and pre-formed refractory combustion throat. Installation requires little more than bolting the entire unit to the boiler-front and making service connections for power and fuel. This represents a substantial saving in installation time and cost—but even more important it means a factory tested and assembled unit for high operating efficiency and fuel savings.

Control panel is totally enclosed, with all instruments wired and tested at the factory.

The units are designed for easy application to all types of boilers including the Scotch Marine. They operate either under normal firebox draft conditions or under positive pressure in sealed firebox boilers which are designed for operation under furnace pressure. Oil capacity—25 to 145 gph (No. 6 or lighter) and equivalent ratings in gas.

PETRO

3242 West 106th Street, Cleveland 11, Ohio

Quality Heating and Power Equipment since 1903

PETRO REG U S PAT OFF — INDUSTRIAL AND COMMERCIAL OIL AND GAS BURNING SYSTEMS FOR AUTOMATIC, SEMI-AUTOMATIC OR MANUAL OPERATION

FOR UNHEATED COMMERCIAL OILS

Model W-A: Automatic ignition with synchronized control of oil and air.

FOR HEATED OILS: HEAVY NO. 5 (MODEL W-A-E); BUNKER "C" NO. 6 (MODEL W-AH)

Models W-A-E; W-AH: Automatic ignition with synchronized control of oil and air, and automatic control of oil heaters.

ALL PETRO MODEL W ROTARY-CUP OIL BURNERS:

A Petro Model W consists of a self-contained assembly of fan, pump, rotary cup atomizer and all oil and air adjustment apparatus. It is either direct-driven by means of a built-in motor, or belt-driven from motor mounted on burner housing. Interlocking oil and air control mechanism permits operation at any point within the burner's minimum and maximum range.

CAPACITIES OF PETRO MODEL W OIL BURNERS AT 75% BOILER EFFICIENCY

Burner Size	Maximum Gals. Per Hour	B.T.U. Release Per Hour	Rated Capacity Boiler Horsepower	Steam Radiation, Sq. Ft., E.D.R.
W-2½	11	1,650,000	36.9	5,150
W-3	15	2,250,000	50.4	7,030
W-4	25	3,750,000	83.8	11,720
W-5	35	5,250,000	117.3	16,400
W-6	50	7,500,000	167.5	23,400
W-7	70	10,500,000	234.8	32,800
W-8	100	15,000,000	335.0	46,800
W-8½	120	18,000,000	403.0	56,200
W-9	145	21,750,000	486.0	68,000
W-10	200	30,000,000	670.0	93,600

INDUSTRIAL COMBINATION GAS-OIL BURNERS USING NATURAL AND INDUCED DRAFT:

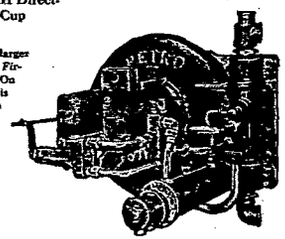
Model WG Industrial Gas-Oil Burner will fire low or high pressure natural gas or any grade of fuel oil, with natural or induced draft. It is a fully integrated unit in which primary air for combustion of both oil and gas is supplied by the oil burner fan. Secondary air is admitted around the burner opening, eliminating the deep firebox pit and brick checkerwork required in the conventional rotary oil burner setting. Enclosed control panel, with instruments installed, wired and tested at factory, may be mounted at burner or on boiler room wall. Fuel change is accomplished quickly by electrical switching. Oil capacities, 11 to 120 gph; gas 1,650,000 to 18,750,000 Btu input.

INTEGRAL AIR REGISTER CONVERSION KITS:

The Petro Air Register is ideal for converting oil installations to gas-oil. The rotary oil burner bolts on the register making an easy field conversion that is complete with refractory, combustion throat and the gas ring. All combustion air is admitted around the burner opening as required by Underwriters' Laboratories standards.

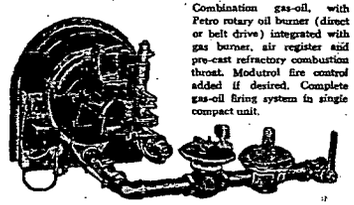
Model WD-AH Direct-Driven Rotary-Cup Type Burner

On W-5 and larger sizes, Modutrol Firing is standard. On smaller sizes, it is supplied as an extra.



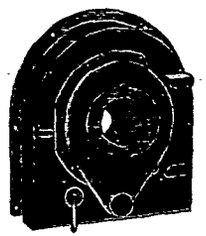
Model WG Burner

Combination gas-oil, with Petro rotary oil burner (direct or belt drive) integrated with gas burner, air register and pre-cast refractory combustion throat. Modutrol fire control added if desired. Complete gas-oil firing system in single compact unit.



New Air Register

Readily converts any Petro Rotary Oil Burner to combination gas-oil firing. Kit includes register, gas controls and standard parts groups. A minimum of installation time required. Air registers available also for straight oil firing.

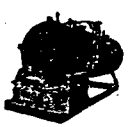


• All Petro Model W burners are available in either direct- or belt-driven *motor* types.

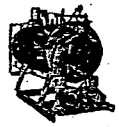
YORK-SHIPLEY, INC.

MAIN OFFICES AND PLANT, YORK, PA.

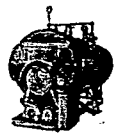
OIL AND/OR GAS-FIRED YORK-YS POWER EQUIPMENT FOR INDUSTRY


STEAM-PAK


HOT WATER BOILERS


YORK-POWER PAKETTE


FIRE-PAK


SCOT-PAK


YORK-POWER BURNER

STEAM-PAK GENERATORS

High or low pressure steam. No. 3, 5 or 6 oil and/or gas firing. Fully automatic operation

Model No.	Nominal HP	Steam Lbs/Hr	EDR Sq Ft Steam	H In.	W In.	L In.
SPE-10	10	345	1397	44¾	35	85 15/16
SP-32A-15	15	525	2100	58½	47¾	90¾
SP-32A-20	20	690	2780	58½	47¾	90¾
SP-54A-30	30	1035	4250	85½	67¾	118
SP-40	40	1380	5533	71½	59 5/16	122
SP-50	50	1725	7000	71½	61 5/16	127
SP-60	60	2070	8375	71½	61 5/16	165½
SP-75	75	2587.5	10469	78	71⅜	151¼
SP-80-O.S.	80	2760	11160	78	71⅜	151¼
SP-100-O.S.	100	3450	13958	78	71⅜	151¼
SP-100	100	3450	13958	85¼	73½	165½
SP-125	125	4312.5	17447	85¼	73½	185½
SP-68-150	150	5175	20937	88¼	71	187½
SP-200	200	6900	27916	115¼	93¼	210
SP-250	250	8625	34895	115½	91¼	247½
SP-300	300	10350	41874	118	93	247½
SP-350	350	12080	48800	118	102¾	277¾
SP-400	400	13800	55800	135	107½	293½
SP-500	500	17250	69200	133	114¼	256¾
SP-600	600	20700	83700	133	114¼	329

YORK-POWER HOT WATER BOILERS

Designed specifically for hot water heating applications. No. 3, 5, and 6 oil and/or gas-firing. Fully automatic operation.

Model No.	Mfg. Cert. Rating HP	Mfg. Cert. Rating Sq Ft Water	Gross output 1000 Btu/H	H In.	W In.	L In.
SPW-57-40	40	6,833	1340	81	63	109½
SPW-57-50	50	11,167	1725	81	63	123¾
SPW-57-60	60	13,400	2010	81	61	157¾
SPW-57-90	90	20,100	3015	88	68½	151¾
SPW-57-100	100	22,333	3350	88	68½	171¾
SPW-57-125	125	27,916	4188	95	74¾	165¾
SPW-57-150	150	33,500	5025	95	74¾	185¾
SPW-57-175	175	39,150	5822	108	80½	221¾
SPW-57-250	250	55,833	8375	121	93	210¾
SPW-57-300	300	67,000	10,050	121	93	247¾
SPW-57-350	350	78,100	11,720	121	94½	277¾
SPW-57-500	500	111,500	16,750	133	103	279¾

YORK-POWER PAKETTES

High or low pressure steam or hot water. No. 3, 5 or 6 oil and/or gas firing. Fully automatic operation

Model No.	Nominal HP	EDR Sq Ft Steam	EDR Sq Ft Hot Water	H In.	W In.	L In.
Heavy Duty						
SP-57-A-20	20	2780	4600	69¼	51	80
SP-57A-30	30	4185	6700	69¼	51	80
SP-57-A-40	40	5583	8933	69¼	51	80
SP-57A-50	50	7000	11167	69¼	61	89
Regular Duty						
SP-56-10	10	1397	2287	44¾	35	78¼
SP-56-15B	15	2100	3400	44¾	35	78¼
SP-53A-15	15	2100	3400	58½	43	85¾
SP-53A-20	20	2780	4600	58½	43	85¾
SP-54A-30	30	4185	6700	85½	61	115
SP-53-40	40	5583	8933	71½	61	109
SP-53-50	50	7000	11167	71½	61	127

FIRE-PAK BOILERS

Packaged fire-tube boilers for low pressure steam or hot water. No. 3, 5, or 6 oil and/or gas firing. Fully automatic operation.

Model No.	Mfrs Certified Rating HP	Steam EDR Sq Ft	Hot Water EDR Sq Ft	H In.	W In.	L In.
YFB321	35	4930	7890	95	57	96
YFB301	40	5575	8920	95	57	109
YFB351	46	6400	10240	95	57	115
YFB401	61	7080	11320	95	57	124
YFB431	66	7810	12500	100	64	115
YFB501	63	8850	12850	100	64	124
YFB4891	90	13450	19900	104	70	143
YFB48106	107	14220	22750	104	70	158
YFB5491	121	18020	27040	115	76	154
YFB54106	135	17450	27900	115	76	169
YFB6098	150	20950	33500	138	83	163
YFB60117	150	20950	33500	138	83	182
YFB66119	225	31400	50300	138	83	185
YFB78127	280	38850	62300	139	95	194
YFB78140	343	47830	76500	140	103	207
YFB84127	615	58320	93200	173	103	201
YFB84140	485	67180	107500	173	109	215

SCOT-PAK BOILERS

Packaged scotch marine boilers for forced draft, natural draft or induced draft firing. No. 3, 5, & 6 fuel oil and/or gas. Fully automatic.

Model No.	Mfr. Certified Rating HP	Mfr. Certified Rating Sq Ft Steam	H In.	W In.	L In
YSM56-40	52	7290	73⅝	58⅜	117⅛
YSM56-45	59	8220	73⅝	58⅜	126⅜
YSM56-50	65	9125	73⅝	58⅜	145⅛
YSM56-60	78	10920	78⅜	64⅜	133⅛
YSM56-70	91	12750	78⅜	64⅜	154⅛
YSM56-85	111	15500	84¾	70¾	137⅛
YSM56-100	130	18250	84¾	70¾	178⅛
YSM-56-125	163	22750	90¾	76¾	173⅛
YSM-56-150	196	27350	90¾	76¾	196⅜
YSM-56-175	228	31900	90¾	83¾	201⅛
YSM-56-200	261	36450	96¾	83¾	221⅛
YSM-56-250	326	45550	108⅛	94⅛	220⅛
YSM-56-300	391	54600	114⅛	102⅛	223⅛
YSM-56-350	400	55800	114⅛	102⅛	247

YORK-POWER BURNERS

Horizontal Rotary burners for conversion application. No. 2, 4, 5 or 6 oil and/or gas firing. Factory-coordinated fuel burning systems used with burners to provide fully automatic system.

Burner Model	Max GPH	EDR Sq Ft Steam at Boiler Efficiency of 60 percent	65 percent	70 percent	78 percent
A-15	6	2255	2440	2630	2930
A-45	15	5630	6095	6565	7320
A-90	30	7500	8120	8750	9750
B-100	33	12400	13430	14450	16120
B-125	43	15740	17080	18390	20480
C-175	59	22500	24360	26250	29250
C-225	75	28150	30495	32800	36550
D-300	100	37550	40600	43750	48750
D-400	125	50650	54800	59000	65700

Aerco Corporation

Paris Avenue, P. O. Box 248
Northvale, N. J.

AERCO'S Controlled "HTR" (Heat Transfer Rate) Heat Exchanger is a Compact Storage Water Heater featuring an Internal Compensator which:

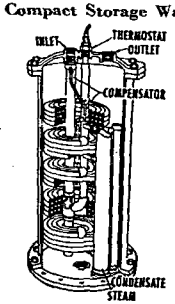


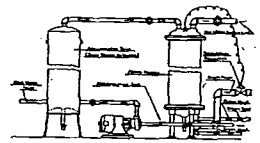
Figure 1—Sectional drawing of Aerco Compact Storage Water Heater

- *Anticipates* changes in *required steam flow before* the actual heat transfer takes place.
- *Meters the steam flow* accurately in direct proportion with load.
- *Prevents surges* in steam flow, *eleminating need for large storage.*
- *Maintains ACCURATE control* of outlet water temperature.
- *Prevents OVERRIDE* of outlet temperature on sudden interruption of water flow.

For **UNIFORMLY VARYING LOADS** (i.e. Hospital General Water Supply), the ability of the above heater to prevent steam flow surges **ELIMINATES** the need for large **STORAGE.**

For **NON-UNIFORMLY VARYING LOADS** or periodic high peak flows and low average demands (i.e. Small Commercial Laundry), the AERCO COMPACT STORAGE WATER HEATER with an Accumulator and Recirculating Pump (as shown in Figure 2) gives a low cost combination which also permits:

- *Minimum storage volume* for peak demands.
- *Minimum steam input* for required load.
- *Accurate* controlled outlet water temperature under *all* load conditions.


Figure 2—Aerco Compact Storage Water Heater in combination with Accumulator and Recirculating Pump

CONSTRUCTION

Shell fabricated to *ASME* Code Standards, of Flange Quality Steel with plastic or copper lining. Shell ends Cast-iron or Bronze. Internal parts of Copper or Brass. Tested for 125 psig operating pressure. Capacities, 300 to 25,000 gph.

OPERATION

Cold water enters top, flows down through the Compensator, then up across steam coils to discharge at top. Steam feeds into coils through Steam Riser and leaves through Condensate Return.

Compensator anticipates and controls heat transfer rate by *sensing* changes in flow or incoming temperature before water passes over the heating coils. It accomplishes this by causing variations of temperature to occur at the Regulator bulb which are definitely related to changes in flow and incoming temperature as well as the final temperature.

Send for booklet: "THE NEW APPROACH To the Design of Hot Water Supply Systems" and Bulletin #22 for capacity data and sizes.

BELL & GOSSETT COMPANY

MORTON GROVE, ILLINOIS

Canadian Licensee: S. A. Armstrong, Ltd., 1400 O'Connor Drive, Toronto 16, Ont.

Reg. U. S. Pat. Off.

Hydro-Flo HOT WATER HEATING & COOLING SYSTEMS & SPECIALTIES



B & G Booster

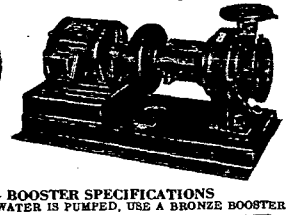


B & G "PD" Booster

B & G Universal Pump



Oil Lubricated Bearings



B & G Booster

The B & G Booster is the basic unit of the B & G Hydro-Flo Forced Hot Water Heating System. The prime requisite of such a pump is quiet operation and in this respect the B & G Booster is completely outstanding.

The oversized, special alloy, mirror-finished shaft provides large, super smooth bearing surfaces. The heat-treated thrust collar, integral with the shaft, prevents end movement. Bronze, sleeve type bearings are extra long to maintain the shaft in exact alignment and help assure smooth, quiet operation. The Seal assembly provides positive protection against water leakage into bearings. The true centrifugal design impeller operates with a minimum of slippage, assuring a full quota of water with each revolution.

B & G Boosters have a genuine oil-circulating lubrication system—another reason for quiet, dependable operation. Oil is drawn from the oil well by wool fiber wicking and dropped on the horizontal bearing surfaces. Medium grade motor oil is used, and only a few drops at infrequent intervals are required.

B & G Universal Pump

Designed for forced hot water heating systems in apartment and office buildings, factories, schools, etc. The Universal is not an ordinary centrifugal pump, but is specially designed to assure quiet operation so essential in heating system applications.

B & G BOOSTER SPECIFICATIONS

WHERE SERVICE WATER IS PUMPED, USE A BRONZE BOOSTER

Size Number	Pipe Size, Inches	Standard Equipment[a] 115 Volt, 60 Cycle, Single Phase
"75"	¾ Flanged	⅓ H.P.
"100"	1 Flanged	½ H.P.
"125"	1¼ Flanged	½ H.P.
"150"	1½ Flanged	¾ H.P.
1¼"	1½ Flanged	¾ H.P.
2"	2 Flanged	¾ H.P.
2½"	2½ Flanged	¾ H.P.
3" LDS LOW DELIVERY	3 Flanged	¾ H.P.
HDS HIGH DELIVERY	3 Flanged	1½ H.P.
1" HV HIGH	1 Flanged	¾ H.P.
1¼" HV VELOCITY	1¼ Flanged	¾ H.P.
1½" HV	1½ Flanged	¾ H.P.
1" PR	1 Flanged	¾ H.P.
PD 35-S	3 Flanged	[b] ½ H.P.—1 PH.
PD 35-T	3 Flanged	[b] ½ H.P.—3 PH.
PD 37-S	3 Flanged	[b] ¾ H.P.—1 PH.
PD 37-T	3 Flanged	[b] ¾ H.P.—3 PH.

[a] Motors with special current characteristics at extra charge as listed in Price Sheets.

[b] PD-35-S and PD-37-S are available with 115, 208 or 230 Volt Motors and have built-in overload protection. PD-35-T and PD-37-T are available with 208-230 Volt or 415-460 Volt Motors.

External overload equipment must be provided on three phase motors.

B & G UNIVERSAL PUMP SPECIFICATIONS

Dimensions are subject to change. When needed for layout, write to factory for certified print.

Pump[a] Cat. No.	Motor hp	Approximate Overall Dimensions, Inches		
		Height	Width	Length
U-1S & U-1T	1	15	16	31
U-2S & U-2T	1½	15	16	31
U-3S & U-3T	2	16	16	32
U-4S & U-4T	3	16	16	32
U-5S & U-5T	5	17	16	33
U-6T	5	18	17	37
U-7T	5	19	17	38
U-8T	7½	22	17	41
U-9T	7½	19	17	38
U-10T	10	22	17	42
U-11T	10	21	17	41
U-12T	15	22	21	44
U-13T	20	26	23	43
U-14T	25	30	22	53

[a] "S" denotes single phase motors; "T", three phase motors.



Straight-Angle Pattern 1 in.—1¼ in. — 1½ in.



Angle Pattern 2 in.—2½ in.—3 in.

B, & G Flo-Control Valves

For preventing gravity flow in forced hot water systems and to permit summer-winter operation of a B&G Indirect Water Heater. Sizes 1, 1¼ and 1½ in. are made in combination straight-angle patterns, and can be installed in either horizontal or vertical pump lines. Easy cleaning is a feature of B & G Flo-Control Valves—no need to break pipe connections for removal of foreign matter.

	Valve Number and Size				Connections
SA Straight-angle pattern Angle pattern	SA 1" A 2"	SA 1¼" A 2½"	SA 1½" A 3"		Screwed Flanged 1 end—screwed 1 end
Straight pattern	S 2"	S 2½"	S 3"	S 4"	Flanged 1 end—screwed 1 end
Bronze—angle pattern Bronze—straight pattern	BA 1" SB ¾"	BA 1¼"			Sweat Connections Sweat Connections

B & G Monoflo Fittings—Cast-Iron or Copper

B & G Monoflo Fittings permit the use of a single pipe main instead of conventional flow and return lines. Installed at the junction of the radiator risers to the single main, they assure proper diversion of water into each radiator, convector or baseboard, regardless of its position in the system. They can be installed at either the return or supply riser, with the other main connection being a common tee.

CAST IRON FITTINGS		COPPER FITTINGS	
Size	Tapping Sizes, Inches	Size	Tapping Sizes, Inches
¾"	¾×½ RETURN ONLY	¾"	¾×½ RETURN ONLY
1"	1 ×½ OR ¾	1"	[a] 1 ×½ OR ¾
1¼"	1¼×½, ¾ OR 1	1¼"	[a] 1¼×½ OR ¾
1½"	1½×½, ¾ OR 1	1½"	1½×½, ¾ OR 1
2"	2 ×½, ¾, 1 OR 1¼	2"	2 ×½, ¾ OR 1
2½"	2½×½, ¾, 1 OR 1¼		
3"	3 ×½, ¾, 1 OR 1¼		

[a] New Red Wing Fittings serve as either supply or return fittings.

B & G Relief and Reducing Valves

B & G Relief Valves, for relieving excess boiler pressures in hot water heating systems and in the lines of water service systems are designed and built to *ASME* specifications, tested by the National Board and labelled with the *ASME* symbol.

B & G Reducing Valves have all working parts made of brass, with built-in strainer and Anti-Siphon Check Valve. Adjustable to meet varying building heights.

Nos. 175–250–350–480 Relief Valves

No.	Size, Inches	Capacity Btu/Hr	Standard Setting, Lb[a]
175	¾	175,000	30
250	¾	250,000	30
350	¾	350,000	30
480	¾	480,000	30

[a] Also available at 15, 25, 45, 50, 75, 100, 125 lb settings.

Nos. 750–1050–3300–4100 Relief Valves

No.	Size, Inches	Capacity Btu/Hr	Standard Setting, Lb
750	1	750,000	30[b]
1050	1¼	1,050,000	30[b]
3300	1½	3,300,000	30[b]
4100	2	4,100,000	30[b]

[a] Also available at 15, 25, 45, 50, 75, 100, 125 lb settings. [b] Also available at 55, 35, 45, and 50 lb settings.

No. 12 Reducing Valve

All working parts are of brass, with easily cleaned, built-in strainer and Anti-Siphon Check Valve. Factory adjusted at 12 lbs; suitable for 1, 2 and 3-story buildings.

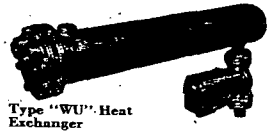
B & G Motorized Valves

Thermostatically operated valves used to control boiler water flow through individual circuits of zoned heating systems.

B & G Comfort Control System

Outdoor type, wind-compensating temperature control. The Regulator projects through the building wall, with the indoor end warmed by water in the radiation circuit. A small amount of this heat is conducted to the outside end, where it is dissipated at a rate dependent upon temperature and wind velocity. This heat dissipation governs the Control Valve, which permits hot boiler water to enter the system in required amounts.

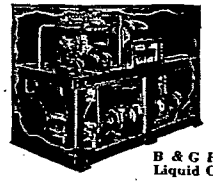
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Type "WU" Heat Exchanger



Type "SU" Heat Exchanger



B & G Package Liquid Cooler

Hydro-Flo PRODUCTS



B & G Cast Iron Monoflo Fitting



B & G Copper Monoflo Fitting

On all orders specify number of supply and return fittings.



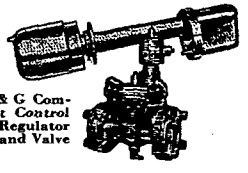
B & G Relief Valve



B & G Reducing Valve



B & G Motorized Valve



B & G Comfort Control Regulator and Valve

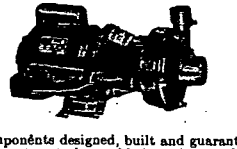
B & G HEAT EXCHANGERS AND WATER HEATERS

TYPE WU—A shell and tube heat exchanger with hot boiler water pumped through the shell by a B & G Booster, making possible big capacity in a small unit. Pumping water through the heater also affords excellent temperature control and permits use of much smaller pipe and fittings.

TYPE "SU"—For heating water with steam. An ideal selection for industrial plants or wherever large volumes of hot water are required continuously for service water supply or process work. No storage tank required—the large heat transfer surface in these units heats water instantly as needed. Available in a wide range of capacities.

B & G Centrifugal Pumps

Design and construction based upon years of experience in the industrial field. Rugged, compact units—built to withstand strain of continuous operation. Semi-open or enclosed impellers—motors flexible coupled or integral with pump. *Series 1522 illustrated.*



B & G Package Liquid Coolers

The only unit of its kind with all major components designed, built and guaranteed by one manufacturer. Everything included, no extras to buy—this is truly a package unit. Because of the many features included as standard equipment, this unit can be installed at smaller final cost. A complete line of refrigeration components is also available.

AMERICAN TUBE PRODUCTS, INC.

General Offices: 100 Pulaski Street, West Warwick, R. I.

AMERICAN

Nationwide distribution through branches and plumbing and heating wholesalers

Foreign Sales through home offices

Diaphragm-type, Pressurized Expansion Tanks • "O" Ring Type Expansion Joints
Hot Water Heating Accessories

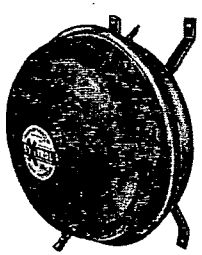
EXTROL EXPANSION CONTROL FOR HOT WATER HEATING SYSTEMS

APPLICATION

The Extrol Expansion Control Unit is a diaphragm-type, pressurized expansion tank developed by American Tube Products, Inc., specifically for use on hot-water heating systems. It replaces the conventional expansion tank.

DESCRIPTION

The Extrol Unit is constructed of two steel *ASME* hemispheres bonded together in a compact, sealed unit. A butyl diaphragm or bladder separates the unit into two chambers, one of which is factory pressurized with air. Thus there can be no contact between system water and air. Mounting and piping provisions are furnished.

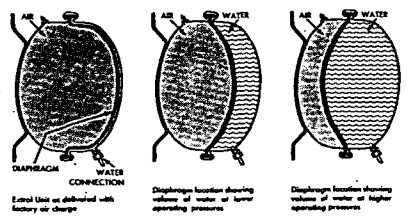


ADVANTAGES

- No loss of air cushion through absorption by boiler water
- Air problems in system substantially reduced
- Use of compressed air to charge expansion tank reduces the required tank volume
- No maintenance or servicing—no draining or recharging necessary
- Reduces system corrosion
- Smaller space requirements
- Simpler installation
- Lower initial cost

OPERATION

As thermal expansion build-up develops in the system, water enters one chamber, displacing the diaphragm towards the other chamber and further compressing the air cushion. When water temperature decreases, the resulting contraction of water pressure allows the air cushion to deflect the diaphragm, returning water to the system.



INSTALLATION

Extrol units may be installed at any point in the system and in any position. Positioning for drainage is not required.

CAPACITY

Extrol volume is based on the *ASME* Formula as it appears in the ASHAE GUIDE under "Hot Water Heating Systems."

HISTORY

When the closed type hot-water heating system was introduced, permitting operation at higher temperatures, the expansion tank, usually installed in the basement, replaced the open attic tank.

Although superior to the open attic tank, the expansion tank created several adverse operating conditions.

For one, because of the solubility of air and water, the conventional expansion tank readily lost its air cushion. Thus the air gravitated to other parts of the system, causing improper heating, excessive venting, noise, and corrosion.

Introduction of the Extrol unit, with its flexible butyl rubber barrier, prevents the loss of air. Furthermore, the barrier retains the air charge above atmospheric pressure, so that water is held in position at high points in the line as system water is alternately heated and cooled.

Until recent years, no suitable material was available for use as a diaphragm of this type—flexible, yet air-tight.

The Extrol Unit thus helps solve a number of problems formerly common to hot water heating systems.

Write for complete information and specifications on Extrol units for commercial, industrial, institutional, and residential applications

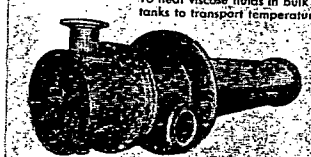
DAVIS ENGINEERING CORPORATION

1064 East Grand Street, Elizabeth 4, New Jersey
30 Rockefeller Plaza, New York 20, New York

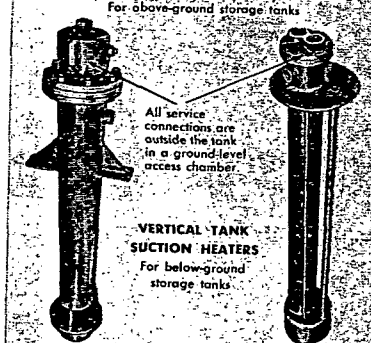
Paracoil
HEAT TRANSFER EQUIPMENT

Paracoil fuel oil heaters for every application.

TANK SUCTION HEATERS
To heat viscose fluids in bulk storage tanks to transport temperatures.



HORIZONTAL TANK SUCTION HEATERS
For above-ground storage tanks



VERTICAL TANK SUCTION HEATERS
For below-ground storage tanks

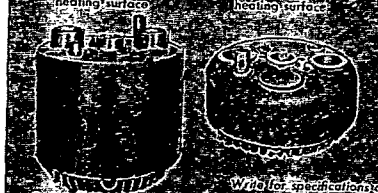
For high-pressure STEAM as heating medium

For hot circulated FLUID as heating medium

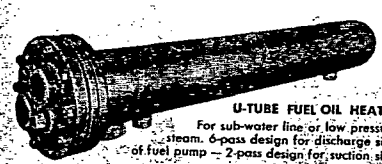
TANK SUCTION BELL HEATERS
For below-ground storage tanks—small-to-moderate demand installations.

12 square feet of heating surface

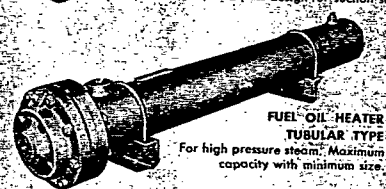
6 square feet of heating surface



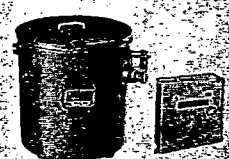
U-TUBE FUEL OIL HEATER
For sub-water line or low pressure steam. 6-pass design for discharge side of fuel pump — 2-pass design for suction side.



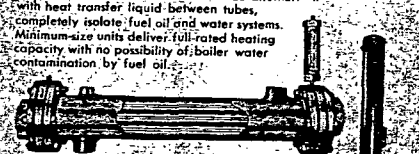
FUEL OIL HEATER TUBULAR TYPE
For high pressure steam. Maximum capacity with minimum size.



PARAPROBE OIL LEAK DETECTION SYSTEM
Protects against oil contamination by continuously checking purity of condensate leaving steam-type fuel oil heaters.



"THERMO-FILM" SAFETY TYPE FUEL OIL HEATER
Double tube elements (one tube inside another) with heat transfer liquid between tubes, completely isolate fuel oil and water systems. Minimum-size units deliver full-rated heating capacity with no possibility of boiler water contamination by fuel oil.



SELF-CLEANING TYPE HEATER
Interior baffles can be moved back and forth along tube surfaces to scrape away fouling deposits and restore full efficiency.

INDIRECT TYPE GAS FIRED FUEL OIL HEATER
Combination steam evaporator and tubular fuel oil heater. No flame contacts oil tubes; gas generates steam to supply heat. Maintains oil line temperatures during periodic main boiler plant shutdown.



Write for specifications on any of the above heaters.

GENERAL FITTINGS COMPANY

East Greenwich, Rhode Island, U. S. A.
P. O. Box 151K
HEAT EXCHANGERS

GENERAL Converters



GENERAL RC CONVERTERS

Steam Converter

GENERAL RC Converter ... shell and tube heat exchanger, two pass construction, steam in shell, water in tubes. U-bend copper tubing eliminates expansion strains. Removable tube bundle. Used for forced circulation hot water heating systems. Also for maintaining temperature in swimming pools. Capacities to 1800 gallons per minute flow.

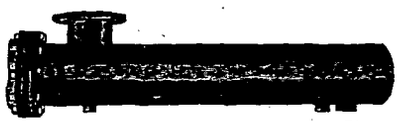


GENERAL BC CONVERTERS

High Temperature Water Converter

GENERAL BC Converters ... for high temperature hot water heating systems—high temperature, high pressure water circuit in tubes, medium temperature water circuit in shell. Number of passes in head varies from two to ten, depending upon design conditions and is combined with varying baffle pitch to give very high heat transfer efficiency. Design pressures tube side 400 psig and 660 psig. Inhibited admiralty tubing, semi-confined high pressure head gaskets, high temperature nozzles constructed with welding neck flanges. Design permits installation of temperature control bulb in shell for sensitive response. Capacities to 1800 gpm flow medium temperature circuit.

GENERAL Instantaneous Heaters



GENERAL ST Instantaneous Hot Water Heater

Steam Type

GENERAL ST Instantaneous Hot Water Heater ... for hot water supply in hospitals, institutions, hotels and industrial plants. No storage necessary—heats water as fast as it is drawn—should be sized to peak load requirements—can be used as a central hot water supply or as a spot source at point of use, such as a booster to provide sterilizing rinse 190° water for dish-washing machines. Capacities to 300 gpm.

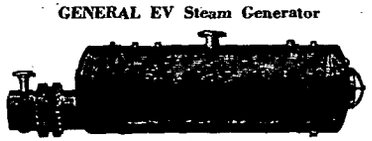


GENERAL BT Instantaneous Heater

Boiler Water Type

GENERAL BT Instantaneous Heater ... Baffle construction for greater efficiency with boiler water circulated through the shell—circulator must be installed on boiler water circuit with control bulb in service water outlet. Affords accurate regulation. Excellent hot water supply for apartment buildings.

GENERAL EV Steam Generator



High Temperature Fluid Type

GENERAL EV Steam Generator ... evaporator designed for service with high temperature water, Dowtherm, Aroclor or Heat Transfer Oil through the tubes—boiling water in lower half of shell, steam in upper half. Built-in steam separator, removable tube bundle. Design pressure to 660 psig. Design temperature to 800F tube side.

GENERAL BO Fuel Oil Preheater



Baffle Type

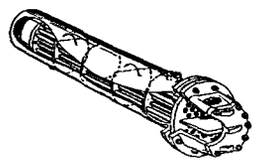
GENERAL BO Fuel Oil Preheater ... Steam in tubes—oil in shell. Transverse baffles are designed to give outstanding turbulence and heat transfer characteristics within conservative pressure drop limitations. For installation on discharge side of pump. Can be used in conjunction with GENERAL Suction Oil Heater in tank. Capacities to 3800 gpm.

For full details, write Heat Exchanger Division

KILLEBREW Engineering Corporation

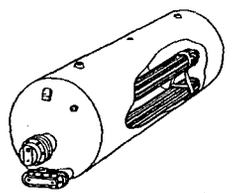
8640 Pardee Lane, Saint Louis 23, Mo.

ASME FABRICATION



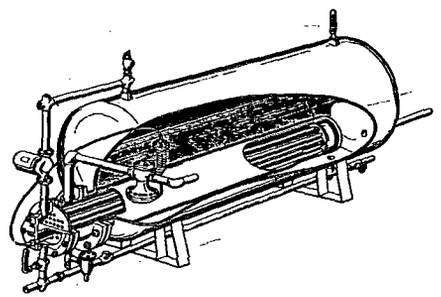
FRE-FLO INSTANTANEOUS WATER HEATER

Easily cleaned, straight tube with removable cover plate and floating head. Designed for heavy loads and fouling fluids. Standard sizes from 50 gph up.



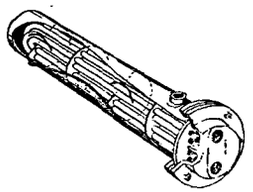
FRE-FLO STORAGE WATER HEATER

Steel (rust resistant lining optional) or alloys. Copper tubes on square pitch or flat bundles to set low in tank for use as condensate cooler or used where heavy fouling requires complete access for cleaning.



FRE-FLO LOAD LEVELER

A simple, rugged, *packaged* water heater designed specifically for handling extreme surge loads, yet levels the steam load on your boiler. Will deliver one or two temperatures of your selection, individually controlled. The Fre-Flo Load Leveler is completely assembled, ready to connect to utilities. Sizes from 450 gph and up.



FRE-FLO RADIATION CONVERTER

Complete range of sizes. Used with steam or boiler water. Cast iron head, steel shell, copper U tubes and bronze tube supports or baffles.

WRITE US REGARDING YOUR HEAT EXCHANGE PROBLEMS
FRE-FLO CATALOG AVAILABLE

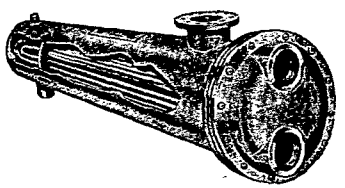
RICHMOND ENGINEERING CO., INC.

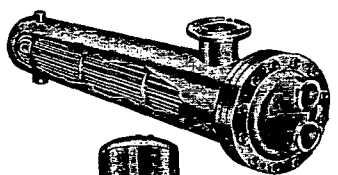
7th & Hospital Streets, Richmond, Va.

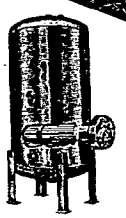
RECO TANKS, INC. Holts Chapel Road—Greensboro, N. C. P.O. Box 147—Columbia, S. C.

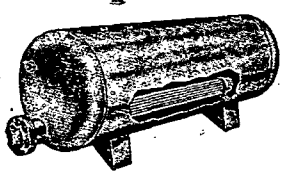
RECO

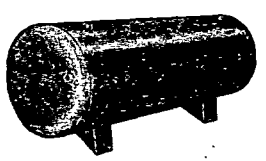
HEAT EXCHANGE EQUIPMENT











The **RECO** price is competitive! Delivery is fast and sure. All **RECO** products are guaranteed. Because **RECO** water heating plants handle maximum demand without straining, there will be a minimum of maintenance costs to add to your client's overhead. Specify **RECO** for your next heating job. For catalogs, write: Richmond Engineering Company, Inc., 7th & Hospital Sts., Richmond, Virginia.

CONVERTORS

For **forced circulation** hot water heating systems specify **RECO** convertors. Live steam in a steel shell heats water in a copper tubing network. Removable heating elements are seamless U-bend tubes to eliminate internal joints and expansion stresses.

INSTANTANEOUS HEATERS

Make it a **RECO** instantaneous heater when your steam supply is adequate for peak loads, and a continuous supply of hot water is needed. Live steam in a steel shell heats water or other liquids in a multiple-pass network of copper tubing.

STORAGE HEATERS

Vertical and horizontal **RECO** storage heaters furnish large quantities of hot water for use at irregular intervals. Removable heating elements of seamless copper U-bend tubes. Fabricated from steel, copper-silicon or stainless. Linings: Baked-on phenolic resin, galvanized, cement, copper or clad materials.

STORAGE TANKS

RECO custom-builds storage tanks to **your specifications.** Bulk tanks, hydro-pneumatic tanks, above or underground tanks . . . **ANY SIZE OR MATERIAL.** All carry the famous **RECO** one-year guarantee of materials and workmanship. Underwriter's label for oil tanks or *ASME* stamp for pressure tanks.

CHECK THIS LIST OF 20 RECO HEATING & PLUMBING PRODUCTS

Convertors • Hi Temp Hot Water • Convertors • Instantaneous Heaters • Kitchen Booster Heaters • Snow Melting Exchangers • Feed Water Heaters • Condensate Coolers • Economizers • Steam Generators • Oil Tank Suction Heaters • Fuel Oil Heaters • Pneumatic Tanks • Surge Tanks • Condensate Tanks • Expansion and Compression Tanks • Blow Off Tanks • Swimming Pool Heaters • Storage Heaters • Boiler Blow-Down Tanks • Flash Tanks

Sterling, Inc.

5208 West Clinton Avenue, Milwaukee 18, Wisconsin

Heating and Temperature Control Equipment

Distributed through leading Heating and Plumbing Wholesalers

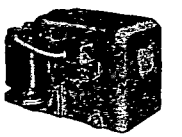
Sales Representatives in Principal Cities

Sterlco®

CONDENSATION PUMPS

4100-4200 Series

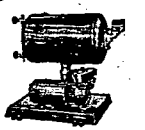
A complete line of low pressure units for ratings up to 15,000 sq ft EDR at 20 psig. Four sizes assure accurate, easy selection of pump to boiler requirement. Steel or cast iron tanks. Centrifugal pumps, mechanical type seal, built-in Simplex or Duplex. Many modifications available to suit special applications.



4100–4200 Series

3500 Series

Heavy duty units built as condensation or boiler return systems. Pedestal mounted tanks. Horizontal pumps. Ratings up to 65,000 sq ft EDR, up to 150 lb psig. Available in standard units or built to specification. Special units for packaged boilers.



3500 Series

3700 Series

Built with extra heavy cast iron tanks for underground returns and pit installations. Standard units from 2,000 to 20,000 sq ft EDR, 30 psig. Special units for larger ratings and higher pressures.

SPECIAL UNITS

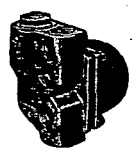
We specialize in designing and building special equipment to solve difficult condensate and boiler return pumping problems.

THERMOSTATIC TRAPS

Exceptional for keeping radiation free of air and condensate. Uses the "Sterlco Vacuum Thermostat" which closes uniformly regardless of pressure variation. If damaged, the thermostat fails in the closed position shutting off the radiation and giving positive indication that the trap needs attention. Trap maintenance greatly simplified. All traps equipped with replaceable seats. Three pressure ratings: 15, 65, 100 psig ½ in. L.P. angle body has smallest overall dimension for use on fin tube radiation.



Thermostatic Traps



Float and Thermostatic Traps, Popular 69B Illustrated

FLOAT AND THERMOSTATIC TRAPS

Used wherever it is desired to prevent flow of steam, but to permit passage of air and water . . . on unit heaters, large radiators and convectors, water heaters, steam main drips and other types of equipment where steam is used. Compact, rugged, easy to service. "Vacuum thermostat" vents air, float valve has plenty of capacity for large amounts of condensate. Sizes: ¾ in., 1 in., 1¼ in., 1½ in., 2 in. Maximum pressures: 15, 100 psig.

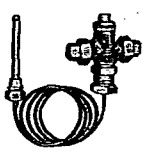


Thermotrols

120-A THERMOTROLS

150-E SERIES TEMPERATURE CONTROL VALVE

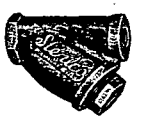
Self-contained, thermostatically operated radiator valves for controlling individual room temperatures. For two pipe steam or hot water systems. A simple, relatively inexpensive system of control for new or old buildings. Especially good for remodeling old systems. ½ in., ¾ in., 1 in. sizes. Angle, straightway, corner body patterns.



150-E Series Temperature Control Valve

"Y" STRAINERS

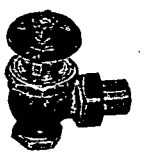
A complete line of "Y" Strainers. Brass strainers from ⅜ in. to 1 in. Cast iron from ½ in. to 3 in. 125 psig pressure rating.



"Y" Type Strainers

RADIATOR VALVES

A complete line of Radiator Valves from hot water supply valves to heavy duty bellows seal type valves for Vacuum heating systems.

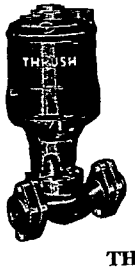


Radiator Valves

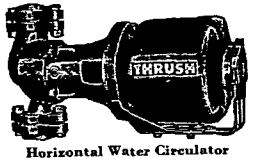
H. A. Thrush & Company

Peru, Indiana

FORCED CIRCULATING THRUSH FLOW CONTROL SYSTEM OF HOT WATER HEATING AND HEATING SPECIALTIES



Vertical Water Circulator



Horizontal Water Circulator

Thrush Flow Control System of Radiant Hot Water Heating assures continuous radiant heat with practically no variation of room temperature, regardless of outdoor weather changes. It offers an ideal means of control for use with radiant panels, radiant baseboards, convectors or radiators.

Circulation is forced and operation is entirely automatic. The Thrush Radiant Heat Control automatically compensates for outdoor weather changes. Provides an ample supply of hot water for kitchen, laundry and bath, both summer and winter. Wasteful overheating is completely prevented. Piping plans and engineering assistance are available to the trade.

THRUSH WATER CIRCULATORS

Forced circulating pumps for Hot Water Heating and Domestic Water Systems. Insure uniform heating. Quiet and efficient, long lived and vibration-free. Sealed-in lubrication. Vertical and Horizontal are made in seven sizes, ¾ in., 1 in., 1¼ in., 1½ in., 2 in., 2½ in. and 3 in. Hi-Head Vertical and Horizontal Thrush Water Circulators, designed for use with radiant baseboards, convectors, radiant panels, etc., which require higher heads, are also available in ¾ in., 1 in., 1¼ in. and 1½ in. sizes. Thrush Water Circulators are also available with Bronze body in 1 in. and 1¼ in. sizes, both horizontal and vertical. Female threaded connections.

THRUSH FLOW CONTROL VALVES

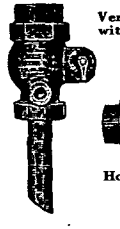
Special check valves for use with automatically fired boilers, automatically control circulation when installed with a Thrush Circulator. Close tight when Circulator is not running, prevent gravity circulation when heat is not needed in the radiators. This permits maintaining boiler water temperature high enough to heat domestic water, winter and summer, without overheating the building. Available with Air Tube which vents air from boiler into pressure tank, greatly improving heating efficiency by eliminating air from the system. 1 in. through 1½ in. valves also available with solder type outlet unions. Also made in angle type with or without air tube. Six sizes in Angle or Vertical types, 1 in., 1¼ in., 1½ in., 2 in., 2½ in. and 3 in. Horizontal Thrush Flow Control Valves, 1 in. through 2 in. sizes, are recommended for zoned systems and installations where head room makes it necessary to keep mains low.

THRUSH AIR ELIMINATOR

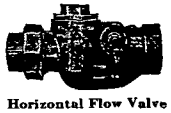
Thrush Air Eliminator Tank collects and removes air from water in the heating system. Air in the water moves with circulation and will be trapped and released in the Eliminator preventing its interfering with circulation. Designed primarily for floor and ceiling panel systems, it can also be used with baseboards, convectors or radiators. Made in seven sizes, all with 1 in., 1¼ in. or 1½ in. tappings.

THRUSH PRESSURE TANKS NOW HAVE VACUUM BREAKER

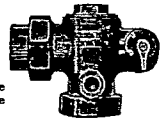
Thrush Pressure Tanks are made of heavy welded steel shells and heads and are guaranteed to be air tight. Each Thrush Pressure Tank comes equipped with a Thrush Vacuum Breaker at no extra cost. Makes tank draining simple and quick. It is only necessary to close valve in tank line, open drain and unscrew vacuum breaker.



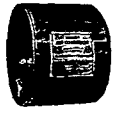
Vertical Flow Valve with Air Tube



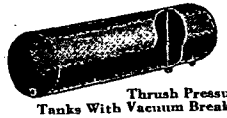
Horizontal Flow Valve



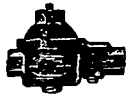
Angle Type Flow Valve



Thrush Air Eliminator Tanks



Thrush Pressure Tanks With Vacuum Breaker



Automatic Filling Valve ½ in., ¾ in. and 1 in.



No. 4 Relief Valve ¾ in. and 1 in.

THRUSH AUTOMATIC FILLING VALVES AND LOW PRESSURE RELIEF VALVES

Whenever system pressure drops below setting of valve, city water pressure will be admitted by Thrush Automatic Filling Valve until system pressure is restored. Pressure setting is adjustable.

Thrush Water Relief Valves protect heating boilers from excess pressures. Large diaphragm assures positive opening and closing. Convenient testing lever.

H. A. Thrush & Company

Peru, Indiana



ASME RATED No. 46 THRUSH SAFETY RELIEF VALVE FOR HOT WATER HEATING SYSTEMS

Rated output of 283,500 Btu based on steam capacity discharge. Primarily a water relief valve. Will also relieve steam should critically high temperatures occur. Testing lever. 30 lb pressure. ¾ in. inlet, 1 in. outlet. Same type available in settings to 125 lbs.



No. 75



No. 76

No. 75 Thrush High Pressure Water Relief Valves

To protect hot water heaters, range boilers and automatic water heaters from excess pressure. Factory setting 85 lbs. Adjustable at factory to 150 lbs maximum. *Other types available in ½ in., ¾ in. and 1 in. sizes.*

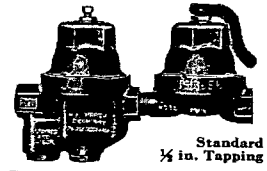
No, 76 High Pressure and Temperature Relief Valves

The valve illustrated above at right has an *added safety* feature. It not only guards against excessive pressures, but also relieves if excessive temperatures develop. A fusible element melts at 210 F. Other types available in ½ in. and ¾ in. sizes.



No. 170M - 170F AUTOMATIC RESEATING TEMPERATURE AND PRESSURE RELIEF VALVES

Where an Automatic Reseating Temperature and Pressure Relief Valve is required by code, the No. 170M which has a *male inlet or the No. 170F* which has a female inlet will provide the positive protection required. Reseats automatically after each operation. Has convenient testing lever. 6 in. stem or short stem. *ASME* rating 75 lbs, 450,000 Btu per hr; 100 lbs, 575,000 Btu per hr; 125 lbs, 700,000 Btu per hr. *AGA* rating, 550,000 Btu per hr input.



Standard ½ in. Tapping

DUAL CONTROL UNITS

Thrush Dual Control Units provide automatic filling and pressure relief for hot water heating systems. Proper water supply and working pressures in boiler maintained automatically. All have built-in strainers and check to prevent back flow from system. Low in cost, safe and dependable. Made in four types, all *with brass working parts. Adjustable* pressure setting. ½ in., and ¾ in. sizes.



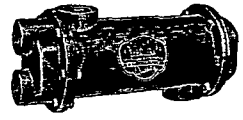
Adjustable Supply Tees for One Pipe Systems

Provide exact balancing of forced circulating one-pipe hot water heating systems. Available in bronze with solder connections for use with copper pipe and in threaded cast iron for steel pipe. Non-adjustable tees also available.



Thrush Twin-L Balancers

Used to join two supply or two return branch mains into one common main. Handy lever permits balancing of circulation in each circuit easily and quickly. Saves ten fittings commonly used to accomplish this. *Available in cast iron or* bronze in three sizes.



THRUSH WATER HEATERS

Highly efficient heat exchangers or converters. Provide Domestic Hot Water at low cost. Also used industrially for heating or cooling liquids. Fourteen sizes, for Hot Water or Steam. Pressure up to 125 lb for water, 50 lb for steam.

Constructed with heavy cast iron shells and straight seamless copper tubes. Both heads are removable for inspection of *tubes and for easy cleaning.*



1 *in. and 1¼ in.*, 2, 3 and 4 outlets



⅜ in. or ½ in.

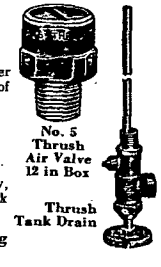
THRUSH MANIFOLDS AND BALANCING VALVES

Thrush Manifolds save time, space and labor installing Radiant Heating. Avail*able in 1 in. and 1¼ in. sizes, each with* two, three or four ⅜ in. or ½ in. threaded branch outlets. Also available in 1¼ in. and 1½ in. sizes with three or five ¾ in. solder branch connections. Ends have solder connections.

Manifold with Balancing Valves as shown above is used in return lines only. Used in supply lines without valves. Balancing Valves permit individual filling of each coil, eliminate air quickly and save time. Two types, three sizes, ⅜ in., ½ in. or ¾ in. Flared coil fittings for ⅜ in. and ½ in. *tubing, ¾ in. has solder connec*tions.

No. 5 Hot Water Air Valve

An inexpensive, dependable, automatic air vent for Hot Water coils, convectors, radiators, unit heaters, and high points of mains. Tapping ⅛ in., IPT.



No. 5 Thrush Air Valve 12 in Box

Thrush Tank Drain

THRUSH TANK DRAIN

Combines Drain Valve and Vacuum Breaker in one handy, *inexpensive fitting. Removing plug admits air to permit quick* draining of waterlogged tank.

Write For New Condensed Catalog

TACO HEATERS, INCORPORATED

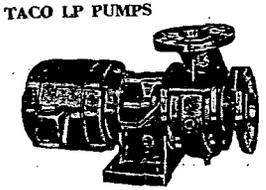
1160 Cranston Street, Cranston 9, Rhode Island

Corporate Office: 324 Madison Avenue, New York. In Canada: Taco Heaters of Canada, Ltd., 4 Gilead Place, Toronto 2

Serving the Hydronic Industry Since 1920

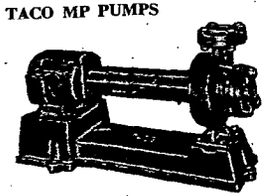
Rigid control of quality, advanced designs, nation-wide service and engineering assistance are your assurance that you can specify Taco with complete confidence in product performance.

TACO LP PUMPS



A line of single stage centrifugal pumps with: relatively few sizes; many interchangeable parts; broad range of performance; high efficiencies.

TACO MP PUMPS



A base mounted line of pumps with the rugged dependability large installations require.

TACO WATER HEATERS, CONVERTERS, AND CHILLERS



Taco Converters and Water Heaters provide hot water and radiation for commercial and institutional heating purposes. Chillers are designed for chilling water in the air conditioning and refrigeration fields.



TACO CIRCULATOR

Packed with quiet power and proven performance; sizes ¾ in. through 3 in.



TACO 170 CENTRIFUGAL PUMP

Excellent for cooling tower jobs; air-conditioning installations . . . wherever a small top-quality built centrifugal pump is needed.



TACO PERFECTA

Outstanding circulator; so quiet you won't believe it's running.



TACO FLOW REGULATOR

Limits water flow to specified requirements.



TACO FLOW CHECK

Easily installed in either angle or horizontal position, the Taco Flow Check prevents hot water from entering the system when the thermostat stops the circulator.



TACO PANELTROL

Adjustable to deliver water to radiant heating panels for any temperature between 110F and 150F even though boiler water is kept at higher temperature for summer-winter hot water.

TACO VENTURI FITTINGS

Powerful suction action draws water through radiator for positive circulation. May be used for above or below main radiation. Taco Super Venturis save you money by reducing branch sizes, copper tubing, pipe fittings and labor costs.



TACO AIR-SCOOP

Scoops air bubbles from hot water system by a series of baffles. Air is then vented to either atmosphere or expansion tank. When system is first filled . . . vent radiators and high points once . . . no return to job!



TACO TEMPERING VALVE

Guards against excessively hot domestic hot water. Mixes cold water with hot water from Taco Tankless for storage heater to provide tempered water at fixtures.



TACO AUTOMATIC AIR VALVES

Taco-Vent for water systems is leak-proofed. It has a porous plug that limits the flow of water through valve. The Steam Vent eliminates pinging, clicking or hissing.



TACO RELIEF AND REDUCING VALVES

Automatically maintains sufficient water and pressure in system. The Taco *ASME* Relief Valve handles 80 percent of all hot water heating jobs, and is set to relieve at 29 lbs. Reducing Valve is factory set at 12 lbs.



SEND FOR CATALOG CONTAINING COMPLETE INFORMATION AND ENGINEERING DATA

YULA CORPORATION

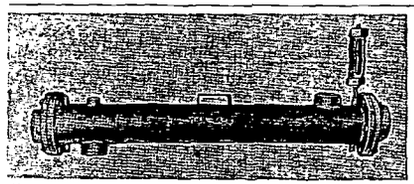
(formerly)

Yula Water Heaters Inc.—Established 1926.

330 Bryant Avenue, New York 59, New York

Designers and Manufacturers of Heat Transfer Equipment

Member Fuel Oil and Water Heater Manufacturers Association

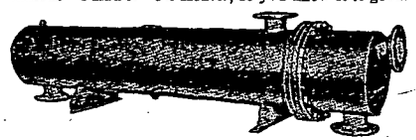


"YULATOM" FUEL OIL PROTECTION HEATER." Type "YA", approved by the Board of Standards and Appeals, City of New York-Cal. No. 678-57-S.A.

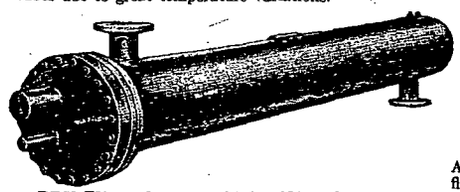
Here is a truly low cost, light weight, easy to install below water line oil heater.

Due to its counterflow operation the exit oil can be delivered at high temperatures.

Should a leak occur it is impossible for oil to get into Boiler or Piping. Made by the designer of the famous "Yulatrol" Oil Heater, so you know it is good.

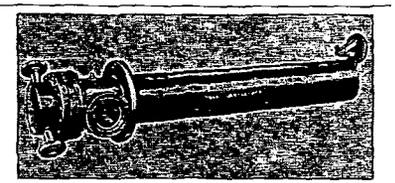


"U" TUBE CONVERTOR—used in hot water heating systems where steam is passed through the shell while circulated water passes through the tubes. "U" Tube construction allows for expansion of the tubes due to great temperature variations.

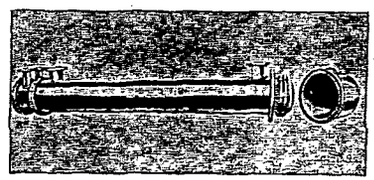


DRY EXPANSION TYPE FREON COOLER

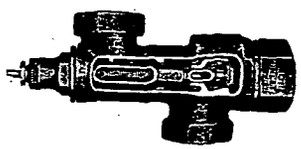
The Yula Dry Expansion Freon Water Cooler is for use in air conditioning, refrigerating, industrial and process cooling, using water and other fluids. The refrigerant passes through the tube side and is vaporized. The fluid being cooled passes through the shell side which is baffled. The carefully designed tube arrangement in these coolers provides high thermal efficiency and little pressure loss. Sound engineering has assured a positive oil return with the refrigerant in these units. Heat transfer and turbulent flow is assured.



OIL TANK SUCTION HEATER—used to lower the viscosity of oil in bulk storage tanks to facilitate pumping. Only oil being drawn off is heated making it unnecessary to heat entire tank.



FULL FLOATING HEAD EXCHANGER—used where straight tube construction is desirable affording easy access and cleaning of interior chambers while allowing for expansion of tube bundle.



YULA THERMOSTATIC MIXING VALVE

Automatically mixes hot and cold water and other fluids. Delivers mixture at any pre-determined constant temperature. Sizes ¾ in. through 3 in. Special calibrations are possible.

Special Features: Liquid expansion thermostat gives uniform power at all points of the range with positive, gentle throttling action. The disc controlling the ports has two seats on a 35 deg angle; precluding freezing. Seat is in one part. Castings are heavy bronze. Close temperature regulation is obtained because valve is extremely sensitive. Seat spring is made of durable, corrosion resistant stainless steel. Complete shut-off on hot and cold water port in "non-draw" periods.

We manufacture all types of Heat Transfer Equipment for use in many fields

Agents in Principal Cities

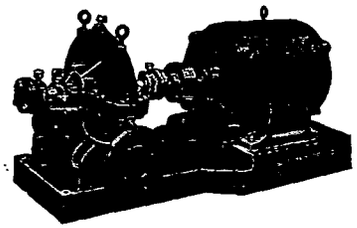
Buffalo Pumps

Division of Buffalo Forge Co.

450 Broadway, Buffalo, N. Y.

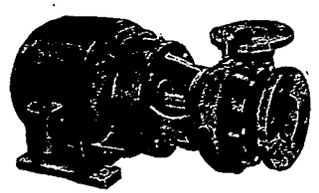
Canada Pumps, Ltd., Kitchener, Ont.

A complete line of Centrifugal Pumps, Single and Double Suction, Single and Multistage, for all types of heating and air-conditioning installations. Write for Engineering Bulletins mentioned below, or consult the Buffalo Engineering Representative in any principal City.



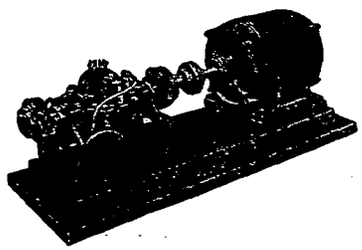
DOUBLE SUCTION SINGLE STAGE PUMPS

Extensively used for handling chilled water and other air-conditioning and clear-water applications including air washers, these pumps are available in capacities from 10 to 14,000 U.S. gallons per minute, against heads up to 350 ft. Highly efficient operation is provided by their simply formed water passages and their hydraulically balanced impellers (water enters each side of impeller with equal pressure and volume). All parts are ample for durability under hard service. Top half of the casing is readily removable for servicing without disturbing piping. Write for *BULLETIN 955S.*



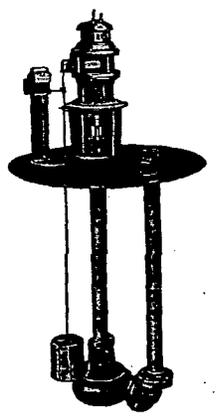
CLOSE-COUPLED SINGLE SUCTION PUMPS

These pumps offer the advantages of compactness, high performance and simple installation without need for shaft alignment. (A 4 in. pump, delivering 600 gpm at 60-ft head, requires less than 2 cu ft of space.) Available in 1 in. to 6 in. discharge sizes. Design permits vertical or horizontal installation, with discharge adjustable to desired angle. These pumps are suited to handling hot water with low submergence. Write for *Bulletin 975E.*



SINGLE SUCTION MULTISTAGE PUMPS

Used for boiler-feed and other clear water service, these heavy-duty pumps are especially suited to delivering at high pressures and temperatures. Available in two- and four-stage models for capacities from 20 to 900 gpm at heads up to 1500 ft. Shaft is extra-heavy and oversize bearings support the shaft on each end. Top-half of pump casing is removable for servicing without disturbing piping. Write for *BULLETIN 980E.*



AUTOMATIC SUMP PUMPS

These are shipped as preassembled units ready to put into the sump pit, wire and start pumping. No priming is necessary, since there is no suction lift. Thrust load is carried by ball bearings, and the shaft is fully enclosed to prevent fouling with waste or stringy matter from the sump. All parts are readily accessible. Write for *BULLETIN 963H.*

DOMESTIC PUMP & MANUFACTURING CORP.

SHIPPENSBURG — "DOMESTIC" ® MAKERS OF DEPENDABLE COMMERCIAL PUMPS SINCE 1903 — PENNSYLVANIA

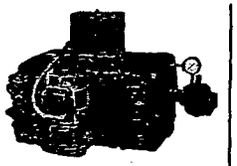
Products: Return Line Vacuum Heating Pumps; Condensation Pumps; Boiler Feed Pumps; Condensate Accumulators; Vertical, Horizontal and Flange Mounted Centrifugal Pumps; Air Separators.

FOUR COMPLETE LINES OF VACUUM HEATING PUMPS

Type VLR and VL Single and Duplex Units for Standard and Special capacities thru 65,000 sq ft EDR and up to 75 psi. Bulletins GS-110A and GS-131.

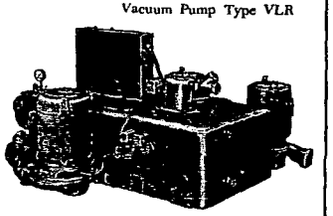
Type VCA with individually sized and separately controlled vacuum and condensate pumps. Air capacities thru 75 cfm and 20 in. Hg and condensate capacities thru 150 gpm at 75 psi. Bulletin GS-152.

Type VCM custom designed for each application. Automatic make-up. Pumps individually sized and controlled for maximum flexibility and economy. Available with 1, 2 or 4 vacuum pumps, each rated up to 90 cfm and 20 in. Hg and 1 to 4 boiler feed pumps, each rated up to 400 gpm and thru 75 psi discharge pressure. Bulletin GS-152.



Vacuum Pump Type VLR

The heart of all Domestic Vacuum Pumps is the multi-jet vacuum producer. Average air evacuating rate is twice its rating, reducing operating time and power cost. Removes up to 300 per cent of rated air capacity at start of heating cycle to shorten warm-up period. It is simple, efficient and capable of producing a near perfect vacuum. It will retain its original effectiveness year after year.

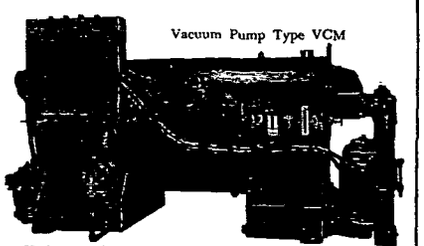


Vacuum Pump Type VCA

CONDENSATION & BOILER FEED PUMPS

Types SC and CC Single and Duplex units for systems thru 100,000 sq ft EDR and 60 psi. A compact dependable condensation pump with motor vertically mounted above floor dirt and water. Each motor is controlled by its own float switch. Low inlet receiver of lifetime cast iron. Bronze fitted centrifugal pumps for long life at peak efficiency. 1750 and 3500 RPM motors. Bulletins GS-202 and GS-221.

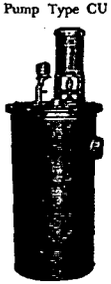
Type CMS and CM units provide automatic water make-up to replace water lost from the system and to maintain the most efficient boiler steaming level for systems thru 200,000 sq ft EDR and 60 psi. Single and Duplex units with receivers sized for ample condensate storage. 1750 and 3500 RPM bronze fitted centrifugal pumps designed to handle near boiling water. Bulletins GS-202 and GS-230.



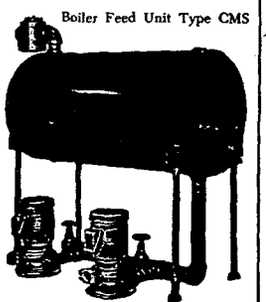
Vacuum Pump Type VCM

Type CU units for use where the return mains are below the floor or lower than the inlet of a horizontal condensation pump. The lifetime cast iron receiver is buried with the cover flush with the floor and with the motor and float switch extending above it. Bronze fitted centrifugal pumps have ball thrust bearings, stainless steel shafts and flexible couplings. Single and Duplex units for capacities thru 50,000 EDR and 75 psi, 1750 and 3500 RPM motors. Duplex units have automatic mechanical alternation and stand-by or double capacity in event of abnormal demand. Bulletin GS-240.

Underground Condensate Pump Type CU

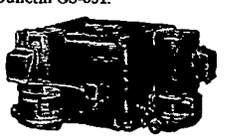


Boiler Feed Unit Type CMS



CENTRIFUGAL PUMPS

Horizontal, vertical and flange mounted pumps have same high quality that is built into all "Domestic" heating pumps. Heads to 230 ft, capacities to 450 gpm. Bulletin GS-691.



Condensation Pump Type CC

General Purpose Close Coupled Centrifugal Pumps



Putting Ideas to Work

FOOD MACHINERY AND CHEMICAL CORPORATION

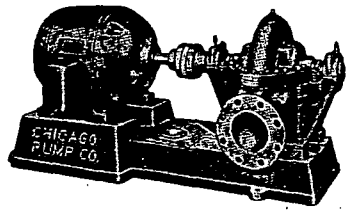
CHICAGO PUMP COMPANY

622 DIVERSEY PARKWAY • CHICAGO 14, ILLINOIS

Distributors In Principal Cities Throughout The United States and Foreign Countries

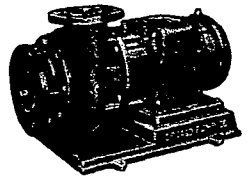
PRODUCTS

Condensation, House, Booster and Fire Pumps . . . Return Line Vacuum Heating and Boiler Feed Pumps . . . Circulating, Brine, Sewage, Sump, Sludge Pumps . . . Pneumatic and Tankless Water Supply Systems . . . Sealed Electrode Floatless Pump Controller . . . Automatic Alternator for Duplex Sets of Pumps



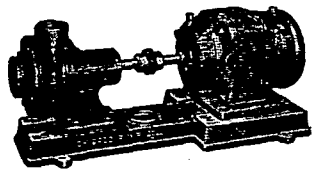
HORIZONTALLY SPLIT-CASE PUMPS

For house, fire and booster service. Ruggedly constructed, heavy duty, providing lasting, maintenance-free operation. The rotating assembly may be inspected without disconnecting the piping. Capacities up to 1900 G.P.M.—heads to 530 ft. Ask for Bulletins 98 and 102.



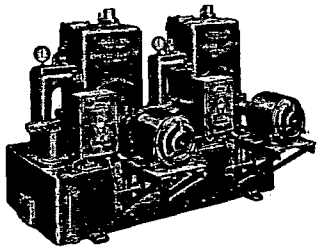
TYPE CC, CLOSE COUPLED BOOSTER PUMP

Designed to handle the toughest pumping jobs efficiently and in the least possible space. May be mounted in any position as long as the motor is not lower than the pump. Provided with leak-proof mechanical seal. Capacities up to 500 G.P.M.—heads to 215 ft. Ask for Bulletin 108.



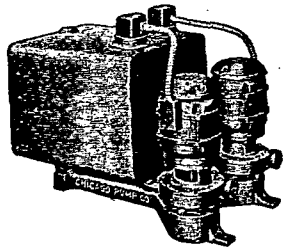
TYPE FC, PEDESTAL MOUNTED SIDE SUCTION PUMP

Manufactured to the same exacting engineering standards as the Type CC Pump, and with the same capacities and heads. The flexible coupling drive permits easy field servicing . . . choice of motor manufacture. Ask for Bulletin 107.



DUPLEX CONDO-VAC® RETURN LINE VACUUM AND BOILER FEED PUMPS

Specifically designed for vacuum heating systems. The water capacity of CONDO-VAC® pumps remains constant up to the boiling point. The air capacity remains constant up to the evaporation point. Pressures up to 40 lbs. Capacities from 2500 to 150,000 EDR. Ask for Bulletin 270.



SURE RETURN® CONDENSATION PUMP & RECEIVER

For systems up to 75,000 EDR and for low and medium pressures. Available in either single or duplex units. The base serves as a large capacity manifold through which the condensate flows by gravity from the receiver to the pump. SURE RETURN® pumps never steam bind. Ask for Bulletin 250-F.



TYPE AVC CONDENSATION PUMP & RECEIVER

Heavy cast iron receiver with low inlet for floor mounting or shallow pit. Sets on floor — no foundation bolts necessary. Weight of unit and piping hold unit firmly in place. For 500 to 10,000 EDR and 10 to 30 lbs. pressure. Ask for Bulletin 245-F.

Putting Ideas to Work

FOOD MACHINERY AND CHEMICAL CORPORATION

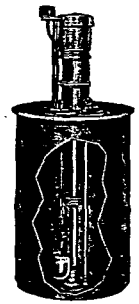
CHICAGO PUMP COMPANY

622 DIVERSEY PARKWAY • CHICAGO 14, ILLINOIS

Distributors In Principal Cities Throughout The United States and Foreign Countries

PRODUCTS

Condensation, House, Booster and Fire Pumps . . . Return Line Vacuum Heating and Boiler Feed Pumps . . . Circulating, Brine, Sewage, Sump, Sludge Pumps . . . Pneumatic and Tankless Water Supply Systems . . . Sealed Electrode Floatless Pump Controller . . . Automatic Alternator for Duplex Sets of Pumps



TYPE NVC VERTICAL CONDENSATION PUMP & RECEIVER

Specifically designed for low return service and underground installations. Flanges are accurately finished and gasketed to prevent steam leaks. Pump and receiver are shipped as a unit, ready to install. For 1000 to 40,000 EDR and 10 to 25 lbs. pressure. Ask for Bulletin 254-A.



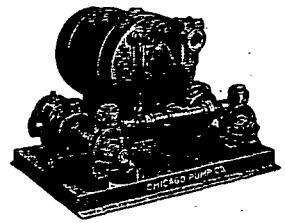
TYPE LGL2 AUTOMATIC ELECTRIC SUMP PUMP

Specifically designed to drain basements, pits and sumps. Centrifugal volute pump with open bronze impeller. A cast iron strainer of large area surrounds the suction opening. Ask for Bulletin 118-F-3.



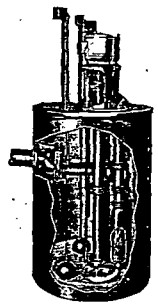
VERTICAL CLOSED SHAFT NON-CLOG PUMP

Available in either single or duplex units . . . suitable for either sump or sewage service. Pump is mounted on separate floor plate to allow easy removal for servicing. Ask for Bulletin 124-F.



TYPE FC DUPLEX CONDENSATION UNIT

Single or duplex units are specifically designed for operating conditions where pressures are high and capacities relatively small. For duties up to 50,000 EDR and discharge pressures up to 150 lbs. Ask for Bulletin 260-C.



TYPE F FLUSH KLEEN® CLOG-PROOF SEWAGE PUMP

Only liquids are handled by the impeller. Solids and other coarse matter are retained in the strainer housing until cleanly flushed to the sewer. Ask for Bulletins 122-C, 122-S.

SEWAGE AND SUMP PUMPS NOW AVAILABLE WITH sealtrode*

*a trademark of Chicago Pump Company



SEALED ELECTRODE FLOATLESS PUMP CONTROLLER . . . used exclusively with "Chicago" Sewage and Drainage Sump Pumps.

The SEALTRODE* Controller has every advantage of other floatless controllers, plus these advantages of being sealed:

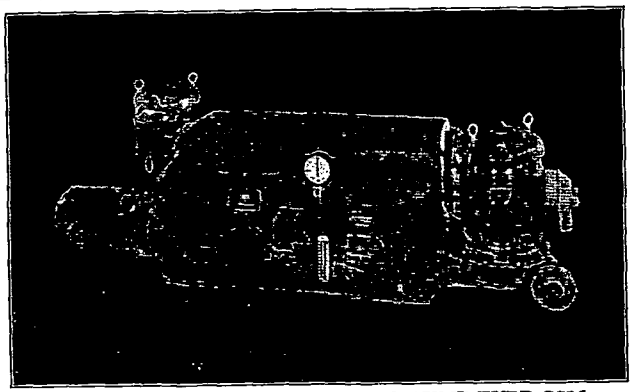
- Electrodes never become insulated and coated with grease, oil or soap.
- Electrodes are never affected by corrosive elements in sewage and drainage water.
- No moving parts in liquid.
- Low original cost . . . eliminates need for duplex float switches and float guide pipes.
- Low maintenance cost.

Ask for Bulletin 128

THE NASH ENGINEERING COMPANY

234 Wilson Road **South Norwalk, Conn., U. S. A.**

Sales and Service Offices in all Principal Cities

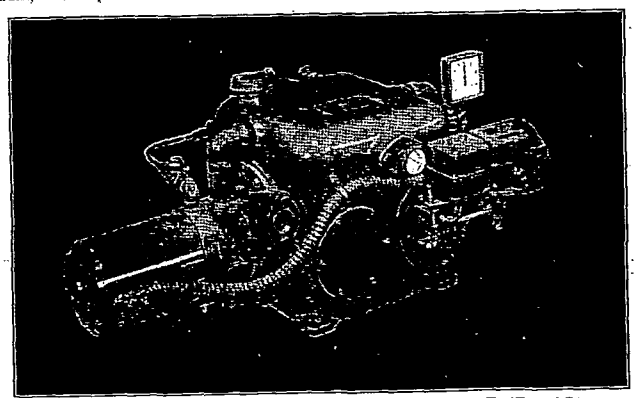


RETURN LINE VACUUM HEATING PUMP TYPE CSM

The new CSM Heating Pump achieves a high level of performance and efficiency. Complete flexibility is secured in this design, since each pumping element is driven by its own motor, actuated by its own control.

This makes possible a wide range of air and water capacities that can be selected to meet the exact requirements of each particular job.

These pumps are now available in a full range of air and water capacity combinations, for operation up to 20 inches Hg vacuum, and for pressures from 10 lb to 70 lbs. *Bulletin on request.*



RETURN LINE VACUUM HEATING PUMP TYPE CSI

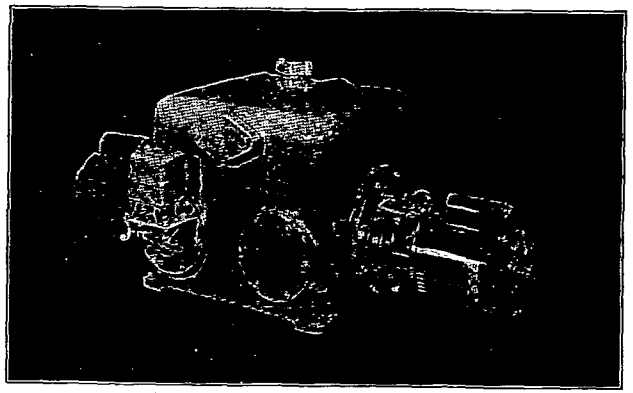
Now a Nash quality Vacuum Heating Pump can be economically installed and operated on any steam heating job. Engineered for high performance and low installed first cost, this new pump still makes use of time tested Nash principles of operation.

The Nash CSI has generous air capacity and features a wide choice of water capacities and discharge pressures. The right combination of capacities is at hand to match the requirements of the job. It is no longer necessary to pay extra for a pump with excessive water capacity, excessive discharge pressure, or in an attempt to get adequate air capacity. Capacities to 40,000 sq ft EDR. *Bulletin on request.*

THE NASH ENGINEERING COMPANY

234 Wilson Road **South Norwalk, Conn., U. S. A.**

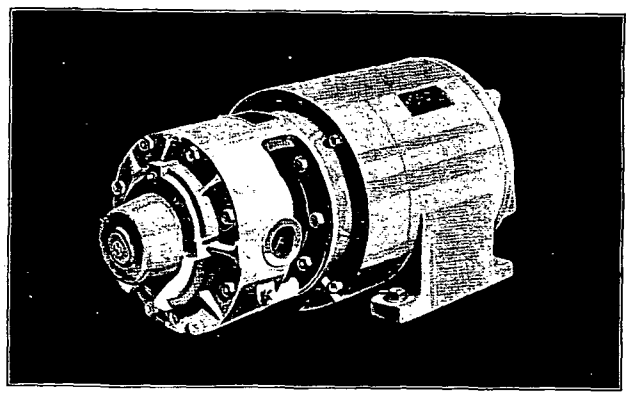
Sales and Service Offices in all Principal Cities



CONDENSATION PUMP TYPE CS

Following the advanced design of our CSM Heating Pumps, the CS Condensation Pump has features representing long life, dependable operation and low maintenance and installation cost.

Pump is single suction centrifugal in bronze fitted construction with renewable bronze sealing rings. Balanced impeller is designed for high efficiency handling hot condensate. Pump and motor form a compact unit. Simple mechanical seal replaces conventional stuffing box, eliminating maintenance. Strainer area nearly three times that in previous pumps. Easily removed for cleaning. Inlet connection is only 12, 14 or 15 inches, depending upon size of unit, permitting easy and low cost installation. *Bulletin on request.*



NASH HOSPITAL PUMPS

For vacuum and pressure systems in hospitals to provide vacuum or pressure to operating rooms, laboratories and wards. Nash pumps possess particular characteristics adapting them to this exacting service. Silent service is assured, because the pumps and equipment are installed in a mechanical equipment room. Automatic control assures constant maintenance of the correct vacuum or pressure. No pulsation is present in either vacuum or pressure produced by Nash Pumps. Air delivered by Nash Compressors is entirely free from dust, heat or oil. Even dust entering the pump inlet is effectively removed. No oil or dust filters are required. No expert attendance is required. *Bulletin on request.*

THE KRAISSL COMPANY

INCORPORATED

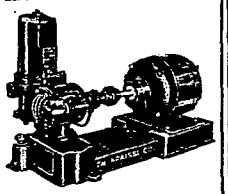
PUMPS—SEPARATORS—ENGINEERING EQUIPMENT

Main Office and Factory

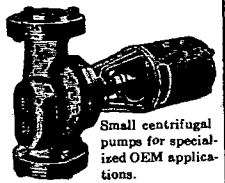
Kraissl Building, 297 Williams Ave., Hackensack, N.J.

Sales Representatives in Principal Cities

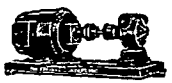
INDUSTRIAL MARINE



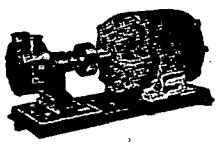
Class 25 small-size air pumps
Bulletins A-1685-1679



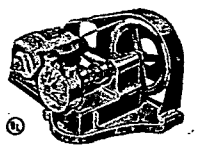
Small centrifugal pumps for specialized OEM applications.
Bulletins A-1551-1401-1344



Class 50 roller pumps
Bulletin A-1550



Class 60 direct drive pumps
Bulletin A-1904



Class 60 reduction drive pumps
Bulletin A-1102

APPLICATION

We have concentrated for a period of over 30 years on the design, development, manufacture and supply of pumps, separators and fluid handling accessories to meet the requirements of original equipment manufacturers where our products become an integral part of machines and installations. We do not believe any one design will meet all requirements, and our extended line reflects the wide variety of standardized equipment now available. If none of these items meet your needs, Kraissl Engineers will be most happy to consider modifications or design for your application.

KRAISSL AIR PUMPS

Rotary Compressors and Vacuum Pumps
Single stage; vane and roller designs
Capacities: ¼ to 170 scfm
Pressures: to 50 psig
Vacuums: to 28 in. Hg (except laboratory models to 29⅓ in. Hg)
Patented force feed system of lubrication on oil lubricated models. Dry lubricated models also available. Convection, fan, and water cooled models.

KRAISSL LIQUID PUMPS

Gear, roller and centrifugal designs; direct and reduction drives for standard and high pressures.
Designed for applications in the fuel oil and hydraulic fields. Class 60 fuel oil pumps are *UL*-listed in most commonly used sizes.
Capacities: to 210 gpm
Pressures: to 400 psig

KRAISSL SEPARATORS

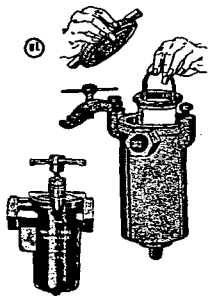
Filters and strainers in single and duplex models for liquids and steam. Listed by *Underwriter's Laboratories* in most commonly used sizes.
Sizes: ¼ in. to 6 in. std., single special to 12 in.
Body materials: C.I., C.S., Bronze, S. S.
Basket materials: Brass, Steel, Monel, S. S.
Basket openings: .0017 in. to .250 in.
Working Pressures: To 600 psig standard, special 1200 psi.

Write for General Bulletin A-1803 which summarizes our line and indexes available literature or specific bulletins describing products of your interest.

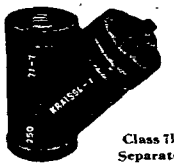
In addition to the products indicated, we currently manufacture a line of special TRANSFER VALVES, small capacity CENTRIFUGAL PUMPS, and AIR FILTERS.



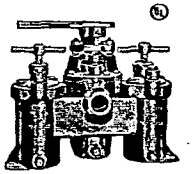
Class 60 M motorized pumps
Bulletin A-1267



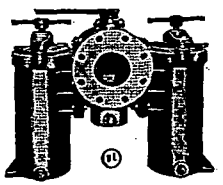
Class 72 single separators
Bulletins A-1755-1882



Class 71-Y Separators
Bulletin A-1889



Class 72 integral duplex separators
Bulletins A-1430-1698



Class 72-A 3-piece duplex separators *Bulletins A-1214-1430-1431-1879*

PEERLESS PUMP DIVISION

Food Machinery and Chemical Corporation

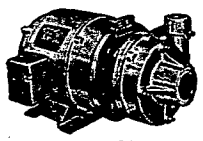
PLANTS: Los Angeles 31, Calif. 301 West Avenue 26 — Indianapolis 8, Ind. 2005 Northwestern Avenue

Offices: New York; Atlanta; Indianapolis; Chicago; St. Louis; Dallas, Plainview, Lubbock, Texas; Albuquerque, New Mexico; Phoenix, Arizona; Fresno, Los Angeles, Calif.

Products:

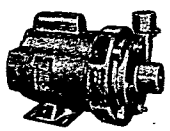
Vertical Pumps: Deep and shallow well pumps; water supply, boosting, circulating and waste disposal pumps; condensate pumps; sump pumps. All types of drive.
Horizontal Pumps: General purpose horizontal split case and end-suction pumps; turbine vane type pumps; boiler feed pumps; condensate pumps; circulating pumps; process pumps. All types of drive.

Peerless Type PE integral HP general purpose end-suction close-coupled pump



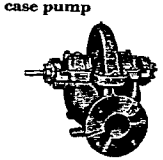
HP range: 1 to 40 HP. **Capacities:** 30 to 800 gpm. **Heads:** up to 260 ft. **Design:** single-stage, end-suction. **Drives:** standard horizontal *NEMA* motor. Screwed and flanged connections. Bronze impeller. Compact, versatile, dependable and efficient. Good hydraulic characteristics. Extremely durable and economical to operate. **Completely described in** *Bulletin B-2300.*

Peerless Type PE-A fractional HP general purpose end-suction close-coupled pump



Very good for adaptation to liquid moving equipment or systems. **HP range:** ¼ to 1½ HP at 3450 rpm. **Capacity:** 10 to 65 gpm. **Heads:** up to 110 ft. **Design:** single-stage, close-coupled, end-suction. **Drives:** ball bearing motor with stainless steel shaft. Cast iron volute, cast bronze impeller. Designed for installation at any angle; compact, flexible. **Completely described in** *Bulletin B-2300.*

Peerless Type AquaLine general purpose horizontal centrifugal split case pump



For hot and cold water handling, circulation and boosting. **Discharge sizes:** 1½ in. to 8 in. **Capacities:** up to 2600 gpm. **Heads:** up to 350 ft. **Temperature range:** up to 250°F. **Drives:** motor. The AquaLine volute is cast iron, bronze fitted, impeller is cast bronze. Available with packing gland or replaceable mechanical shaft seals. Highly dependable and economical to operate. **Fully described in** *Bulletin B-2100.*

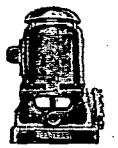
Peerless Type A general purpose horizontal centrifugal split case pump



For the widest variety of pumping services. **Capacities:** to 70,000 gpm. **Heads:** to 300 ft. Single stage, split case design. **Drives:** direct-connected electric motor; belted drives; diesel, natural gas, and gasoline engine; steam turbine; combination drives. Double-suction sizes 1½ in. through 48 in., bottom suction sizes 12 in. through 42 in. **Completely described in** *Bulletin B-1500.*

Other comprehensive, illustrated bulletins describing Peerless' complete line of vertical and horizontal pumps for all types of domestic, industrial and commercial service are immediately available. Request copies describing the type pump in which you are interested to the factory, address shown above.

Peerless deep well pump



For water supply services. **Capacities:** 15 to 30,000 gpm for installations in 4 in. wells or larger, to any practical depth. Adaptable to motor, belt, gear or steam drives. This turbine pump features several design advantages special with Peerless; the cooling method, unique bearing contact, and packing arrangements are examples. Available in water or oil lubricated line-shaft types. **Described in** *Bulletin B-141.*

Peerless Vertical Pump



For application to cooling tower service, tank pumping, line pumping, boosting and recirculation. **Capacities:** 30 to 40,000 gpm. **Heads:** to 1000 ft. **Hp range:** electric, up to 2500 hp; right angle gear drives, up to 300 hp; steam turbine, as required. **Pump column sizes:** 2½ in. to 30 in. inclusive. **Liquid temperature range:** to 250 F. Available in water lubricated open line-shaft construction and enclosed line-shaft oil lubrication types. **Described in** *Bulletin B-505.*

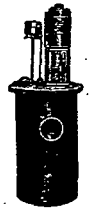
Peerless Hydro-Line encased, close-coupled vertical centrifugal pump

Especially suited to pumping hydrocarbons such as gasoline, butane, propane, hot oils, etc.; hot or cold water; mild acids, basic and salt solutions. **Capacities:** up to 3000 gpm. **Heads:** to 1000 ft. **Temperatures:** up to 400 F. **Drives:** as required, standard vertical solid shaft or explosion-proof motors, steam turbines. **Request** *Bulletin B-1700.*

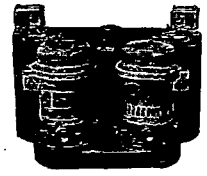
SKIDMORE CORPORATION

ST. JOSEPH, MICHIGAN

Quality Heating Pumps Since 1921



Write for Bulletin No. 19-A on type UV Pump



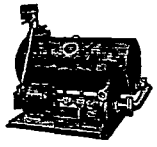
Write for Bulletin No. 21-C on type CCVS Pump

SKIDMORE TYPE UV PUMP

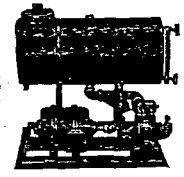
The Skidmore Type UV Condensation Pump, single or duplex unit, is designed for use where returns are too low to permit the operation of horizontal type pumps. Type UV Pump operates very satisfactorily where condensate is to be returned to the boiler or other points, from all types of steam heating systems or process equipment. Type UV Pump consists of vertical pump and motor unit with float and float switch mechanism mounted on cover plate which is bolted to sump type receiver. Capacities 2000 to 40,000 sq ft EDR and discharge pressures from 10 to 75 lb.

SKIDMORE TYPE CCVS PUMP

The Skidmore Type CCVS Condensate Pump is new close-coupled design. Built to give maximum efficiency, yet requires a minimum of space. Shown above is the duplex pump with two float switches. This model is also furnished with automatic alternator. The CCVS Pump is also built in single unit. Motor and pump are bolted to receiver. Pump is bronze fitted throughout with enclosed centrifugal impeller of special design for hot water. Capacities 500 to 10,000 sq ft EDR and pressures from 10 to 40 lb.



Write for Bulletin No. 14-B on type HS Pump



Write for Bulletin No. 17 on type TM Pump

SKIDMORE TYPE HS PUMP

The model HS Condensation pump is economically designed to offer maximum efficiency with lower operating cost. The Type HS will operate at high temperature encountered in condensation pump service. The pump can be disassembled without disturbing piping connections. Furnished in either single or duplex type. Pump casing is constructed of heavy close grain cast iron, vertically split, bronze fitted throughout. Capacities 1,000 to 125,000 sq ft EDR, pressures 10 to 75 lb.

SKIDMORE TYPE TM PUMP

Type TM pump consists of motor, steel receiver, automatic make-up valve mounted on receiver, water level gage glass mounted on receiver, self-cleaning strainer between pump and receiver, assembled and mounted on steel base. Cast iron base furnished if required. Electric boiler water level controller furnished for mounting on boiler. The Type TM Condensate and makeup water pump is of turbine type designed for boilers up to 250 hp and pressures up to 150 lb.

Skidmore Pumps are built in single and duplex units

SKIDMORE CORPORATION ST. JOSEPH, MICH. *Choose with Confidence!*

Write for complete engineering data on all Skidmore Pumps

TUTHILL PUMP COMPANY

979 East 95th Street, Chicago 19, Illinois

Representatives in all principal cities

Pumps for Your Purpose

Tuthill manufactures a complete line of positive displacement rotary pumps for transfer oil burner, lubrication, and other services. Capacities from ⅓ to 200 gpm, for pressures to 500 psi, speeds to 3600 rpm. Also, pump units and special models for built-in applications. Write for catalog describing complete line.

SU Belt-Driven Pump Unit. For handling heavy fuel oil in commercial oil burner applications or with packaged boilers. Internal-gear pump design and 2 to 4 "V" belt drive insure quiet operation. Economically priced, it includes quality construction features such as two ball-bearing pillow blocks, one point take-up which simplifies adjustment of belt tension and special fabricated steel base with versatile motor mounting.



SU pump units are compact—can be provided with overall dimensions of only 14⅜ in. x 25¼ in. Furnished with a wide variety of pump and motor combinations, for pressures up to 300 psi, capacities to 2030 gallons per hour. Sub-assemblies of pump, stub shaft, pillow blocks and sub-base can be provided for incorporation into larger equipment. Ask for *Catalog 107*.

Model L—Small Industrial Pumps. *Underwriters' Laboratory* approved heavy oil pump series widely used for oil burner service, lubrication, liquid transfer and general industrial use. Model L is a compact, highly efficient, positive displacement, internal-gear rotary pump with mechanical-seal. It is provided for capacities from 20 to 360 gallons per hour at pressures up to 500 psi. Available with a variety of flange or foot mountings. Model L is directional in rotation and designed for direct drive at standard motor speeds. Ask for *Catalog 101*.



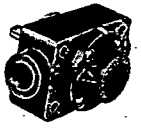
Model R—Automatic Reversing Pumps. Model R incorporates the Tuthill automatic reversing feature—designed to maintain constant direction of flow from the same port regardless of the direction of rotation of the driving shaft, without the use of check valves. Available in 11 sizes ranging in capacities from ⅓ to 200 gpm, for pressures up to 400 psi. Variations of models are available with mechanical shaft seal, internal-relieving feature and various mountings. Write for *Catalog 106*.



Stripped Pumps for Built-in Applications. For incorporation into products of manufacturers, Tuthill pumps can be furnished with the mounting bracket removed (Series S), or with mounting bracket and housing removed—providing pumping elements only (Series SA). Stripped pump units are furnished in 10 sizes in capacities from ⅓ to 200 gpm for pressures up to 200 psi. The series includes nonreversing and automatic reversing type pumps as well as units with automatic internal-relieving feature. Ask for *Catalog 106*.



Cartridge-Type Lubrication Units. These compact, cartridge-type lubrication units are provided in 4 different models, each in 3 sizes, with a capacity range of from 55 to 170 gallons per hour, for pressures up to 200 psi. They may be furnished for non-directional service or with the automatic reversing feature where the pump must be driven by a reversing shaft. Choice of internal or external porting. Designed for flange mounting. Ask for *Catalog 103*.



Integral Motor—Pump Units. Tuthill provides a complete range of close coupled pump and motor combinations for use where space and weight are primary considerations. Units such as illustrated can normally reduce space required for pump and motor by at least six inches and cut weight by several pounds. Capacities range from 20 to 300 gph. Pumps are available for pressures to 500 psi.

Larger pump and motor combinations can be provided with capacities to 50 gpm, for pressures to 400 psi.

For original equipment applications involving substantial quantities Tuthill can develop a special POWERMITE combination where pump and motor are incorporated into one unit which takes up no more space than a standard electric motor (a typical POWERMITE measures only 3⅞ in. x 7½ in.).



Water Pumps. Tuthill provides vane-type water pumps for capacities up to 75 gph at pressures up to 150 psi. These are particularly adapted for applications involving boiler make-up water, or other boiler feed applications.

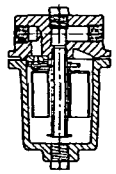
Catalog No. 100 Describing Complete Tuthill Line Also Available on Request.

The V. D. Anderson Company

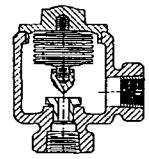
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1960 West 96th Street, Cleveland 2, Ohio

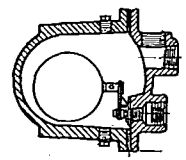
Representatives and jobbers in all principal cities
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Quality Steam Specialties Since 1886

FOUR TYPES OF STEAM TRAPS
A Trap for every purpose


Super-Silvertop Inverted Bucket

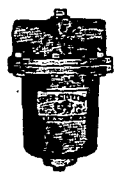

Heat-Kwik Combination


Quik-Flex Thermostatic


Anderson Float

SUPER-SILVER TOP Inverted Bucket Steam Traps

Application—The inverted bucket trap is recognized as the best trap for most inside installations, as it is actuated by a liquid or gas rather than temperature. Available standard or with air eliminator (sometimes called blast type trap) where it is necessary to remove air quickly.



Description—Hooks up as an elbow or straight-in-line—only one nipple needed. Head and body of cast semi-steel; valve and seat of Anderloy; stainless steel mechanism and copper bucket (stainless steel buckets available).

Precision Parts Alignment—The bucket does not swing free. It is guided on a hexagonal tube. Keeps all parts in proper alignment and prevents bucket damage. Positive closing of the valve is insured.

SPECIFICATIONS

Size No. Trap	110	118	119	120	121	122	123	124	125
Size Connections (See Note)	½" only	½" or ¾"	½" or ¾"	½" or ¾"	½" or ¾"	¾" or 1"	1" or 1¼"	1¼" or 1½"	1½" or 2"
Max. Ga. Pressure	100	150	250	250	250	350	250	250	250
List Price	$10.00	$9.00	$10.00	$12.00	$16.00	$25.00	$38.00	$48.00	$80.00
WITH THERMAL AIR ELIMINATOR									
List Price	$11.60	$10.60	$11.60	$13.60	$17.60	$28.20	$39.20	$53.80	$86.40

Differential Pressure	CAPACITY IN POUNDS OF WATER PER HOUR								
10	435	505	795	1190	2480	3550	7100	9200	29650
50	453	650	900	1400	2490	4180	8300	13500	27400
100	440	455	675	1200	2330	3740	8500	14000	25300
150	....	450	580	1000	2200	4400	5500	12500	20500
250	....	....	....	....	....	3800	7800	11500	13750

*Capacities based on continuous flow. When ordering be sure to specify maximum steam pressure. *Note:* Pipe sizes shown in heavy type are standard and traps will be shipped tapped standard unless otherwise specified. Pipe sizes shown in light type furnished at no additional cost but only from Cleveland stock.

Write for *Bulletin* 657.

HEAT-KWIK Combination STEAM TRAPS



Application—For fast air and condensate elimination, even at extremely low pressures. This feature is an advantage in systems requiring fast heat such as kettles, autoclaves, dryers, cookers, etc.

Description—Traps have separate orifices, one for condensate and a separate bellows orifice for air. Head and body of cast semi-steel; valve and seat of Anderloy; stainless steel mechanism; copper bucket; bronze bellows for BL series for 50 lb service and monel bellows for BH series for 150 lb service.

Write for *Bulletin* 23.

SPECIFICATIONS

Size No. Trap	119BL	119BH	120BL	120BH	121BL	121BH	122BL	122BH	122BL	122BH	124BL	124BH	125BL	125BH
Size Conn.	¾"	¾"	¾"	¾"	¾"	¾"	1"	1"	1¼"	1¼"	1¼"	1½"	2"	2"
Max. Gage Press	50	150	50	150	50	150	50	150	50	150	50	150	50	150
List Price	$14.00	$22.00	$8.00	$4.00	$7.00	$8.00	$1.00	$7.00	$3.00	$3.00	$4.00	$0.00	$8.00	$04.00

DIFF. PRESS	CAPACITY IN POUNDS OF WATER PER HOUR												
10	1584	1584	2088	2088	3336	3336	4728	4728	7008	7008	11220	11220	23410	23410
50	2751	2751	3521	3521	4321	4321	6011	6011	10631	10631	19400	19400	42200	42200
100		3143		3663	4843	4843	6356	6356	11968	11968		23100	.....	53400
150		3434		3904		5134		7304		11494		22200	.....	50250

QUIK-FLEX Thermostatic STEAM TRAPS



Application—Designed primarily for outdoor installation where a freezeproof trap is required; such as tracer lines, meter boxes, drips, etc., found in chemical, refinery, solvent and other industrial plants. Air and condensate is automatically discharged.

Description—A bellows actuated trap for fast air removal and condensate drainage for pressures up to 250 lb. Furnished with either bronze or monel bellows. Body and head of brass and cast semi-steel (see below) Valve and seat of Anderloy, a tough chrome nickel alloy. Guaranteed against freezing.

SPECIFICATIONS

Size Number of Trap	B-1	B-2	C-1	C-2	C-3	C-4
Size Pipe Connections	½	¾	¾	¾	1	1¼
Maximum Gauge Pressure	150	150	250	250	250	250
Body Material	Bronze	Bronze	Semi-Steel	Semi-Steel	Semi-Steel	Semi-Steel
List Price (Monel Bellows)	$21.00	$21.00	$33.00	$33.00	$48.00	$48.00
List Price (Bronze Bellows)	$21.00	$21.00	$30.00	$30.00	$45.00	$45.00

Differential Pressure	CAPACITIES IN POUNDS OF CONDENSATE PER HOUR		
10	2,760	5,200	7,500
50	3,900	11,400	15,600
100	5,100	15,300	21,000
150	9,700	19,000	25,000
250	.....	23,000	31,000

Write for *Bulletin* 257

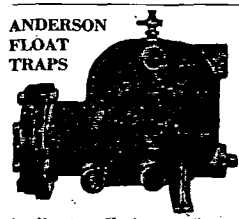
Write for our 36-page engineering catalog "How To Select Steam Traps and Other Fluid Specialties"

The V. D. Anderson Company

Division of International Basic Economy Corp.
1960 West 96th Street, Cleveland 2, Ohio

Representatives and jobbers in all principal cities
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Quality Steam Specialties Since 1886

ANDERSON FLOAT TRAPS



SPECIFICATIONS

Size Number of Trap	81	82	83	84
Size Pipe Connections	½"-¾"	½"-¾"	1"-1¼"	1½"-2"
Maximum Gauge Pressure	250	250	250	250
List Price (standard)	$17.00	—	—	—
(stainless steel float)	$27.00	$23.00	$ 91.00	$170.00
(with gage glass)	—	75.00	105.00	185.00

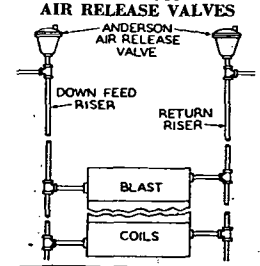
Application—Use in steam lines or systems where traps must discharge continuously. (Inverted bucket traps discharge intermittently.) Use to drain moisture and entrainment from air and gas pipelines, purifiers, separators and gas equipment.

Description—Anderson Float Traps are continuous flow type desirable for draining moisture from air or gas equipment and ahead of condensate meters where intermittently discharging traps cannot be used. Traps are of rugged construction for heavy duty service. The head and body are cast semi-steel, float stainless steel and working parts bronze and stainless steel.

Differential Pressure	CAPACITY IN POUNDS OF WATER PER HOUR				
20	2200	4400	18000	24600
50	1580	3850	11000	27500
100	1200	4100	10700	25800
150	1000	3100	7800	20200
250	600	1900	5850	13600

Write for *Bulletin* 31.

ANDERSON AIR RELEASE VALVES



Application—Designed to eliminate air and gases from pipelines in applications such as domestic hot and cold water systems, hot water heating systems, chilled water coolers, air conditioning systems, filters and any individual pipe-line or equipment carrying liquid.

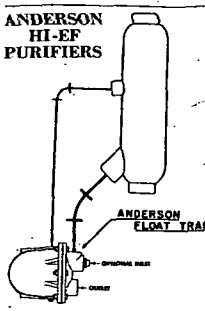
Description—Anderson Air Release Valves are automatic and release air from the lines as fast as it accumulates. They are built to withstand pressures up to 500 lbs. Install at the highest point in all your liquid systems.

Size Number of Valve	70	72	73	13
Size Pipe Connections (inlet)	½"	½" or ¾"	1"-1¼" -1½"	½" or ¾"
Size Pipe Connections (outlet)	½"	½"	¾"	½" or ¾"
Maximum Gauge Pressure	150	250	250	500
List Price	$12.00	$22.00	$48.00	$90.00

Operating Pressure	CAPACITY—CU. FT. FREE AIR PER MINUTE			
10	7.0	20.0	52.0	
50	9.3	22.5	77.0	
100	12.2	30.0	77.5	
150	14.7	33.3	90.0	
250		38.0	89.0	
400				24
500				28

Write for *Bulletin* 856.

ANDERSON HI-EF PURIFIERS



* Type LC Small Line Purifiers

Type	LC-50	LC-75	LC-100	LC-125	LC-150	LC-200
Size	¼	¾	1	1¼	1½	2
Diameter	2¾	3¾	3¾	3¾	5	5
Length	9	9	15	15	17	17
Vent Size	¼	¼	¾	¾	¾	¾
Drain Size	½	½	¾	¾	¾	¾

Type L*150, *300 & 600 PSIG Large Line Purifiers

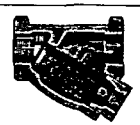
Type	L-250	L-3	L-4	L-5	L-6	L-8	L-10
Size	2½	3	4	5	6	8	10
Diameter	5¾	8¾	10¾	12¾	14	18	20
Length	18	22	26	30	34	44	52
	22	25	30	35	39	49	60
Vent Size	½	¼	¼	½	½	¾	¾
Drain Size	1	1¼	1½	3	3	3	3½

All above dimensions in inches. Bold face figures for 600 PSIG Line Purifiers. Write for Bulletin 501 for sizes ½ in. through 2 in. Bulletin 201 for sizes 2½ in. and larger.
* Available from stock.

Application—On steam lines and in boilers for clean, dry steam. Type L ahead of steam engines and turbines protect against slugs, line scale, moisture, salts and dirt. Type LC ahead of smaller steam air equipment such as heat exchangers, dryers, drums, etc. Internal type in boilers to deliver clean dry steam. For cleaning and drying air and other gases.

Description—Type L (flanged) for sizes 2½ in. and larger. Type LC (threaded) for sizes ½ in. through 2 in. Units are installed as a compact unit in pipeline adjacent to the equipment they protect. Low and high pressure types available, many from stock. Drained automatically by Float Trap (air or gas) or Inverted Bucket Trap (steam).

ANDERSON STRAINERS



SPECIFICATIONS AND PRICES

Pipe Size	¼"	⅜"	½"	¾"	1"	1¼"	1½"	2"	2½"	3"
Overall Length	3¼"	3¼"	3⅝"	3¾"	4⅝"	5½"	6"	7¾"	8¾"	10"
Blowdown Connection Pipe Size	½"	½"	½"	¾"	1"	1¼"	1½"	1½"	1¼"	1¼"
Shipping Weight (lbs.)	1	1	1¼	2¼	3¾	5½	9¼	16¼	25½	40
List Price (Stainless Steel Screen)	$3.00	$2.50	$2.75	$3.25	$4.00	$5.00	$7.25	$11.00	$28.00	$36.00
List Price (Brass Screen)	.....	.....	.....	.....	.....	.....	.....	.....	$22.00	$27.00

Write for *Bulletin* 252.

Application—Anderson Self-Cleaning Strainers remove scale, grit and sediment from the line ahead of steam traps, reducing valves, air tools, pumps, etc.

Description—Stainless steel screens, (brass available on 2½ in. and 3 in.) standard equipment, have extremely large free area. Screen for steam and gases has 1/64 in. openings—for liquids 1/32 in. openings. Other sizes furnished when specified, prices on application.

Dunham-Bush, Inc.

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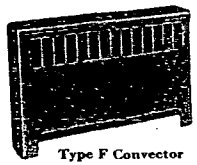
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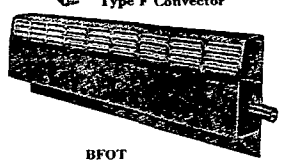
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CONVECTORS—Attractive, *easy-to-install*. Use with hot water or steam. Eight models, capacities with steam 1 lb entering air 65F: 9 to 139 EDR. Dimensions: 4, 6, 8 and 10 in. wide; 18 to 112 in. long; 12 in. window convector, 20, 24, 26, 32, and 38 in. high. Picture window, extra long, plaster front and institutional types also available.



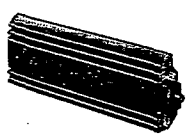
Type F Convector

TYPE BFOT—High output baseboard with "snap in" features for even greater ease of assembly. Same damper choice and heating element selection as BE-10. All accessories tailored to close tolerances enhancing beauty and simplifying installation.

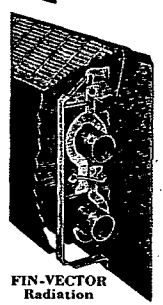


BFOT

BE-10 BASEBOARD—Snap-on front baseboard for forced hot water, steam or vapor heating. Fits flush with floor. 10 in. high. Shipped in 2-ft to 10-ft lengths for easy, on-the-job cutting. Steel or copper elements shipped in 1- to 6-ft lengths. Semi-recessed and flush-back enclosures. 12 in. baseboard also available for commercial applications.



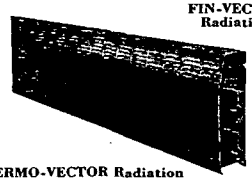
Baseboard Radiation



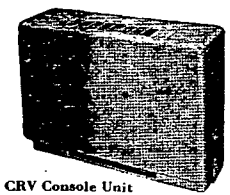
FIN-VECTOR Radiation

FIN-VECTOR®—Can be installed at any level above 4 in. from floor. Three style covers: sloping top, flat top, expanded metal. Adjustable hook-and-link hanger or new ball bearing bracket supports elements. One, two, or three tiers high. Two tube sizes, 1¼ in. and 2 in. steel or 1¼ in. copper tube. Self-spacing fins.

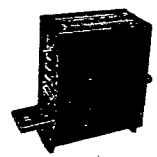
THERMO VECTOR® RADIATION—For along-the-wall heating of educational, commercial or institutional buildings with steam or hot water. Flush-type model, one, two or three tiers high. Two tube sizes: 1¼ in. and 2 in. Lever- or knob-operated damper.



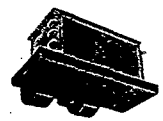
THERMO-VECTOR Radiation



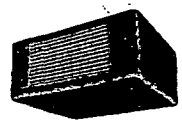
CRV Console Unit



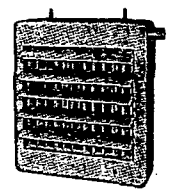
CRV Basic Unit



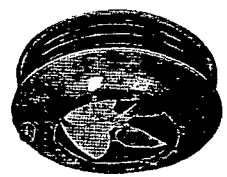
CRH Basic Unit



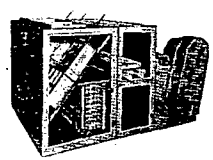
CRH Unit with Cabinet



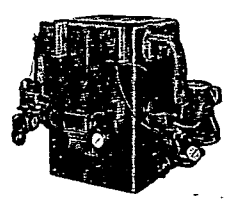
Horizontal Unit Heater



Vertical Unit Heater



Multi-Zone Unit



Model C2 Vacuum Pump

HEATING-COOLING UNITS—For individual cooling with chilled water or heating with hot water; for individual cooling with direct expansion or heating with steam; cooling and ventilating; or heating and ventilating. Four sizes of vertical floor mounting type. Five sizes of horizontal ceiling mounted type. Can be exposed or recessed.

UNIT HEATERS—Type M, Model H and HM, not shown. Cabinet type. Seven styles, 3 sizes. Blower fans directly mounted on double end shaft motor. Capacities at 2 lb steam, 60 F ent. air: 18,000 to 108,000 Btu.

UNIT HEATERS—Type H. Horizontal discharge. 11 sizes, 2 models. Capacities at 2 lb steam, 60 F ent. air: 18,600 to 285,800 Btu per hour.

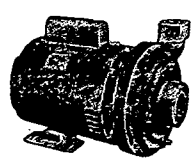
UNIT HEATERS—Type C. Vertical discharge. Four types of diffusers available. Ten sizes, 2 models. Capacities at 2 lb steam, 60° F ent. air: 45,400 to 598,000 Btu.

MULTI-ZONE UNITS are available in 9 sizes from 2560 to 28,800 cfm. They provide for simultaneous air conditioning of separate rooms or zones at desired temperatures. They make possible individualized conditioning of such areas with a single unit instead of several units.

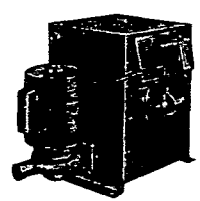
VACUUM PUMPS—Pull and maintain, at low amperage, up to 26 in. vacuum without dependence on close clearances. Only ONE principal moving part. Single motor easily handles BOTH air and condensate removal. Low 9⅛-in. return line connections. Capacities: 2,500 to 65,000 EDR. Single or duplex units.

CENTRIFUGAL PUMPS—Volute type for horizontal or vertical installation. Thirteen sizes ranging from ⅓ hp to 2 hp, capacities to 150 gpm. Typical supply system applications include boiler feed, processing, drainage, washer service and booster service. Also used as circulators in evaporative condensers, cooling towers, hot water and chilled water systems etc. For applications involving fluids other than water (glycols, brines, light oils) consult factory for recommendation.

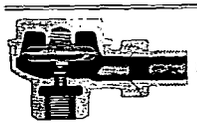
CONDENSATION PUMPS—Pumps feature mechanical seal that eliminates stuffing box worries. Capacities: 2,000 to 50,000 EDR. Single and duplex units.



A5 Centrifugal Pump



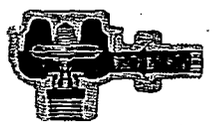
Type CHV Condensation Pump



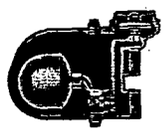
No. 1E Radiator Trap

RADIATOR TRAPS—For all types of low pressure steam heating systems. Body and cover of cast bronze, disc of non-corrosive metal. Available in 6 patterns. Capacities from 200 to 700 EDR.

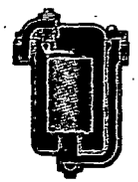
Type TH1-A Thermostatic Trap



THERMOSTATIC TRAPS—Trap has cast brass body and cover; replaceable stainless steel valve and seat; disc of non-corrosive metal. For 5 to 100 lb working pressures. Capacities from 125 to 4,500 lb of condensate per hour.



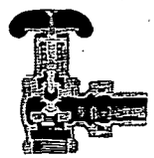
Series 30 Float and Thermostatic Traps



Types OB and OBS Bucket Traps

FLOAT AND THERMOSTATIC TRAPS—Use as drip trap for steam mains, unit heaters, heat exchangers. Float, copper; float valve and seat, Monel metal. Up to 25 psi gauge working pressure. Sizes: ¾, 1, 1¼, 1½ and 2 in. with capacities from 100 to 5,750 lb of condensate per hour.

INVERTED BUCKET TRAP—Body and cover of high tensile iron castings. Valve and seat (renewable and interchangeable) are specially hardened, corrosion-resistant steel. Operating pressures 20 to 150 lb gauge. Sizes ½, ¾, 1 and 1¼ in. Capacities from 220 to 5,600 lb of condensate per hour.



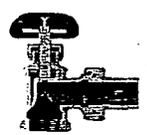
No. 740 Radiator Valve



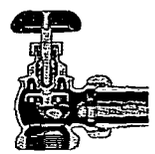
Type 1140 Radiator Valve

SPRING PACKED VALVES—For low-pressure systems up to 25 lb. Heavy coil spring keeps packing tight around valve stem. Body and bonnet, cast brass. Sizes: ½ to 2 in. in 4 body patterns.

PACKLESS VALVES—Packless construction (no packing spring or stuffing box) provides tight permanent seal. Sizes ½ to 1½ in. in 4 body patterns.



No. 400 Radiator Valve



No. 600 Radiator Valve

HOT WATER VALVES—For hot water heating systems with pressure to 125 lb. Cast brass body. Packing nut, one-piece asbestos ring provides tight stem seal: Sizes ½ to 2 in.

PACKED STEM VALVES—For low-pressure steam systems and hot water systems up to 125 lb. Cast brass body and bonnet. Sizes: ½ to 2 in.

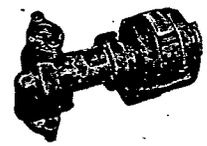


Vari-Vac Panel

DUNHAM-BUSH HEATING SYSTEMS

VARI-VAC® DIFFERENTIAL VARI-VAC HEATING saves up to 40 per cent on fuel because VARI-VAC automatically provides the exact amount of heat needed by using a continuous flow of sub-atmospheric steam at temperatures that vary with the weather.

METRO® single-riser method of piping, costs less to install and maintain. Provides continuous path for flow of steam from top to bottom of building. Continuous pipe runs down through overlying rooms and is offset into each room into convectors or baseboard radiation.

























DUNHAM-BUSH CIRCULATORS—Standard and high head capacity models for all water heating or cooling systems. Single spring motor coupling. Special brand name motor with built-in overload protection. Extra-large oil reservoir. Field replaceable seals.

FLOW CONTROL VALVES—Completely automatic. No adjustment for winter-summer operation. Easy to clean without breaking pipe connections or removing valve mechanism. Universal pattern: 1 through 1½ in. incl. Horizontal patterns: 2 to 3 in. incl.

AIR SEPARATORS—Automatically remove air from forced hot water heating system. One-piece construction. **Positively prevents re-entry of air** into system. 30 psi working pressure. Sizes: ¾ to 2½ in. incl.

FLOW DIVERTOR FITTINGS—Assure proper water flow through **all radiation.** Only one fitting normally needed (on return) for each unit of radiation. Size (thread to thread): 1 in. x ½ in. to 2 in. x 1 in. incl.; (sweat to sweat): ¾ in. x ½ in. to 1 in. x ¾ in. incl.

RELIEF AND REDUCING VALVES—Pressure Relief Valves have extra-large diaphragm area, non-stick design. Factory set at 30 lb. **Pressure Reducing Valves** have built-in strainer. Factory set 12 lb. Adjustable. **Combination Relief and Reducing Valves** available.

CIRCULATOR VALVES—Pat. 2,817,489. Extra-large flow area reduces water resistance and circulator load. T-type handle operates in only ¼ turn. Straight-way and angle patterns; thread or sweat; full range of sizes; 125 psi working pressure.

BALANCING ELBOWS AND FITTINGS—Extra-large flow area. Unique sloping seat provides V-shaped opening for accurate balancing from full open to any degree of closure. Thread or sweat. Complete line includes vent tees, union elbows, connectors.

AUTOMATIC AIR VENTS—Expansion Type Vents need no adjustment. Manual venting for rapid air removal when system is first filled. 50 lb working pressure. **Float Type Vent** has built-in siphon to prevent water-logging. 35 lb working pressure.

VARI-FLOW CONTROLS—Fully automatic indoor-outdoor temperature controls for all types of hot water heating systems. No seasonal adjustment needed. Immersion bulb and capillary "wrap-around" types.

DISTRIBUTOR HEADS—For radiant panel and "spaghetti" type hot water heating. Same 1¼ in. copper head usable for supply or return, both ½ and ⅜ in. tube connections. Each length has 14 pairs of connections.

DUNHAM-BUSH, Inc., manufactures a complete line of air conditioning, refrigeration, heating products and accessories. For catalogs and literature write: Dept. HVAC-59, Dunham-Bush, Inc., West Hartford 10, Conn., or see your local Dunham-Bush Sales Engineer.

HOFFMAN SPECIALTY MFG. CORP.

GENERAL OFFICE AND FACTORY • INDIANAPOLIS, INDIANA

Sales Representatives in Principal Cities

ENGINEERED VENT VALVES for STEAM and HOT WATER SYSTEMS


1A

VALVE NO.	CONN. (IN.)	RADIATOR—STEAM
1A	⅛	Adjustable for proportional venting.
2A	⅛	Vacuum—Adjustable for proportional venting.
70A	⅛	Meets Federal Spec. WW-V-151 for Type I Valve.
1000	⅛	Large Port—Extra large venting capacity.
40	⅛	Single Port—For small *one pipe systems.*
500	⅛	Disc type—For small one pipe systems.


71A

VALVE NO.	CONN. (IN.)	CONVECTOR—STEAM
1B	¼	Adjustable for proportional venting.
71A, 71B, 71C	⅛, ¼, ½x¾	Single Port—Vents the air, seals against the escape of water and steam in modern convectors that have high condensating rates.


75

VALVE NO.	CONN. (IN.)	MAIN—STEAM
4A	½x¾	Single Port—For small systems.
16A	½x¾	Single Port—Vacuum—small systems.
75, 75A 75H	½x¾	Single Port—For medium and large systems.
76, 76A	½x¾	Single Port—Vacuum—For medium and large systems.


79

VALVE NO.	CONN. (IN.)	WATER VENTS
500	⅛	Disc type—For radiators.
790	⅛	Single Port—30 psi—For convectors.
791	¼	Single Port—50 psi—For convectors and mains.
79	½x¾	*Single* Port—75 psi—For mains.
78	½x¾	Single Port—150 psi—For mains and process applications.


78

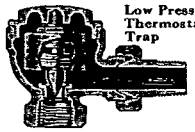
SPECIAL VENTS

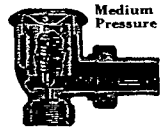
VALVE NO.	CONN. (IN.)	*UNIT HEATER STEAM VENT*
74	½x¾	Single Port—Vents with rising, steady, or falling pressure.

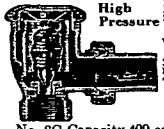

74

		STRAIGHT SHANK VENTS
41, 43, 45	⅛, ¼, ½x¾	*Single Port*—For small steam systems.
		THERMOSTATIC VENT
4	½x¾	For use where water is not encountered.

LOW, MEDIUM AND HIGH PRESSURE THERMOSTATIC TRAPS


Low Pressure Thermostatic Trap


Medium Pressure


High Pressure

Low pressure traps have brass bodies, caps and union nut and tailpiece. 17C is made in Angle, Swivel and Vertical patterns. 8C is made in Angle and Straightway patterns. 9C is made in Angle pattern only. Thermostat and seat both renewable.

No. 17C Capacity 200 sq ft EDR 25 lb pressure ½ in. connection

No. 8C Capacity 400 sq ft EDR 25 lb pressure ¾ in. connection

No. 9C Capacity 700 sq ft EDR 25 lb pressure 1 in. connection

Medium Pressure Nos. 8 & 9 and High Pressure Nos. 8H & 9H have all bronze bodies and caps with union nut and tailpiece. Thermostats are 6 diaphragms of special non-corrosive metal. Thermostats and seats are renewable. ½ in. sizes are furnished in Angle, R.H., L.H. and Straightway patterns, others in Angle only. Medium Press. 50 lb. limit. High Press. 125 lb.

F & T TRAPS, DIRT STRAINERS AND SUPPLY VALVES


F & T Trap

Y-Strainer

Supply Valve

50 Series Float and Therm. Traps are available in pressure ranges to 125 lb. Used for venting and draining risers, steam mains, unit heaters, blast coils, etc. 50 Series Traps are made for easy servicing with all working parts mounted on cover. Remove four bolts to expose all parts. Pipe sizes are from ¾ in. to 2 in. cap. from 70 to 12,000 lb condensate per hour.

Radiator Supply Valves are made in sizes from ½ to 2 in. in Angle, R.H., L.H., and Straightway patterns. Brass bodies, union nut and tailpiece. Nos. 180 and 185 have non-rising handles. Both are packless. No. 186 is especially suited to vacuum systems.

Hoffman Y-Strainers are self-cleaning. Available for 125 lb, 250 lb and 400 lb pressures. Semi-steel and Brass bodies. Should be used in line ahead of all traps, valves, and pumps. Available with brass and monel strainers.

HOFFMAN INVERTED BUCKET TRAPS


Series 600

Working Pressures to 250 lb

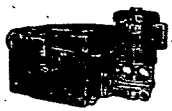
Simple mechanism assures high operating efficiency. *Features include*—straight-through pipe connection; simple seat adjustment; all working parts connected to bonnet and easily removed with it; stainless steel seat and pin. Series 600 AS has integral stainer.

HOFFMAN SPECIALTY MFG. CORP.

GENERAL OFFICE AND FACTORY • INDIANAPOLIS, INDIANA

Sales Representatives in Principal Cities

WATCHMAN CONDENSATION PUMPS


Condensation Pump

A standard unit, single or duplex, for systems up to 12,000 sq ft EDR and 20 lb discharge pressure. A low cost, vertical type condensation pump with cast iron receiver. Pump is bronze fitted with stainless steel shaft, one-piece bronze impeller. Duplex units have two pumps, two motors, and two float switches.

CONDENSATION AND VACUUM PUMPS

Hoffman pumps are available in varying capacities, a-c and d-c current, single, two, or three phase, and in pressures up to 60 lb.

Condensation Pump Single and Duplex Units Capacities from 500 to 150,000 sq ft EDR



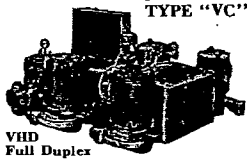

Vacuum Pumps Single and Duplex Units Capacities from 2,500 to 100,000 sq ft EDR

TYPE "UH" UNDERGROUND PUMPS


"UHS" Single Unit

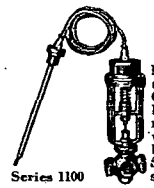
Single or Duplex units for systems up to 50,000 sq ft EDR. Units are equipped with single stage vertical type centrifugal pump with cast iron receiver. Pump has enclosed type bronze impeller and stainless steel shaft. Special pits are not necessary with the Hoffman Underground Pump.

TYPE "VC" VACUUM PUMPS


VHD Full Duplex

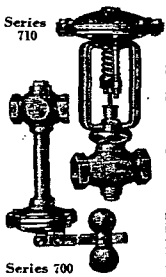
Individually sized and separately controlled vacuum and condensate pumps provide unusual flexibility of arrangement and sizing. A wide selection of vacuum pumps and condensation pumps is interchangeable on a *group of cast iron receivers* so that an ideal assembly can be specified which will best serve the requirements of any particular application. Air capacities up to 70 cfm at 20 in. Hg vacuum. Condensation pump capacities up to 150 gpm and up to 75 psi. Duplex units provide double capacity.

HOFFMAN TEMPERATURE REGULATORS


Series 1100

Series 1100—for steam pressure to 150 lb per sq. in. Standard temperature range, available between 80 F to 250 F. Other ranges available. Self-motivated. Hydraulically formed bellows of selected material. Rugged construction. For water heaters, convectors, fuel oil preheaters and other similar applications. *Single and double seated. Bronze and* stainless steel trim available.

HOFFMAN PRESSURE REDUCING VALVES


Series 710

Series 700

Series 700 Max. Press. 200 psi., Low Press. 1 to 15 psi.

Series 710 Max. Press. 250 psi., Low Press. 5 to 80 psi.

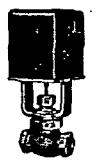
Series 715 Max. Press. 250 psi., Low Press. 2½ to 125 psi.

Series 720-S & 720-W Max. Press. 250 psi., Low Press. 2 to 125 psi.

Series 750 Max. Press. 250 psi., Low Press. 2½ to 25 psi.

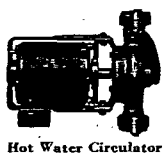
Choice of types specifically engineered for your needs—continuous service, or, for tight sealing against low pressure side when there is no demand—and with typical Hoffman features of design.

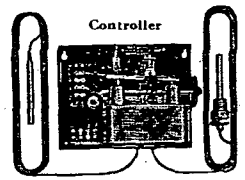
"ZONET" ZONE CONTROL SYSTEM


Control Valve


No. 193 Circulator Valve

"PANELMATIC" HOT WATER SYSTEM CONTROLS


Hot Water Circulator


Controller


No. 201 Vent Tee


No. 155 Flow Control Valve


Control Valve with Manual Control


No. 250 Expansion Joint

Maid-O'-Mist, Inc.

3217 No. Pulaski Road
Chicago 41, Ill.
Manufacturers of
Heating and Air
Conditioning Specialties

Products: Automatic air valves for (hydronic) hot water heating and cooling systems; automatic humidifiers for steam or hot water; automatic humidifiers for warm air furnaces; steam boiler water line controls; water line float control valves; liquid gas and air strainers.

AUTO-VENT®

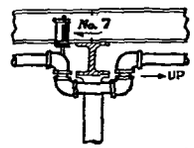
Air Eliminators

Maid-O'-Mist Auto-Vents permit automatic venting of the air which causes circulation trouble in heating and cooling systems. More and more, contractors who value satisfied customers are installing these trouble stoppers.

No. 7 Auto-Vent. For mains, pipe lines, unit heaters, chillers, convectors, coils, etc. Designed for vertical mounting only. Size 4⅜ in. x 2¼ in. with ⅛ in. IP female connection. Made of brass and equipped with a self-closing, float operated valve. All working parts, including valve and copper float, mounted on a removable bonnet for quick servicing or replacement. Valve is equipped with a Monel metal spring and a Neoprene valve seat which is unaffected by high temperatures, oil, antifreeze, etc. For pressure not exceeding 75 lb.



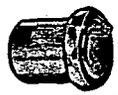
No. 7 Auto-Vent



Venting trapped mains and circulating lines

No. 77 Auto-Vent. Identical to No. 7 Auto-Vent, except for an additional ⅛ in., IP side opening to permit its use on overhead pipe lines, coils, etc., where head room is factor. Ideally suited fo Diesel engine and cooling manifold use, where vibration demands secure mounting. No air chamber required.

Auxiliary Equipment: No. 7X—bonnet assembly for No. 7 and 77; No. 777—self-closing valve core; No. 7A—connector for safe waste.



No. 27 Auto-Vent For Horizontal Mounting Only with ⅜ in. IP female side connection —Size 3 in. x 2⅛ in.

No. 27 and 37

Auto-Vents



No. 37 Auto-Vent For Horizontal Mounting Only with ⅜ in. IP vertical male bottom connection—Size 3 in. x 2⅛ in.

For convectors or baseboard radiation;—high points of overhead Mains; Radiant Panels; small Unit Heaters. Construction and internal working parts are same as in No. 7 Auto-Vent. Made of non-ferrous metals and designed for pressure not over 50 lb. No air chamber required. Auxiliary Equipment: No. 27X—bonnet assembly for Nos. 27 and 37; No. 7A—connector for safe waste. No. 666 valve core.

Write for price sheet and descriptive literature.



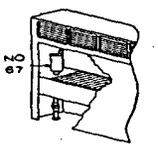
No. 67 Auto-Vent

No. 67 and 68

Auto-Vents

For convectors and baseboard radiators. Designed for vertical mounting only. Size 3 3/16 in. x 1½ in. with a ⅛ in. IP male connection for No. 67 and ¼ in. IP connection for No. 68. Designed for limited space, small, self-closing, float operated valve may be installed in trouble spots previously neglected or improperly vented. Valve is equipped with a Monel metal spring and Neoprene valve seat, which is unaffected by high temperature, oil, etc. No air chamber required. For pressures up to 30 lb.

Where a Safe Waste is needed, specify No. 7A connector fitting.



For venting convector radiators

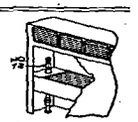
No. 72 Auto-Vent



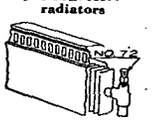
No. 72 Auto-Vent

For convectors, baseboard and free-standing radiation. Designed for both vertical and horizontal mounting. Size 1¼ in. x ⅛ in. with ⅛ in. IP male connection. No. 72 Auto-Vent is a fast venting self flushing valve of the expansion type. All expansion and contraction of the single, non-porous, composition disk is confined to four port vents. Internal siphon tube prevents waterlogging when installed in either vertical or horizonal position. Immediate drainage means fast disk drying, quick venting cycle which is continuous and no air chamber is needed.

Manual venting features plus tight shut off are all controlled by valve cap adjustment. Valve cap is tamper-proof and can be removed and replaced without damage or special tools for cleaning or flushing, if needed.



For convector radiators



For baseboard radiation

No. 14, No. 15 and No. 16

Balancing Valve Adapter Units for Hot Water Heating and Cooling Systems



No. 14



No. 15

No. 14 for copper and bronze tees. Nominal pipe sizes ⅜ in., ½ in., ¾ in., 1 in., 1¼ in.,

No. 15 for tees. Sizes ½ in., ¾ in., 1 in., 1¼ in., 1½ in.

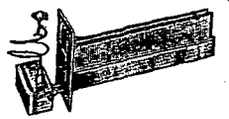
Maid-O'-Matic Valve Adapter Units make any copper, bronze, or cast brass or iron tee a balancing valve. These units, quickly soldered or sweat fitted into copperfi brass, and bronze tees or threaded into cast iron tees, regulate hot water flow through radiators, convectors, baseboard panels, radiant coils, return mains, and branches. Can be inserted in side outlet or run of tee of same pipe-size to complete either a straightway or angle balancing valve. Full, free flow of water through the tee is possible. Since there is no inside reduction of pipe diameter, there is no water restriction except for the balancing required. This permits the use of additional balancing in a heating or cooling system without preliminary planning. Precision made of non-ferrous metals. Simple balancing requires only a screw driver.



No. 16 unit for ¾ in. and 1 in. cast brass tees to complete angle balancing valves.

CONVECTOR HUMIDIFIER

Maid-O'-Mist Convector Humidifier has no flat bottom to block the flow of warm air. Their individual ⅜ in. copper water troughs are spaced 1 in. apart to allow unrestricted air flow between the evaporator pads. Maid-O'-Mist Convector Humidifiers are designed for use with radiators and cabinet convectors in hot water heating systems and for all types of warm air furnances.



2 Trough Convector Humidifier, also available in 1 and 4 trough sizes

No. 50 SERIES FLOAT CONTROL VALVES

The No. 50 Series Float Control Valves are precision designed to fill requirements for humidifying units, pan fillers, air conditioning equipment, evaporative coolers, air washers and other applications where dependable water level control is essential in limited space to meet various operating conditions.



(No. 51 float control valve)



(No. 52 float control valve)

***No. 51 Float Control Valve.** Only 5¾ in. long overall, including copper float which is 2⅛ in. diameter by 1¼ in. deep. Valve body is made of hex-brass rod with removable, raised, hard nylon valve seat. Valve stem is brass with an incased heat and oil resistant, special Neoprene washer. Valve seat is protected with a 100 mesh Monel Metal screen. Valve body is fitted with two ¼ in. IP locknuts for horizontal mounting securely in 5/16 in. hole—or valve may be screwed directly into tapped opening. Supply is equipped with union coupling for ¼ in. OD copper tubing. Working pressures up to 85 lb with capacity of ½ gal per minute at 50 lb pressure.

***No. 52 High Duty Float Control Valve.** Same general construction as No. 51 but designed for use where a larger volume of water is required. Capacity 1 gal per minute at 50 lb pressure, with working pressures to 125 lb. Overall length 8 in. with oversized float 1½ in. diameter by 4⅛ in. long.

* **No. 53Y High Duty Float Control Valve.** Same general construction as No. 52 except valve is vertically mounted with specia bracket as integral part for mounting on reservoir or pan, well above water line. Only 6½ in. long. Non-back-siphoning when installed according to instructions.

No. 54 Float Control Valve. Same general construction as No. 59 Valve except designed for use where a larger volume of water is required. Capacity 1 gal per minute at 50 lb pressure. Overall length 5 in.

* **No. 59 Float Control Valve.** Same general construction as No. 51 valve, except valve is vertically mounted with special bracket as integral part for mounting on reservoir or pan, well above water line. Non-back-siphoning when installed according to instructions. Only 5 in. long.

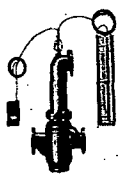
* NOTE: Nos. 51, 52, 53Y, 54, and 59 can be furnished with a ⅜ in. IP male connection at additional cost, when specified on order.

Series 6900—Water Line Float Control Diaphragm Valves. The No. 6900 Series float control valves are heavy duty large capacity water level controls. Rugged, strong and precision made, the cast brass body is fitted with a removable chromium valve seat. Controlled water level is accomplished through the action of the oversized copper float, linkage and special Neoprene diaphragm. Valve can be furnished to discharge water in an upward or downward position as specified. Can also be furnished with 8 in., 12 in., and 16 in. length float arm. Inlet and outlet tapping ⅜ in. IP. Capacities from 1½ gal to 6 gal per min. The No. 6927 Brass Mounting Plate (3 in. diameter) with gasket and screws is used where special mounting of the No. 6900 Series Valves is necessary. Write for prices and descriptive literature.

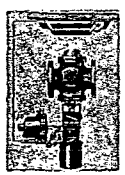
Sarco Company, Inc.

SARCOTHERM CONTROLS, INC.

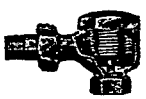
635 Madison Ave., New York 22, N. Y.



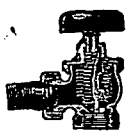
Type ST A-2 for Hot Water and Radiant Heating Systems



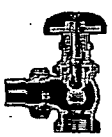
Type W for Steam Heating Systems



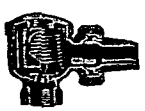
Radiator Trap, Type H



Type 40



Type SM



N-100 Medium Pressure Trap

Fully automatic weather modulated control systems for all types of heating, and all necessary accessories.

Hot Water and Radiant Heating

Sarcotherm provides a carefully engineered control system for hot water or heating which is fully modulating, and allows continuous circulation.

For forced hot water heating system for garden apartments, housing developments, hospitals, and institutions, Sarcotherm provides a completely zoned control system, including automatic night set-back and automatic morning pick-up.

Steam Heating Systems

The Type "W" Sarcotherm Control system can be applied to control condensate, vapor on low and high vacuum systems to promote fully modulating, continuous flow of steam in accordance with requirements of varying outdoor temperatures.

Complete program control panels and adjustment above or below normal requirements are provided.

Engineering Assistance

Consulting Engineers and Contractors are invited to consult with our Engineering Staff on any proposed control system. There is no obligation.

Other Products. Room thermostats, lift traps, balancing fittings, dial thermometers, motor valves, cooling controls.

SARCO RADIATOR TRAPS

Type H for vacuum, vapor, or gravity steam systems. Self-adjusting bronze bellows for highest vacuum to 25 psi. Body and cap of brass. Self-aligning head and removable seat of hard bronze or stainless steel. Angle, straightway, offset, vertical, ½ to 1 in. *Cat. N 225.*

SARCO RADIATOR VALVES

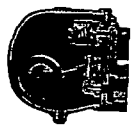
Type 40—Bellows Packless—for steam heating. Valve stem sealed by packless bronze bellows, balanced pressure type. Angle or globe, ½ to 1¼ in.

Brass body, reversible discs, bakelite handle. Available with modulating devices. Wheel or extension handles; lockshield tops. *Cat. N 255.*

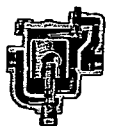
Type SM—Spring-packless or gland packed for all steam and hot water systems. Angle, globe, offset, ½ to 2 in. Same details as for above type 40. *Cat. N 225.*

SARCO N-100 TRAPS

Similar in style to above Sarco Radiator Trap, the N-100 Thermostatic Trap is suitable for operation at pressures up to 125 psi. Stainless steel renewable valve head and seat. Angle, straightway and vertical. Sizes ⅜ in. to 1 in. *Bulletin N 155.*



Float-Thermostatic Trap



Inverted Bucket Trap



Thermo-Dynamic Trap





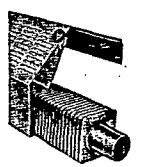
Type 50-21A Regulator for hot water storage tanks, fan units, etc.



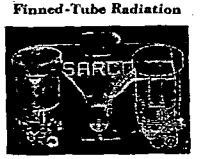
Water Blender Type DB



Type 6 Air Eliminator



Finned-Tube Radiation



Type S Condensate Pump

SARCO FLOAT-THERMOSTATIC STEAM TRAPS

Discharge condensate at steam temperature, continuously and without shock. Automatically vent air and gases by built-in thermostatic by-pass. Immediately responsive to varying loads and pressures. Sizes ¾ in. to 2½ in. for pressures to 200 psi. *Catalogs No. N-450 and N-455.*

SARCO INVERTED BUCKET TRAPS

Sarco Camlift valve mechanism provides rapid and free discharge of the condensate. Special design to prevent prime loss on light loads. Sizes ½ in. to 2 in., for pressures to 900 psi. *Catalog N-360.*

SARCO THERMO-DYNAMIC® STEAM TRAPS

These new type traps virtually eliminate maintenance. Stainless steel throughout . . . immune to both internal and external corrosion. Safe in fire-hazard locations. Not affected by corrosive condensate, superheat, water-hammer. Only moving part is solid stainless steel disc. No valve-closing mechanism to wear or stick. No critical clearances to choke. No gaskets to leak. Same trap for pressures 10–600 psi. Sizes ⅜ in. to 1 in. *Bulletin No. N-257.*

SARCO PIPELINE STRAINERS

(Also Scraper Strainers)

Brass, semi-steel or cast steel for all commercial pressures. Also furnished with hand or motor operated rotary scrapers, which clean the screen without flow interruption. *Catalogs No. N-1200, N-1210, and N-1225.*

SARCO SELF-POWERED TEMPERATURE REGULATORS

These self-powered units use the uniform force of liquid expansion. No stuffing boxes to leak, no auxiliary "power" required. All moving parts are inside the regulator. For steam, gas, oil, water or brine for temperatures to 300°F. *Bulletin N-620 and N-625.*

SARCO WATER BLENDERS AND TEMPERING VALVES

For automatic mixing of hot and cold water at desired temperatures. Type MB for showers, wash basins, etc. Type DB for use with submerged heating coils or tankless heaters. *Bulletin N-800.*

SARCO AIR ELIMINATORS

For steam and hot water heating systems, chilled water lines. Semi-steel or cast brass body. ¾ in. and 1 in. *Bulletin N 170.*

SARCOFIN FINNED-TUBE AND BASEBOARD RADIATION

A complete line with approved ratings as established by *Institute of Boiler and Radiator Manufacturers. Bulletins Nos. N-1640, N-1650, and N-1660.*

SARCO CONDENSATE AND VACUUM PUMPS

Complete line of condensate and vacuum pumps for all capacity ranges. *Bulletin N-1465.*

WARREN WEBSTER

HEATING . . . COOLING

WARREN WEBSTER & CO., 1731 FEDERAL ST., CAMDEN 5, N. J. • SINCE 1888

Representatives in Principal Cities—Consult Telephone Directory

RADIATION AND HOT WATER SPECIALTIES	Webster Walvector Radiation Webster Steel-Fin Radiation Webster Baseboard Heating Webster Hydro-Heat Specialties Webster Convector Radiators Webster Unit Heaters
STEAM HEATING SPECIALTIES	Webster Thermostatic Traps Webster Radiator Valves Webster Float and Thermostatic Traps Webster Process Steam Traps Webster Dirt Strainers Webster Lift Fittings Webster Double Service Valve Webster Boiler Protector
CONTROLS	Webster Continuous Flow Control Webster Moderator Systems Webster Control Valves
AIR CONDITIONING	Webster Riviera Heating-Cooling Conditioners Webster Newport Air Conditioners

WEBSTER TRU-PERIMETER HEATING

Webster Tru-Perimeter Heating uses Webster Walvector and/or Webster Baseboard to replace heat at the perimeter, where heat loss occurs. Heating elements are mounted close to the floor along outside walls, spreading gentle heat the entire length of the exposed walls. Webster Tru-Perimeter Heating warms the air, the floors, and the inside surface of outside walls. Air is drawn to floor level and across floor into inlet opening of radiation. Radiant heat rays strike the floor along the full length of exposed walls.



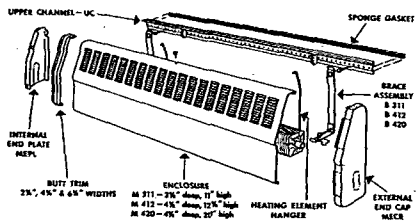
Modern School interior showing wall-to-wall installation of Webster Walvector, blanketing entire window area.

Webster Walvector

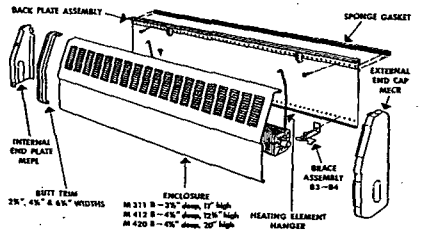
Webster Walvector for steam or forced hot water heating is an elongated convector designed for mounting along the wall, close to the floor line and under windows. For new buildings or modernization. Combines an attractive steel enclosure newly styled and improved, and a highly efficient heating element of copper tubing and aluminum fins in 3 in. and 4 in. fin sizes. Features simplified, low-cost, one man installation.

Custom Webster Walvector employs 3 in. or 4 in. fin size heating elements, single or double row. Front or top outlet enclosures for wall mounting or with integral sill. Custom Webster Walvector is custom made to meet architectural and construction requirements . . . yet sufficiently standard to permit production on an economical basis.

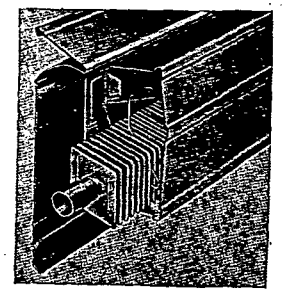
Deluxe Webster Walvector combines all the advantages of Standard Webster Walvector with a wider choice of enclosure, special wall trim and caps to achieve pleasing lower alignment and clear, simple lines.



Exploded view of Webster Walvector showing component parts



Exploded view of Webster Walvector with full back plate showing component parts



Cut-away view showing design simplicity of Webster Tru-Perimeter Baseboard

Versatile lower outlet location puts the heat where needed—excellent for panel wall construction.

Ask for Bulletins on Walvector, Custom Webster Walvector and Deluxe Webster Walvector.

Webster Baseboard Heating

Webster Baseboard Heating is a forced hot water system in which a convector heating element is concealed within a specially built metal baseboard. It is run in a continuous loop around the exposed walls of the house, with a separate loop for each floor or zone. Heating element is copper tubing with aluminum fins.

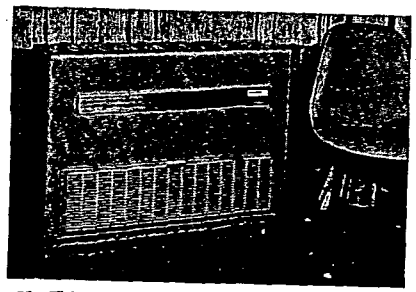
Send for bulletin.

WEBSTER AIR CONDITIONING

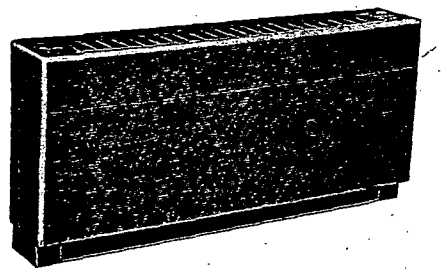
Webster Newport: Provides heating and cooling in a single compact cabinet for through-the-wall installation. Contains heating coil for connection to steam or forced hot water heating system, and self-contained hermetically-sealed refrigeration cycle. Provides easy, inexpensive way to modernize hotels, office buildings, apartments or motels.

Webster Riviera Heating-Cooling Conditioner: Cabinet-contained unit utilizes same piping for winter heating with warm water and summer cooling with chilled water. Boiler and chiller are connected in parallel and fitted with valves for seasonal change-over.

Ask for bulletins.



The Webster Newport in this well-styled executive office illustrates how this modern unit will complement any decor



Model C-160 Webster Riviera Heat-Cooling Conditioner

Illinois Engineering Company

2059 South Racine Avenue • Chicago 8, Illinois

Division of American Air Filter Company, Inc.

HEATING SPECIALTIES • CONTROL SYSTEMS • POWER SPECIALTIES

HEATING SPECIALTIES

Float and Thermostatic Traps Series G

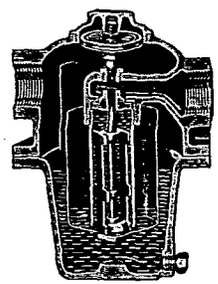
For Low Pressure Service

Compact, simple, accessible, durable, dirt-proof. Positive valve action. Wire drawing impossible. Low discharge temperatures. Efficient air venting. Trap not damaged by freezing.

Recommended Applications

For draining Low Pressure Heating Systems, Ventilating Units, Unit Heaters, Pre-Heaters, Re-Heaters, and other Fan System Heat Surface, Process Heaters, Hot Water Generators, *Drips of Mains and Risers.*

Series G operating ranges from 25 inches vacuum to 15 lbs. pressure.

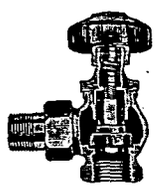


Series G

Illinois Radiator Supply Valves

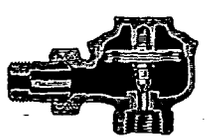
Quick Opening, Spring Loaded, Packless. For any two-pipe steam, vapor or vacuum heating system.

Extremely smooth and easy to operate with a cool, break-proof plastic handle for finger tip control.



Illinois Thermostatic Trap Series G

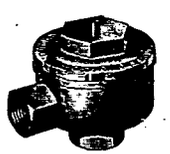
For Vacuum and Vapor Heating Systems—For Working *Pressure Up* to 15 Pounds Positive acting and responsive to the slightest temperature changes, it discharges condensate quickly and freely without loss of live steam.



Illinois Thermostatic Traps for High Pressures

Series HG—Maximum working pressure 150 pounds. *Used where neat* appearance and compactness are desirable, as for trapping sterilizers or water stills in hospitals; steam jacketed kettles, coffee urns, warming tables and for process work.

These traps are also furnished for medium pressures.



Series HG Trap

OTHER PRODUCTS

Air Vent Traps	Strainers
Boiler Return Traps	Power Specialties
Control Valves	Vacuum and Condensate Pumps
Convectors	Control Systems
Hot Water Specialties	Pressure Regulating Valves

A. W. CASH VALVE MFG. CORP.

666 E. Wabash Avenue CASH-ACME Decatur, Ill.

Representatives in principal cities. See the *"Valve" Classification in the Yellow Pages of your local telephone directory*

AUTOMATIC VALVES; BOILER PRESSURE CONTROLS; RELIEF VALVES; BACK PRESSURE VALVES; PRESSURE REDUCING AND REGULATING VALVES

BOILER PRESSURE CONTROLS



TYPE "CQ"

TYPE "CQ" DUAL CONTROL—Combines pressure reducing boiler feeder with a positive *ASME* listed relief valve in one body, plus an easily accessible in-built strainer. All bronze, body, silicone relief seat. High relief design for good water or emergency steam relief. Keeps water in system above 13 psi and below 30 psi continuously and automatically.



TYPE "CBL"

TYPE "CBL" DUAL CONTROL—Combination reducing and regulating valve, by-pass valve and positive relief valve all in one body. Automatically keeps heating system filled with water above 13 psi and below 30 psi. Relief side protects aginst high pressure due to thermal expansion. By-pass affords quick filling at higher pressures.

RELIEF VALVES AND SAFETY DEVICES



TYPE "F-51"

TYPE "F-51-F-52"—*ASME* tested and rated safety relief valves for hot water *space heating boilers, larger storage* tanks and water heaters. Fully meets *ASME* requirements in safe-guarding against maximum steam a boiler might generate in runaway conditions. Available with male and female *inlets in sizes* ¾ in. to 2 in.



TYPE "V"

TYPE "V"—Offers twofold protection for hot water heaters. Relieves in conditions of both excessive pressure and dangerous temperatures. Contains a new patented solid-filled thermostat. Reseating is automatic when dangerous conditions subside. Pressure setting from 50 psi to 175 psi available. ½ in. or ¾ in. size. *A.G.A.* listed.



TYPE "FV"

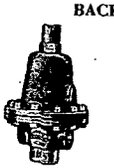
TYPE "FV"—*Provides maximum tank* and heater protection. Combination pressure and temperature relief valve. Contains new patented solid-filled thermostat, eliminating fuse plug replacement. Bronze bodies, brass internal parts, silicone seat discs. Pressure settings 75 to 150 lb. Test lever standard on all models. Size ¾ in. *ASME* and *A.G.A.* listed.



TYPE "F-3T"

TYPE "F-3T"—Newly designed offset valve body permits replacement of short or 6 in. fuse extension without removing or disassembling valve. Relieves from both pressure and temperature excesses. Pressure setting 25 to 200 lb. Fuse plug melts at 210 F. Also available with fuse plug *in side of straight* valve body. ½ in. and 3 in. sizes.

BACK PRESSURE VALVES



TYPE FR

BACK PRESSURE RELIEF VALVE—*High quality industrial type.* Iron or bronze; suitable up to 400 psi. For practically any service (except steam). *Unusually accurate control;* good capacity. ½ in. to 2 in. sizes.



TYPE "FHTL"

DIAPHRAGM TYPE PRESSURE RELIEF—For water heaters, storage tanks, etc. All bronze. *A.G.A.* listed, With or without test lever and temperature fuse plug. ⅜ in, ¾ in. and 1 in. sizes.

PRESSURE REDUCING AND REGULATING VALVES



TYPE "B"

TYPE "B"—Industrial type pressure reducing valve, designed for maintaining *desired pressure in service line regardless* of fluctuations in pressure from supply. Available with pressure settings of 2 to 200 psi, and for gas or fluid service. ¼ in. to 2 in. sizes.



TYPE "E"

TYPE "E"—Water pressure reducing valve designed for domestic supply service. *Keeps pressure in homes at 45 psi* or whatever pressure is desired. All bronze, large in-built strainer with separate cleanout. Maximum inlet 250 psi. 1 in. to 2 in. sizes.



TYPE "BBC"

PRESSURE REDUCING AND REGULATING VALVE—Similar to Type "B" above except modification for heavy, dirty or viscous fluids, including Bunker "C" Oil. Stainless steel working parts. Features new "universal" joint seat, assures good operation. ⅜ in. to 1½ *in. sizes.*



TYPE "BH"

DIFFERENTIAL PRESSURE REGULATOR—For controlling desired rate of flow by maintaining constant pressure "drop" across an orifice. Suitable for Bunker "C" oil. ⅜ in. to 1½ in. sizes.

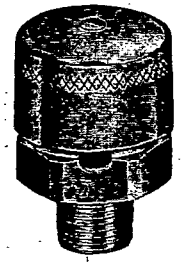
THE DOLE VALVE COMPANY

Plumbing and Heating Division

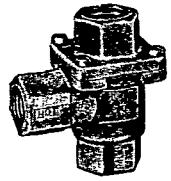
6201 Oakton Street, Morton Grove, (Chicago Suburb), Ill.

WATER MIXERS • AUTOMATIC REGISTERS • AIR and VACUUM VALVES • The All Star Line Control with DOLE

DOLE 20 SR. VENT (LEFT)



This high-quality hot water air valve is designed to provide an automatic air eliminator, a manual shutoff and a manual air vent. A Dole design developed on the hygroscopic principle by The Dole Valve Company. A selector dial permits various settings and provides for faster venting. One of a complete line of Dole Hot Water Air Valves.

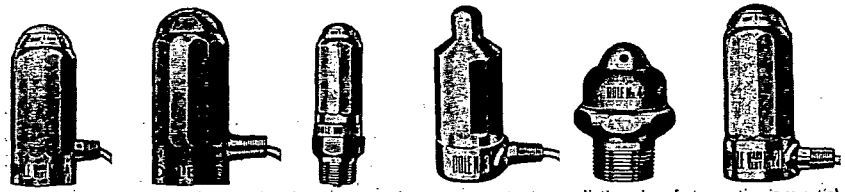


DOLE NO. 100 WATER MIXER (RIGHT)

Here is a new water mixing valve utilising the famous Dole element which gives extra power. It is noncorrosive and trouble-free. The No. 100 is made of a brass forging and yet despite its high quality is low in cost, making it a top value. Nonadjustable. 140 deg setting.

Designed for installation at the water heater or storage tank.

DOLE AIR VALVES



The complete Dole line includes air valves for every type of heating system. Below are six typical valves from the line.

DOLE NO. 1 A VARI-VENT—adjustable valve that provides an infinite number of venting speeds.

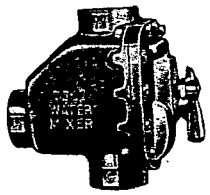
DOLE NO. 1 D—a very fast venting, fixed-orifice, nonadjustable air valve. Factory-set to vent two or three times as fast as regular type air valves.

DOLE NO. 1 B VARI-VENT—provides better control of heat on convector type radiation where faster venting is essential.

DOLE NO. 3—designed to assure utmost heating performance on hand-fired steam heating systems. Complete venting is assured at any pressure up to 10 lbs.

DOLE NO. 4 QUICK VENT VALVE—for quick venting steam mains and returns up to 25 lbs pressure.

DOLE 2 B VARI-VENT VACUUM VALVE—for use on hand-fired gravity one-pipe steam heating systems. Has the Dole patented bellows that locks air out once it has been expelled.



DOLE WATER MIXERS

Dole Water Mixers provide safer tempered domestic hot water on all tankless heaters and storage tank installations. Available in three sizes: ½ in., ¾ in. and 1 in.; adjustable range—125 to 175 deg F.

DOLE AUTOMATIC REGISTER

Provides individual room temperature control. Operates thermostatically from room air temperature. Completely self-contained. Dial permits setting for the heat desired. Made in sizes to fit practically any stack-head opening.



THE FAIRBANKS COMPANY

EXECUTIVE OFFICE: 393 LAFAYETTE STREET, NEW YORK 3, N.Y.

Branches: Boston, Chicago, New York, Pittsburgh and Rome, Ga.
Plants: Binghamton, New York and Rome, Ga.

Dependable Service GUARANTEED From This Complete Line of Bronze And Iron Body Valves Available From Your Local Distributor

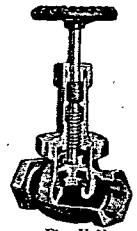


Fig. U-01
150 lb SWP

BRONZE GLOBE AND ANGLE VALVES

Pressures—125 through 300 lb SWP
Ends—Screwed, Flanged, Solder, Brazed and Hose.
Discs—Bronze: Renewable Composition; Nickel Alloy Semi-Plug Disc and Seat; Stainless Steel Semi- and Full Plug Disc and Seat.
Needle Valves
Radiator Valves
LP—Gas and Oxygen Valves

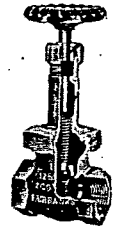


Fig. U-0252
125 lb SWP

BRONZE GATE VALVES

Pressures—125 through 300 lb SWP.
Ends—Screwed, Flanged, Solder, Brazed and Hose.
Stem Action—Rising with Solid or Double Taper Wedges, Non-Rising with Solid Wedges.
Bonnets—Screwed, Union, Bolted and O. S. & Y.
Radiator Valves
LP—Gas Valves

BRONZE CHECK VALVES

Horizontal, Angle and Vertical
Types—Swing and Lift
Pressures—125 through 300 lb SWP
Ends—Screwed, Flanged, Solder and Brazed.
Discs—Bronze, Renewable Composition, Rubber Faced.
LP—Gas Valves

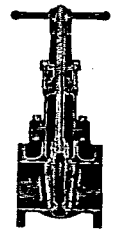


Fig. 0405
125 lb SWP

IRON BODY GLOBE AND ANGLE VALVES

Bronze Mounted
Pressures—125 through 250 lb SWP
Ends—Screwed and Flanged
Discs—Bronze, Renewable Composition, Nickel Alloy Semi-Plug Disc and Seat, 500 Brinell Stainless Steel Plug Disc and Seat.
Bonnets—O. S. & Y. and Union
Automatic Stop and Check Valves

IRON BODY GATE VALVES

Bronze Mounted and All Iron
Pressures—125 through 250 lb SWP
Ends—Screwed, Flanged and Hub
Stem Action—Rising, Non-Rising and Quick Opening
Wedges—Solid and Double Disc
Bonnets—Screwed, U-Bolt, Bolted and O. S. & Y.
UNDERWRITERS', FACTORY MUTUAL, and *AWWA* Approved Valves

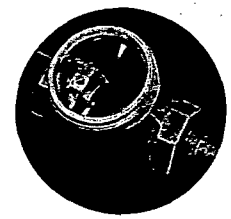


Fig. 0702
125 lb SWP

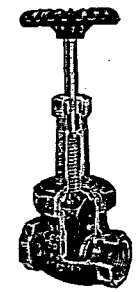
IRON BODY SWING CHECK VALVES

Horizontal—Bronze Mounted
Pressures—125 through 250 lb SWP
Ends—Screwed and Flanged.
Discs—Bronze, or Rubber.
UNDERWRITERS' and *FACTORY MUTUAL* Approved Valves.

NO OTHER VALVE IN THE WORLD LOOKS LIKE THIS



WHEN YOU REMOVE THE BONNET



FAIRBANKS (patented) RENEWABLE SEAT RING BRONZE GATE VALVES

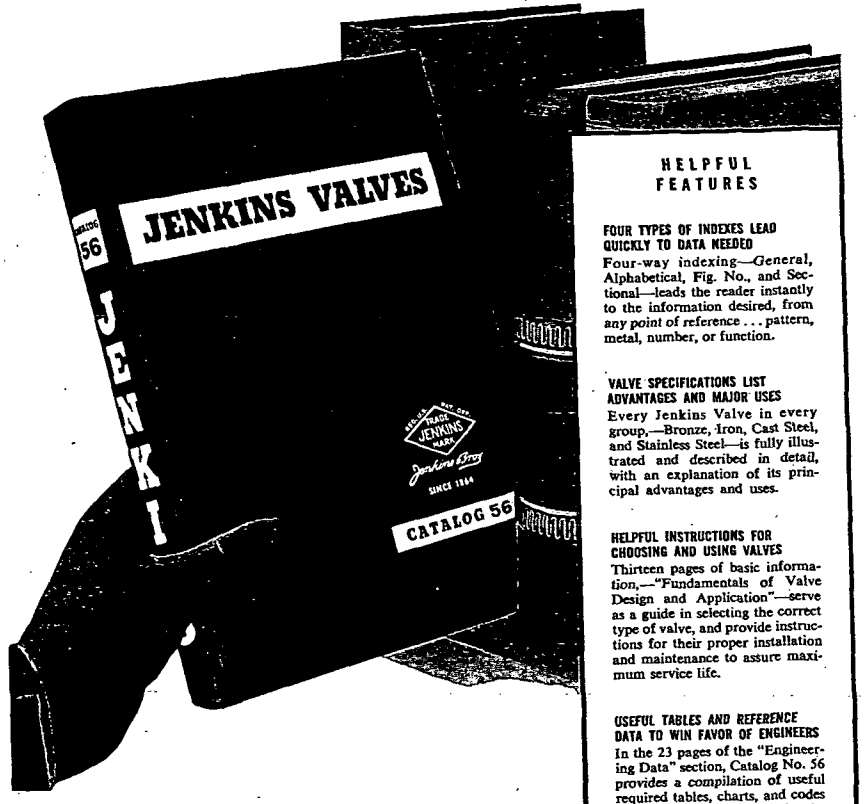
NEVER HAS TO BE REMOVED FROM THE LINE TO REPLACE SEAT RINGS

JENKINS BROS.

100 Park Avenue, New York 17, N. Y.

Bridgeport, Conn.; Boston, Philadelphia, Chicago, San Francisco, Atlanta

DO YOU HAVE THIS HELPER?



For clear, complete, easy-to-find data on valves and how to select and use them, reach for this latest Jenkins Catalog.

HELPFUL FEATURES

FOUR TYPES OF INDEXES LEAD QUICKLY TO DATA NEEDED

Four-way indexing—*General*, Alphabetical, Fig. No., and Sectional—leads the reader instantly to the information desired, from *any point of reference* . . . pattern, metal, number, or function.

VALVE SPECIFICATIONS LIST ADVANTAGES AND MAJOR USES

Every Jenkins Valve in every group,—Bronze, Iron, Cast Steel, and Stainless Steel—is fully illustrated and described in *detail*, with an explanation of its principal advantages and uses.

HELPFUL INSTRUCTIONS FOR CHOOSING AND USING VALVES

Thirteen pages of basic information,—"Fundamentals of Valve Design and Application"—serve as a guide in selecting the correct type of valve, and provide instructions for their proper installation and maintenance to assure maximum service life.

USEFUL TABLES AND REFERENCE DATA TO WIN FAVOR OF ENGINEERS

In the 23 pages of the "Engineering Data" section, Catalog No. 56 provides a compilation of useful required tables, charts, and codes —the information wanted by the men who plan piping layouts . . . *and specify valves.*

From cover to cover this 296 page general catalog of Jenkins Valves was designed to give specifiers and buyers all the data they want . . . and FAST. It's the book to reach for FIRST when you need valves or valve information. Write for your copy.

JAS. P. MARSH CORPORATION

Marsh Instrument and Valve Co. (Canada) Ltd., 8407 103rd St., Edmonton, Alberta.

Dept. 5, Skokie, Illinois

Branches in Principal Cities

Houston Branch Plant: 1121 Rothwell St., Sect. 15, Houston, Texas.

Marsh products include: Pressure, Vacuum and Compound Gages; Dial Thermometers; Steam Traps; Vents; Packless Radiator Valves and Radiator Traps; Refrigeration Water Regulators; Refrigeration Testing Instruments; Solenoid Valves; Needle Throttling Valves.

Radiator Traps—Equipped with the famous Marsh phosphor bronze diaphragm charged with volatile fluid. Extremely rugged, but so sensitive it closes tight on steam and opens instantly to pass condensate and air at only a few degrees below closing temperature. Highly efficient on pressures from below atmospheric up to 25 psi. Other traps available for pressures up to 125 psi.



Marsh Radiator Trap and Packless Valve. Both are available in full range of patterns.

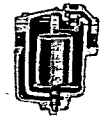
Packless Radiator Valves—Not just packless in name; the ingenious Marsh annular mechanical seal does away with packing of **any kind**. Valves close on less than one turn. For either steam or hot water. Maximum water pressure, 125 psi.

Float and Thermostatic Traps—Designed for complete removal of air and condensate from steam mains, branches and risers. Condensate discharged through float operated valve at bottom; air through vent in top. Wide range of pressure and capacity ratings. One of the many float and thermostatic traps is illustrated. Line also includes float traps.



Marsh Float and Thermostatic Trap.

Inverted Bucket Traps—Marsh quality and performance in a trap for applications where the inverted bucket type is indicated.



Marsh Inverted Bucket Trap.



Marsh No. 50 Air Vent

Air Vents—Automatically remove air and gasses from tanks, coils or piping, liquid pump systems, and other similar applications. Float holds valve tightly closed when system is filled. Opens only when air accummulates. Has vacuum check which prevents return of air. Type illustrated is for working pressures up to 125 psi. Other types, including float and thermostatic types, for low pressure.

Pressure Gauges and Dial Thermometers—The respected Marsh line covers every requirement for pressure, compound, altitude gauges, ther-altimeters and dial thermometers. Marsh instruments are the product of nearly a century of experience. Sizes, ranges and case styles for all needs.

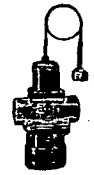


Marsh Ther-altimeter (Square and Round types)



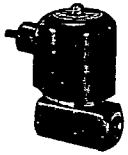
Marsh Ounce-graduated retard gauge

Refrigeration Valves—Recent addition to the Marsh line of water regulating valves is the Type 56 pressure actuated regulator illustrated. It is adjustable for both Refrigerant 12 and 22—adjustable from 60 to 270 psi—*by simply twisting* handy knurled cap on bottom. Although small and compact for installation in tight quarters, it has ample capacity for wide range of applications. Other Marsh Water Regulators, both temperature and pressure actuated, for practically every cooling or heating application.



Marsh Type 56 Regulator, made in sizes of ⅜ in., ½ in., ¾ in. and 1 in. Other types available.

Solenoid Valves—Bodies made from solid bar stock. Packless, smooth-operating, tight seating valves with coils that withstand frost and moisture.



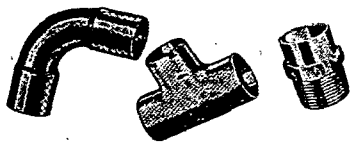
Marsh Solenoid Stop Valve. Direct acting and pilot operated valves which cover a broad range of requirements.

Ask for catalog

The comprehensive Marsh line is merely high-lighted here. Ask for catalog information covering your specific needs.

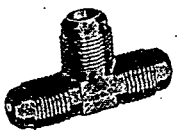
MUELLER BRASS CO.

PORT HURON 12, MICHIGAN



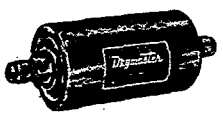
WROT-SOLDER TYPE FITTINGS

Manufactured from seamless copper tubing. Uniform solder cup depths; soldering temperatures reached quickly. Available for all plumbing, heating and air-conditioning systems.



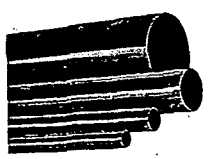
FLARE FITTINGS

Accurately machined threads for tight, leak-proof joints. Made from forgings or drawn brass rod to eliminate porosity.



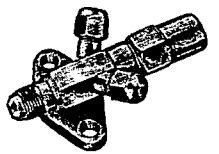
DRIERS, FILTERS, STRAINERS

Many styles and sizes to fit every installation requirement including new Drymaster with balanced filtering and drying combined in one unit, also Deluxe Driers, versatile angle-type drier-strainer; Guardsman Drier, an angle strainer and a Y-type water strainer.



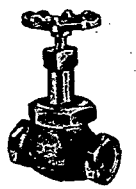
COPPER TUBING

Fine quality dehydrated and sealed copper tubing of dead soft temper. No moisture or foreign particles. Hard-drawn copper in straight lengths available in variety of outside diameters.



COMPRESSOR VALVES

Made of either forged brass or all steel, Mueller Brass Co. compressor valves are sturdy and dependable. Single, double port or adjustable . . . available in both flare and solder type connections in many sizes.



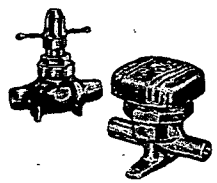
CAST BRASS WATER VALVES

Streamline cast bronze or forged brass valves. Packless and packed types variety of sizes and styles. Compression stops, gate, globe, swing, check and valves in various end connections available. 125 psi, 150 psi water, oil, gas service.



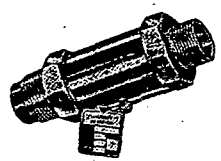
LIQUID INDICATORS

Available in straight-through, single or double port models with flare or solder end connections. Versatile new Sightmaster employs light refraction to indicate refrigerant supply. Easy, leak-proof, trouble-free installation.



REFRIGERATION VALVES

Diaphragm and packed valves for refrigeration and air-conditioning service have forged brass bodies. Carefully engineered and constructed of highest grade materials to give long trouble-free service on every installation. Many sizes in line, angle, two-way and three-way, either backseating or non backseating types.



SAFETYMASTER RELIEF VALVES

Designed and engineered for positive instantaneous action with exceptionally high volume discharge. Built to satisfy *ASA* B.9 code requirements for refrigeration vessels safety devices. Available in either straight-thru or angle types, in a total of 12 different inlet and outlet size combinations. Pressure settings from 150 to 450 lbs.

STRONG®

Strong, Carlisle & Hammond

508 Sandusky Street, Conneaut, Ohio

The Complete Steam Specialties Line

● STRONG *HYDRO-FLEX* STEAM TRAPS (semi-steel)

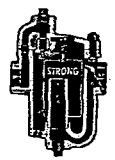
for pressures to 250 psi, temperatures to 450° F

Strong *Hydro-Flex* Steam Traps feature quick, positive *lever-lift* action for high discharge capacity and unmatched performance on light or heavy loads. There are only two moving parts—bucket and lever—reducing wear to a minimum. Pins, fixed fulcrums and hinges are eliminated from larger traps. Screwed-in renewable valve and seat are of wear-resistant *Anum-Metl*, guaranteed leakproof for one year. All internal parts are stainless steel and attached to the cover for ease of maintenance.

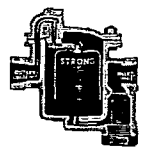
140 Series Standard In-Line Traps and Blast Traps



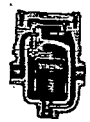
140, 141 and 142



143 and 144



140-S and 141-S



141-T and 142-T

Side inlet and side outlet permit quick removal of cover for maintenance without disturbing piping connections. Extra bottom connection is provided for draining or cleaning.

140-S Series has built-in strainer for removal of foreign matter in the line to prevent clogging of the trap.

140-T Series Blast Traps have thermal vent for use on jobs requiring very rapid starting or excessive air capacity.

Trap No.	Pipe Size (in.)	Continuous Capacity at psi	Continuous Capacity lb/hr	Weight (lb)	List Price 140 Series	List Price 140-S Series	List Price 140-T Series
140	½–¾	150	545	4	$10.50	$14.00	$12.00
141	½–¾	200	1160	7	13.00	16.00	15.00
142	¾–1	200	1520	7½	23.00	*	25.00
143	¾–1	250	2380	17	33.00	*	*
144	1–1¼	250	4970	30	46.00	*	*

* *Note:* When ordering, always state highest pressure.

40 Series Standard Bottom Inlet Traps



43, 44, 45 and 46

Heavy-duty bottom inlet, top outlet traps for severe service on low, medium and high pressures to 250 psi at 450 F. Long operating life at sustained high efficiency with minimum maintenance.

Where excessive air venting is required, the *Strong Thermal Bucket Vent* with a bi-metal element is available.

Trap No.	Pipe Size (in.)	Continuous Capacity at psi	Continuous Capacity lb/hr	Weight (lb)	List Prices
43	½–¾	200	1520	7½	$15.00
43	½–¾	250	2380	14	20.00
44	¾–1	250	4970	28	28.00
45	1¼–1½	250	8720	48	60.00
46	1½–2	250	21600	84	90.00

Note: When ordering, always state highest pressure.

● STRAINERS

Strong Strainers collect foreign matter to protect expensive equipment, reduce maintenance and servicing on pressure and temperature regulating valves, steam traps and other fittings in heating systems. Standard screen is stainless, with 0.027 in. perforations. Stainless screens with 0.045 in., 0.062 in. and 0.125 in. perforations are available at no extra cost. Special screens furnished at extra cost.

Semi-Steel Strainers



Type "SY" with bushing for blow-off connection. Screwed pipe connections.



Type "SYG" with cap and gasket for blow-off connection. Screwed pipe connections.

Ductile Iron Strainers

Type "SYD" strainers are made of *Ductile (Nodular) Iron* that can be bent or twisted without breaking in case of fire or explosion. Suitable for steam pressures of 600 psi at 650 F.

Pipe Size (in.)	Blowoff Size (in.)	List Prices Type "SY"	List Prices Type "SYG"	Type "SYD"	Extra Screens
¼	¼	$2.00	$3.50		$1.30
⅜	¼	2.00	3.50	$8.00	1.30
½	¼	3.80	3.80	7.00	1.60
¾	¼	3.65	5.00	8.00	1.70
1	¼	5.00	7.15	11.00	2.00
1¼	¼	6.40	7.80	13.00	2.60
1½	¼	7.75	10.50	17.00	3.30
2	1	11.50	15.25	20.00	4.40
2½*	1¼	25.00	25.00	—	7.00
3*	1¼	41.00	40.00	—	8.80

* 2½ in. and 3 in. strainers have flanged blowoff.

● PRESSURE REGULATING VALVES

Type "C": Pilot-operated, single-seated, packless valve. Recommended for installations requiring the most accurate and dependable regulation. Semi-steel or cast steel construction. Reduced pressures from 0 to 200 psi. Screwed connections on sizes under 2 in. Others flanged.
Type "K": Self-contained, direct-operated valve for a maximum initial pressure of 225 psi for reduced pressures from 0 to 85 psi. Has internal strainer. Screwed connections.
Type "O": Same as "K", except for sizes and no internal strainer. Screwed connections.



Type "C"

Pipe Size (in.)	List Prices Type "C" semi-steel	Type "C" cast steel	Type "K" semi-steel	Type "O" semi-steel
⅜	—	—	—	$38.00
½	$100.00	$176.00	$41.00	33.00
¾	102.00	176.00	42.00	38.00
1	104.00	176.00	53.00	
1¼	140.00	226.00	97.00	
1½	144.00	228.00	93.00	
2	193.00	321.00		
2½	256.00	405.00		
3	315.00	493.00		
4	400.00	680.00		



Type "K"

* *Note:* When ordering, state type of service, highest initial and range of reduced pressures.

For more information write for condensed *Bulletin No. SS-140-B* or *General Catalog No. 69-A.*

WATTS REGULATOR COMPANY

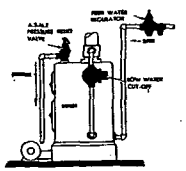
10 Embankment St., Lawrence, Massachusetts

RELIEF VALVES AND CONTROLS FOR

HOT WATER SPACE HEATING



No. 174A



PRESSURE RELIEF VALVES FOR HOT WATER HEATING BOILERS (*A.S.M.E.* Rated)

174A Series

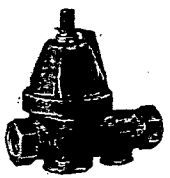
For dependable protection against over pressure. A complete line of sizes, ¾-in. to 2-in. inclusive. Bronze body and internal trim. Non-mechanically guided, stickage-free design.

74A and 740 Series

These valves are the same as the 174A Series except for increased outlets to provide greater capacity. Both series are tested and rated by the *National Board of Boiler and Pressure Vessel Inspectors.*

Size	No.	*ASME* Press. Steam Rating Btu/hr		
		30 lb	36 lb	45 lb
¾"	174A	500,000	575,000	680,000
1"	174-A*	1,008,000	1,145,000	1,355,000
1¼"	174A	1,895,000	2,180,000	2,595,000
1½"	174A	2,050,000	2,310,000	2,720,000
2"	174A	3,515,000	4,250,000	5,130,000
¾"	74A	700,000	795,000	940,000
¾" x 1"	740	2,900,000	3,335,000	3,935,000
1¼" x 2"	740	5,250,000	6,075,000	7,050,000
2" x 2½"	740			

* Model M1



No. 560

FEED WATER PRESSURE REGULATOR WITH BUILT-IN CHECK VALVE

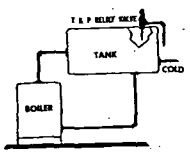
No. 560

Automatically supplies water as needed to maintain minimum system pressure under normal operating conditions. Regulator is set to deliver 12–15 lbs, but may be adjusted to higher pressures. Female inlet and outlet connections. Sizes, ½-in. and ¾-in.

HOT WATER SUPPLY SYSTEMS



No. 40XL



SELF-CLOSING TEMPERATURE AND PRESSURE RELIEF VALVES (*A.S.M.E.* and *A.G.A.* Rated)

40, 140, 240, 340 Series

For hot water supply system protection. Available in short or extended type thermal elements. Temperature relief 210°F. Pressure range 75–150 lbs. Standard setting 125 lbs. Sizes, ¾-in. to 1½-in. inlet connections. Female outlet connections.

Type No.	Connections		Style Thermostat	Temperature Water Rating Btu/hr	*ASME* PRESSURE STEAM RATING (Valves set at 125 lb) Btu/hr
	Inlet	Drain			
40L	¾" m	¾" f	Short	750,000	1,217,600
40XL	¾" m	¾" f	Extension	750,000	1,217,600
140S	1" f	¾" f	Short	850,000	1,217,600
140X	1" f	¾" f	Extension	850,000	1,217,600
141S	1" m	¾" f	Short	850,000	1,217,600
141X	1" m	¾" f	Extension	850,000	1,217,600
340	1" f	1" f	Extension	1,000,000	1,467,000
341	1¼" m	1" f	Extension	1,000,000	1,467,000
240	1½" f	1½" f	Short	3,500,000	5,707,000

(f) female connection—(m) male connection



No. 400TS

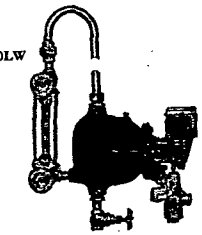
TEMPERATURE ONLY RELIEF VALVES (*A.G.A.* Listed only)

Nos. 400T, 401T

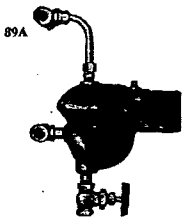
Temperature only relief valves for hot water supply tanks and heaters where separate temperature relief valves are desired or required. Btu/hr ratings up to 1,000,000. Sizes, ¾ in. or 1 in. inlet, ¾ in. female outlet. Available with fixture connection.

STEAM BOILER WATER LEVEL CONTROLS

No. 60LW

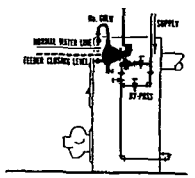


No. 89A



No. 50A





BOILER WATER FEEDERS AND FEEDER CUT-OFF COMBINATIONS (Listed by Underwriters' Lab.)

Feeder automatically feeds water to maintain safe water level in boiler. Cut-off switch interrupts burner circuit in the event of an emergency low water condition.

60 Series

Maximum steam pressure 25 lbs. Gage glass type. ½ in. feed valve connections. Available as feeder only or combination feeder and cut-off. Up to 5,000 sq ft radiation.

76 Series

Maximum steam pressure 35 lbs. Larger float and body design. Available in same models as 60 Series. ¾ in. feed valve connections. Float chamber connections 1 in. female. For over 5,000 sq ft radiation.

LOW WATER CUT-OFFS (Listed by Underwriters' Lab.)

No. 89A

Maximum steam pressure 25 lbs. Gage glass type with deep sediment chamber. Available with single throw mercury switch for cut-off service only, or double throw for cut-off and alarm service.

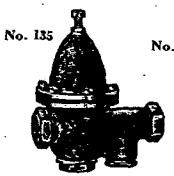
Ideally suited for self-energized control circuits on *gas-fired heating* installations.

No. 50A

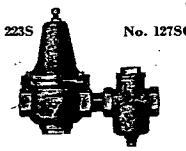
Maximum steam pressure 50 lbs. Heavier float and bellows construction for use on small and medium size process boilers as low water cut-off or pump starter. Also used on hot water heating boilers. 1 in. female top and bottom float chamber connections. See diagram on facing page.

PRESSURE REDUCING VALVES FOR WATER AND STEAM SERVICE

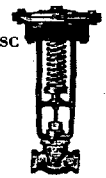
No. 135



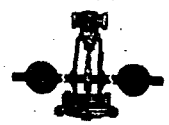
No. 223S



No. 127SC

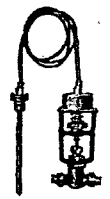


No. 125



No. 152





No. 155

REGULATORS FOR WATER SERVICE

No. 223S

High capacity water pressure reducing valve with separate strainer attached. For initial pressures up to 250 lb. Reduced pressure range 30–75 lb. Sizes 1¼ to 2 in.

No. 135

Water pressure reducing valve and integral stainless steel strainer. For water or air service on household and industrial applications. Initial pressures up to 250 lb. Sizes, ¼ in. to 2 in. inclusive. 135*LP* for reduced pressures under 30 lb.

No. 127W

Remote control, diaphragm-operated spring type pressure reducing valve. Sizes, ½ in. to 3 in. Initial pressures up to 250 lb. Reducing pressure range 10–100 lb.

REGULATORS FOR STEAM SERVICE

No. 127SC and No. 125

No. 127SC is spring type and No. 125 weight and lever type remote control diaphragm regulators. Sizes, ½ in. to 3 in. inclusive. Initial pressures up to 250 lb. Reduced pressure range range 1–100 lb. Sizes 2½ in. and 3 in. available with flanged connections. Also available as double seated regulators in sizes up to 6 in. Stainless steel trim.

No. 152

Self-contained type. Sizes, ½ in. to 1 in. Initial pressures up to 250 lbs. Reduced pressure range 5–40 or 25–100 lb.

No. 155 and No. 257

These regulators are for controlling the temperature of liquids in vats, pipe lines, tanks, etc. No. 155 single seated, No. 255 double seated direct acting regulators. Sizes, ½ in. to 1½ in. inclusive. No. 257 is reverse acting, double seated regulator. Sizes ½ in., ¾ in., and 1 in.

American Gilsonite Company

Municipal Airport P. O. Box 15, Salt Lake City, Utah

Eastern Office: 3537 Lee Road, Cleveland 20, Ohio
Distributors in principal countries of the world
Affiliate of Barber Oil Corporation and Standard Oil Company of California

GILSULATE®

TRIPLE ZONE INSULATION FOR UNDERGROUND HOT PIPES

GILSULATE is a solidified hydrocarbon of high resin content which in granular form is poured directly around underground hot pipes for insulation and corrosion protection.

GENERAL DATA

1. Low cost per installed linear foot
2. Easy-to-use: pour ... tamp ... normal pipe heat does the rest
3. Fuses itself in hours into 3 zones of permanent protection for the pipe against all commonly-encountered buried-line conditions such as acids, alkalies, roots, water, and corrosion
4. Tested and proved in actual use in hundreds of new construction and replacement installations
5. Can be used under all multiple-pipe and cramped space conditions (see illustration below)
6. Needs no housing or mechanical sheaths to protect it
7. Requires no mixing or special handling
8. Can't be punctured, obviating need to remove rocks from backfill
9. Only normal pipe spacing required; steam and condensate lines can run side by side
10. Pipe expands and contracts within GILSULATE structure
11. Makes repairs to the pipe quick, easy and economical

GRADES AND TEMPERATURE RANGES

Type A	220F to 300F
Type B	300F to 385F
Type C	385F to 520F

TRENCH SIZE

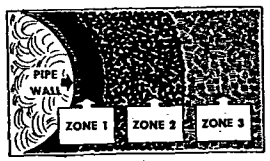
4 in. to 22 in. wider than pipe diameter, 12 in. minimum backfill over the GILSULATE bed.

EFFICIENCY

Heat loss 8-12 percent of bare pipe. Inert to ordinary acids and alkalies found in soil, excellent dielectric.

For further data or information, write: American Gilsonite Co., Municipal Air-Port P.O. Box 15, Salt Lake City, Utah, or 3537 Lee Rd., Cleveland 20, Ohio.





ZONE 1 Dense, semi-plastic coating bonded tightly to pipe surface permits pipe to expand and contract while pipe wall is hot. Zone is waterproof and a thermal and electrical insulator.

ZONE 2 Sintered zone provides excellent thermal insulation under wet or dry conditions. Zone 2 is also water resistant.

ZONE 3 Unconsolidated GILSULATE particles provide excellent thermal insulation and a high load carrying capacity.

GILSULATE DISTRIBUTORS Ask for a demonstration of GILSULATE's unique properties

ALABAMA
Palmer & Lawrence
2110 Clairmont Ave. So., P.O. Box 1573, Birmingham

ARIZONA
Carns-Hoaglund Company, 420 N. 15th Avenue, Phoenix

CALIFORNIA
Industrial Insulators, Emeryville & Los Angeles

COLORADO
Joy and Cox, Inc., 2124 Broadway, Denver 2

CONNECTICUT
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FLORIDA
George A. Israel, Jr., Jacksonville, Miami & Tampa
Manufacturer's Sales Co.,
1239 No. Pace Blvd., Pensacola

GEORGIA
John A. Dodd Co., 299 Techwood Drive, N.W., Atlanta

ILLINOIS
Eagle Industrial Co., Inc., 1021 N. 18th St., E. St. Louis
Roy O. Nelson Company, 330 S. Wells St., Chicago 6

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MASSACHUSETTS
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MICHIGAN
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John P. Wolff Company, 726 Lothrop, Detroit 2

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NORTH & SOUTH CAROLINA
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NORTH & SOUTH DAKOTA
Lignite Combustion Engineering Corp.,
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OHIO
William G. Jewett Co., 23697 Aurora Rd., Solon
The Henry P. Thompson Co., Cincinnati & Columbus

OKLAHOMA
Loeffler-Greene Supply Company,
1604 N. W. 5th, Oklahoma City

OREGON
Winsor Co., 20031 Marine View Drive S. W., Seattle 66, Washington

PENNSYLVANIA
Holly Specialties Co., Inc.,
Walnut & 63rd Streets, Philadelphia 39
J. F. Scouler Company, 2120 Market St., Camp Hill
Tuteu Equipment Co., P. O. Box 4458, Pittsburgh 5

TENNESSEE
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Hurston-Conaway, Inc., 3470 Poplar Avenue, Memphis

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UTAH
Williams, Gritton & Wilde, 204 Dooly Bldg., Salt Lake City

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Engineering Equipment Co., Ltd.,
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DURANT INSULATED PIPE COMPANY

PALO ALTO, CALIFORNIA

Durant Insulated Pipe Co., Inc. Williamstown, New Jersey

DURANT Pre-Sealed INSULATED PIPE

DURANT
manufactures pre-fabricated underground conduit systems to meet ALL federal and state specifications.

DURANT
pipe is suitable for steam, hot water, refrigerants, process fluids, and for tracer type systems.

DURANT
maintains complete manufacturing facilities on both the East and West Coasts to render maximum delivery service at the most favorable freight rate.

DURANT
representatives are at your service in the principal cities of the U. S. and Canada.

Fabricated in all standard sizes

PRE SEALED®

Manufacturers of conduit systems since 1934

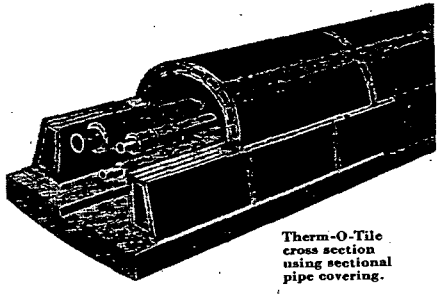
Therm-O-Tile®

conduit
FOR UNDERGROUND PIPING

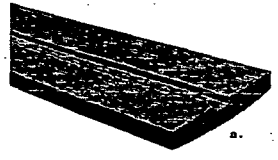
H. W. Porter & Co., Inc.

Newark 12, N. J.
or Subsidiary
REID HAYDEN, INC.
Baltimore, Md. Charlotte, N. C. Richmond, Va.

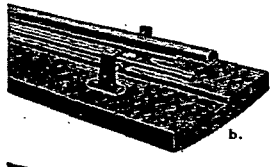
Thermo-O-Tile conduit is comprised of a vitrified tile envelope joined by cement mortar on a concrete base. It is designed to provide permanent protection of underground steam, high temperature or domestic hot water lines. Thermal efficiencies of 90 percent and higher are obtainable through the use of the proper commercial insulations.



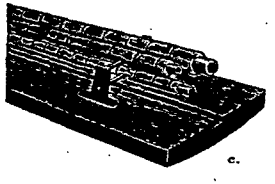
Therm-O-Tile cross section using sectional pipe covering.



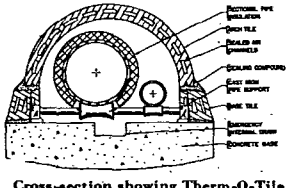
a.



b.



c.



Cross-section showing Therm-O-Tile features

Thermo-O-Tile offers other distinct advantages: 1) Reasonable in cost; 2) Unusually strong—extra strength tile; 3) Accessible for piping repairs; 4) Internal drain assures dry pipe area.

Four simple steps are required in the installation of Thermo-O-Tile:

a. **Concrete Base.** The concrete base is poured directly on the trench bottom and provides a permanent foundation for the piping and conduit.

b. **Piping and Supports.** The piping is installed on factory assembled pipe supports which have been placed on the concrete base.

c. **Insulation.** The piping may be insulated separately with sectional pipe covering or with compacted inorganic filler, as approved by H. W. Porter & Co.

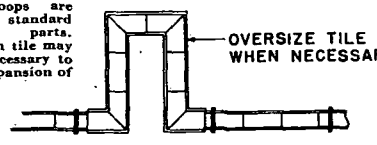
d. **Tiles.** The tile sections are manufactured from selected clays which have been salt glazed and vitrified. The base tiles are made from 4 in. to 12 in. high in 2 in. increments, and are 18 in. long. The arch tiles are made 9 in. to 30 in. in diameter in 3 in. increments, and 3 ft in length.

For complete specifications, ask for *Bulletin 9156*.

LOAD BEARING COMPARISON TABLE

Diameter Size	Pounds per linear foot, Sand Bearing Method, Pittsburgh Testing Lab	Average strength, minimum pounds per linear foot, Sand Bearing method, *ASTM* Specifications	
		Standard Strength Clay Sewer Pipe	Extra Strength Clay Sewer Pipe
	Therm-O-Tile	ASTM: C 13 - 57T	ASTM: C 200 - 57T
15"	4400	2625	4125
18"	5200	3000	4950
21"	6000	3300	5775
24"	6700	3600	6600

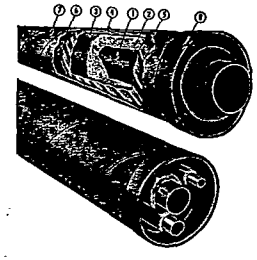
Expansion loops are formed of standard Therm-O-Tile parts. Oversized arch tile may be used if necessary to provide for expansion of pipes.



Ric-wiL Incorporated

Barberton, Ohio

Prefabricated Insulated Piping Systems Underground and Overhead
Representatives in Principal Cities



HEL-COR* INSULATED PIPE UNITS

1. Steam, hot water, oil or refrigeration pipes
2. *Insulation*
3. Asbestos-felt jacket—asphalt saturated
4. *Air space*
5. Hel-cor conduit
6. High temperature asphalt coating
7. Asbestos felt—asphalt saturated
8. Welded collar

*Hel-cor is a registered Trade-Mark of Armco Steel Corporation

Ric-wiL Hel-cor Insulated Pipe Units are factory prefabricated to meet specifications for insulating and protecting single pipe or any combination of multiple pipes—for steam, hot water, oil, other viscous fluids, process liquids and refrigeration lines. Pipes may be of any conventional material and pipe joints made as required.

Insulation may be Ric-wiL Sectional Fiberglas insulation which has a lower K factor *than most other insulations now on the* market. Return lines or other pipes not requiring insulation can be left bare.

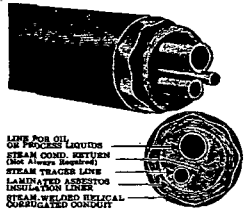
Pipe and insulation, covered with a jacket of asbestos roofing felt, are housed in exceptionally strong, load-carrying, Hel-cor conduit—16 gauge, helically corrugated steel of full welded construction, hot dipped galvanized. The *conduit is* given two heavy applications of high temperature asphalt, forming a thick reinforced asphaltic coating to assure maximum protection *against corrosive elements.*

For overhead piping, asphalt coatings and felt wrappings are omitted. Heavier gauge spiral weld conduits are also available.

Sturdy pipe supports, spaced according to requirements, carrying the weight of the pipe and assuring proper alignment, while *leaving the pipe free to expand and con*tract in the conduit without injury to the insulation.

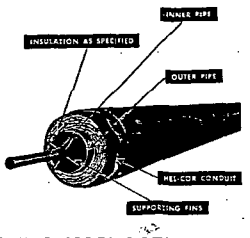
10-gauge steel collars are welded to the ends of the conduit.

This prefabrication technique produces units of exceptionally high *thermal efficiency* and durability. Because of their inherent strength and flexibility, they can be installed in shallow, narrow trenches —*even under highways* and railroad tracks. Shipped in 21 or 41-foot lengths with odd lengths and accessories as required, they are easy to handle and easy to install with a minimum of field work.



TYPE UL UNILINE INSULATED PIPE UNITS

In this type of Ric-wiL Insulated Pipe Unit pipes are nested inside an insulation liner so that they are insulated from the *exterior* but not from each other. Any desired temperature can thus be maintained in system with tracer lines for steam, hot water, brine or any other heating or cooling medium, liquid or gas. Uniline Insulation Liner is of highly efficient insulating material, in any specified thickness, fitted to inside diameter of conduit. Individual pipes may be insulated if required.



TYPE J INSULATED JACKETED PIPE UNITS

Ric-wiL Insulated Jacketed Pipe Units are designed so that desired temperatures can be maintained in the system, with either pipe carrying the process liquid while the other carries the temperature controlling medium. Outer pipe is covered with insulation as specified *in any desired thick*ness, covered by asbestos roofing felt jacket. Firm supporting fins are welded to the inner pipe only, providing free move*ment of pipes independently of each other.*



LOW TEMPERATURE UNITS

Ric-wiL Insulated Pipe Units with Low Temperature insulation are highly efficient for either *single or multiple pipe distribution* systems carrying brine, ammonia, freon, ice water, and other low-temperature liquids or gases. Low Temperature *insulation in any specified thickness is* given a heavy coating of asphalt and a tension wrap of asbestos felt to form a moisture barrier before it is inserted in conduit when moisture barrier is required. Low Temperature insulation may also be used for Insulated Jacketed Pipe Units.



CAST IRON INSULATED PIPE UNITS

Ric-wiL Cast Iron Units provide a heavy-duty outside housing for insulated pipe lines buried underground—specifically designed to afford protection from corrosion and moisture infiltration. Mechanically-sealed joints provide ease of installation.

It is an ideal system for steam or hot water lines for heating purposes, oil lines, *service lines for railroad yards, and insulated* piping for handling viscous fluids, chemicals, and refrigerants in various manufacturing and process industries.

Cast Iron Units with pipe, insulating material and conduit as per individual project specifications and requirements, are completely assembled at the factory and delivered to job site in approximately 18½ Ft. lengths, with pipe at each end extending 3" beyond insulation and conduit.

Three different methods are provided for making joint closures.

Solid Sleeve—The best flexibility and the easiest to install.

Standard Mechanical Joint—Good flexibility, lower in material cost but usually more time required to install.

Flanged—May also be furnished.

Conduit diameters *are available in all* commercial sizes of Cast Iron Pipe, allowing a wide range of pipe sizes and insula*tion thicknesses for any temperature* or *pressure conditions.* Ric-wiL Cast Iron Insulated Pipe Unit systems are complete, with all accessory units and fittings prefabricated and insulated to conform with straight length units to make a finished piping system. Including housed expansion devices.

UTILIDOR CONDUITS

Meet all service tunnel requirements of industrial plants, railroads, municipalities, institutions, etc. Wide range of conduit diameters 36" through 84".

G. S. ZIEGLER & COMPANY

Great Neck, New York

TRI-SUL-ITE

3-WAY PROTECTION SYSTEM

FOR HOT UNDERGROUND PIPES

Representatives throughout the United States and Canada

Tri-Sul-Ite is a specially selected pure gilsonite used for protecting hot underground pipes against water, corrosion and heat loss.

After being poured around and under the pipes and tamped, **Tri-Sul-Ite** forms, by proper curing, *3 efficient zones of protection*. Curing is accomplished by transmitting heat through the pipes.

ZONE 1 PROTECTION

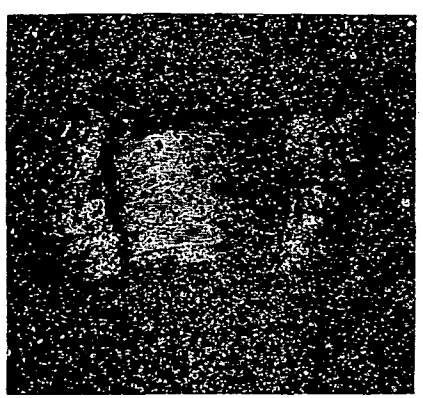
Permanent Water Barrier—The first zone of **Tri-Sul-Ite**, adjacent to the pipe, fuses from pipe heat into an impermeable, waterproof, corrosion-proof coating. It resists normal soil acids and alkalis. The first zone of **Tri-Sul-Ite** gives positive assurance that water cannot follow the grade of the pipe.

ZONE 2 PROTECTION

Excellent Insulator—The second zone of partially fused **Tri-Sul-Ite** is a highly effective insulator which minimizes heat losses.

ZONE 3 PROTECTION

Insulator and Water-Barrier—The third zone of unfused **Tri-Sul-Ite** granules adjacent to the soil, provides maximum insulating efficiency, high resistance to water penetration and protection against surface shocks and subsoil stresses.



Close-up photo illustrates the plastic zone of Tri-Sul-Ite peeled back to show excellent adhesion for pipe protection

- **Water cannot follow the grade of the pipe**
- **Excellent corrosion protection**
- **High insulating value**
- **Minimum capital investment**
- **Adaptable to wide range of multiple piping arrangements**
- **Very simple to install**
- **Cushions shocks and stresses**
- **Pipe maintenance practically eliminated**

TYPES OF TRI-SUL-ITE AND PIPE TEMPERATURE RANGES

Type I	220° F to 300° F
Type II	300° F to 385° F
Type III	385° F to 520° F

COMES READY TO INSTALL

Tri-Sul-Ite arrives at the job site ready to install. No special mixing or preparation of the material is required. It is supplied in 100-lb (2½ cu ft) black and silver multiwall bags.

FUTURE PIPING CHANGES SIMPLE

Should branch lines or additions to the piping system be necessary, it is a relatively simple matter to open the trench, expose the piping, perform the necessary work and obtain quality protection with **Tri-Sul-Ite** as in the original installation.

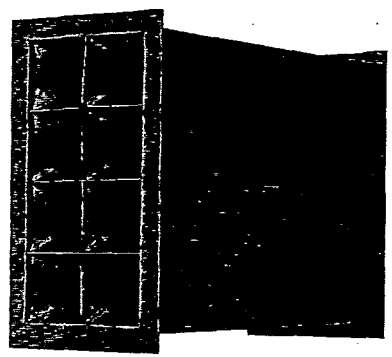
TECHNICAL SERVICE—FIELD SUPERVISION

Trained men are available to cooperate with the engineer in adapting **Tri-Sul-Ite** to the piping system. Supervision is available for installations providing it is arranged in advance.

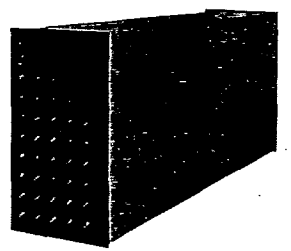
GENERAL SOUND CONTROL INC.

engineered noise control equipment

6711 S. Sepulveda Blvd.—Los Angeles 45, California—ORegon 8-3724



AD3-918 ACOUSTI-DUCT



AD8-2342 ACOUSTI-DUCT
AD STANDARD SERIES
8 LENGTH IN FEET
-2342 DUCT SIZE IN INCHES

SAMPLE SPECIFICATION

Noise within all duct work shall be controlled by installation of Acousti-Duct units as manufactured by General Sound Control, Inc., 6711 South Sepulveda Boulevard, Los Angeles 45, California. Appropriate series and model of Acousti-Duct shall be used in accordance with the drawings.

Construction of the Acousti-Duct shall be No. 22 gage or heavier sheet steel, and shall be constructed to allow straight through airflow. Smooth aerodynamically contoured sheet metal transition sections shall be built into inlet and outlet ends of each Acousti-Duct to flow air smoothly from the entire cross section of the unit into the sound absorbent tubes. All sound absorbent material shall be faced with perforated sheet metal of not less than No. 22 gage.

Acousti-Ducts shall be provided with standard 1 in. flanges at each end for connection to the duct system, or they may be provided with other standard forms of duct jointing when specifically called for on the drawings.

The contractor shall furnish and install transition sections at inlet and outlet ends of each Acousti-Duct to the adjoining ducts whenever their cross section is different from the Acousti-Duct.

Noise reduction in each frequency band shall be not less than is specified in Acousti-Duct *Bulletin 205* by General Sound Control, Inc.

Pressure drop through each Acousti-Duct shall not exceed values specified in Acousti-Duct *Bulletin 205* by General Sound Control, Inc.

ACOUSTI-DUCT

- Are complete **ECONOMICAL** duct silencer units ready, to install in heating, ventilating, and air conditioning systems.
- Are **ENGINEERED** to provide straight through air flow with aerodynamically formed sheet metal entrance and exit sections for minimum pressure drop.
- May be selected by length to provide any required amount of **SOUND REDUCTION.**
- Are available in **STANDARD SIZES** from small to large duct sizes and from 2 to 10 feet in length.
- May be easily selected from **BULLETIN 205** which provides complete engineering data.
- Send for **Bulletin 205.**

Additional information and technical data may be obtained by writing to: Mr. Richard S. Dryden, % General Sound Control, Inc., 6711 S. Sepulveda Blvd., Los Angeles 45, Calif.

ACOUSTI-DUCT

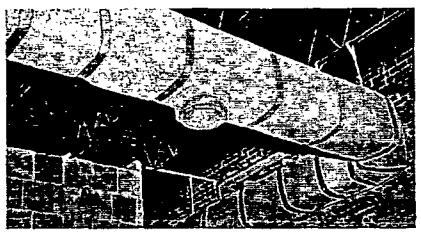
GUSTIN-BACON MANUFACTURING COMPANY

206 W. 10th St. Kansas City, Mo.

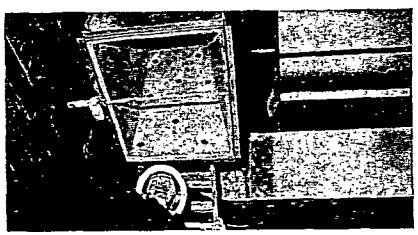
Distributors in All Principal Cities (Consult Classified Phone Directory)



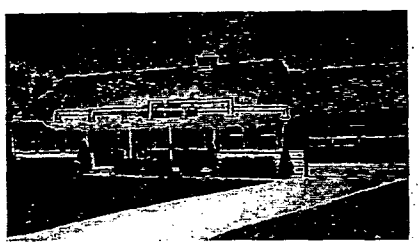
G-B Snap* On Pipe Insulation



Ultralite Duct Insulation (Thermal)



Ultralite Duct Liner (Acoustical and Thermal)



G-B Duct

G-B Snap* On® Pipe Insulation

A one-piece molded pipe insulation of fine glass fibers for heated and chilled piping. "K" factor is truly exceptional—0.23 at 100 deg mean. The insulation is feather-light . . . flexible and resilient . . . insoluble in water . . . easy to cut with a knife . . . won't break or crumble . . . dustfree . . . clean and pleasant to handle . . . available with canvas, vapor barrier, or roofing felt jackets. Snaps quickly on pipe. Single seam can be closed with any standard fastening means. Available in 6-ft and 3-ft long, single-piece sections, sizes ½ in.–36 in. and copper tubing sizes to 6 in. See *Sweet's Architectural File* or write for *Bulletin SO*.

Ultralite® Duct Insulation (Thermal)

A highly efficient "wrap-on" type thermal insulation weighing only 1 oz per board foot in the popular ¾ lb/cu ft density. Available plain or with your choice of a number of facings already adhered to insulation (when a vapor barrier and/or a base for finished appearance on duct runs is required). Shipped in compressed rolls. (Write for *AIA File 37-D-8*)

Characteristics of Ultralite. Ultralite Duct Insulation and Ultralite Duct Liner are composed of long, fine, textile-type glass fibers, bonded with a thermo-setting resin. Both are manufactured by a patented process and shipped in blanket-like rolls. Both are immune to fire, rot, corrosion, age, rodents and insects, odors, vibration, and are low in moisture absorption. Ultralite is resilient—quickly returns to original dimensions if compressed.

Application Characteristics. Ultralite Duct Insulation and *Ultralite Duct Liner* are exceptionally light in weight. Can be cut readily with a knife, quickly run around curves and corners without special fitting. Can be adhered with adhesives, metal screws and washers, even staples. Ultralite is not unpleasant to handle. Extreme simplicity of application keeps applied costs down.

Ultralite® Duct Liner (Acoustical and Thermal)

A sound-absorbing insulation to be applied to interior of duct. Effectively absorbs objectional fan, air-rush and transmitted noises. Also an excellent thermal insulation. 3 types to cover full velocity range up to 7500 fpm. Won't break, chip or dent. Available in ½ in., ¾ in. and 1 in. thicknesses, coated one side with a fire resistant coating. Can be adhered to flat metal sheets and fabricated with the metal through brakes and shears. A new coating is available which complies with all provisions of NBFU Bulletin 90-A. Carries *Underwriters'* label with flame spread classification less than 25. (Write for *AIA File 37-D-8*)

G-B® Duct

New prefabricated glass fiber for residential heating and air conditioning. Made of glass fibers and encased in an air-tight plastic vapor barrier. Hence, thermal insulation, sound absorber and vapor barrier are built-in. Comes in 6-foot sections in 4 in., 5 in., 6 in., 7 in., 8 in., 9 in., 10 in., 12 in., 14 in., 16 in. and 18 in. sizes. Approved for use in *FHA* insured homes. Tested by *Underwriters Laboratories* and now under label service.

Pipe Coupings and Fittings

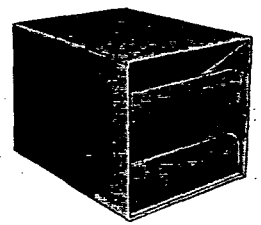
Write for descriptive literature on G-B's complete line of couplings and fittings for plain end and grooved pipe.

ELOF HANSSON, INC.

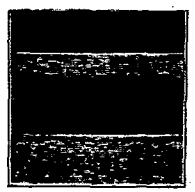
Acoustical Division, 711 THIRD AVENUE, NEW YORK 17, N.Y.

UNI-SILENCER®

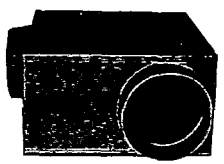
For AIR CONDITIONING and VENTILATING SYSTEMS



UNI-SILENCER* MODEL A
Moderate pressure drop, standard acoustical efficiency



UNI-SILENCER* MODEL B
Higher pressure drop, maximum acoustical efficiency

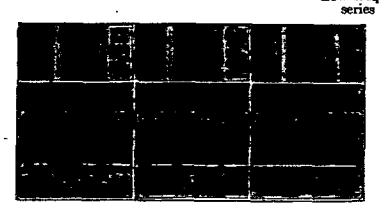


T. O. SILENCER
Elbow silencer for ducts

A full line of Elof Hansson silencers to effectively reduce airborne noise generated by fans in air conditioning, ventilating systems and by power roof ventilators. They are easily and conveniently installed in building structures at low cost due to new mass-production methods made possible by standardizing on modular sizes. Savings are of such proportions that UNI-SILENCER* prices compare favorably with conventional duct lining.

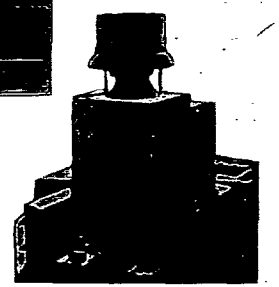
The performance of UNI-SILENCERS,* both as to acoustical attenuation and pressure loss have been tested and *certified* by independent, impartial outstanding authority. The certified data are *guaranteed in writing* and bonded. Free engineering service!

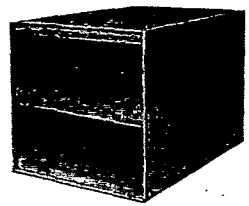
Distributors in all major cities.



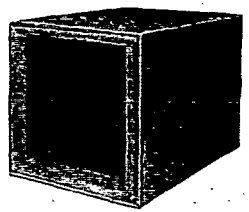
UNI-SILENCERS* are slip-joined in simple manner as with children's building blocks, to fit almost any conditions or requirements

E H ROOF VENTILATOR SILENCER for power roof ventilators





UNI-SILENCER* MODEL LPD
Lowest possible pressure drop, moderate acoustical efficiency



UNI-RESONATOR*
Low frequency duct silencer—used in series with UNI-SILENCERS*

* Reg. U. S. Pat. Off.

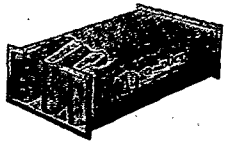
INDUSTRIAL ACOUSTICS COMPANY, INC.

341 Jackson Avenue • New York 54, N. Y.

SPECIALISTS IN NOISE AND PULSATION CONTROL
Manufacturers of Engineered Acoustical Products for Industry

Quiet-DUCT Silencers

STANDARDIZED "PACKAGE" SILENCERS for SOUND CONTROL of AIRFLOW

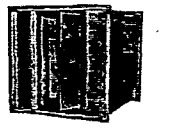


Quiet-DUCT Silencers

200 standard sizes

Quiet-DUCT Silencers are engineered units having proven acoustical and airflow performance in the silencing of heating, ventilating and air conditioning systems. Easily fitted into existing systems, or in the design stage of new projects, Quiet-DUCTS Silencers provide the following advantages:

1. Quiet-DUCT Silencers measuring in length from 2 to 10 feet will give as much attenuation as a conventional lined duct measuring from 30 to 200 ft long, depending on the frequency involved.
2. Quiet-DUCT Silencers eliminate empirical formulae and coefficient "guesstimates" out of the acoustic design of air-conditioning, heating and ventilating systems as all attenuation data has been determined by actual tests in our laboratories and verified by independent laboratories.
3. Air-flow vs pressure drop data has also been determined in our laboratories and verified by independent laboratory tests.



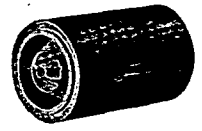
"MOD-U-SIZE" Quiet-DUCT Silencers

3 economical sizes

Full Size	Half Size	Quarter Size
24 x 24	12 x 24	12 x 12

"MOD-U-SIZE" Quiet-DUCT Silencers are availabe in three stock production sizes; Full Size—24 in. x 24 in. x 30 in.; Half Size—12 in. x 24 in. x 30 in.; Quarter Size—12 in. x 12 in. x 30 in.: These MOD-U-SIZE Quiet-DUCT Silencers actually cost 20 percent less than other silencers, an economy made possible by production-line volume of popular sizes carried in stock.

MOD-U-SIZE Quiet-DUCT Silencers achieve further economies by reducing handling and installation costs. They can easily be "built-up" in parallel or in series to meet any capacity and/or acoustic requirement. Their smaller "Module" size enables them to be easily taken through existing doorways, etc. MOD-U-SIZE Quiet-DUCT Silencers are certified for air-flow vs pressure drop, as well as acoustic performance by tests from recognized independent laboratories.



Conic-FLOW Quiet-DUCT Silencers

100 standard sizes

"CONIC-FLOW" Quiet-DUCT Silencers are designed especially for axial flow fans and the higher pressure systems. CONIC-FLOW Quiet-DUCT Silencers are available in from 4 to 10 ft lengths and provides the attenuation achieved with 30 to 200 ft of lined duct! Engineered and manufactured to a special design for exceptional attenuation, Conic-Flow Quiet-DUCT Silencers can be used on high pressures. They are also used on lower pressure systems. They are especially applicable on jobs having high acoustic sound pressure levels.

Selecting Quiet-DUCT silencers

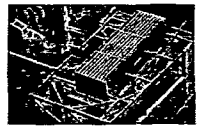
Complete technical data; attenuation, airflow vs pressure drop, and sizing, is available in the new Quiet-DUCT Silencer catalogs. Will be sent upon request.





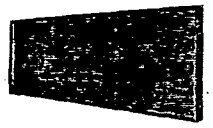
Quiet-VENT Silencers

Quiet-VENT Silencers are special size Quiet-DUCT Silencers for installation on doors and in walls of air-conditioning and ventilating systems! These unique silencers prevent the passage of sound in either direction, yet permit the free circulation of air. Quiet-VENT Silencers are designed to fit snugly inside walls or to "hug the wall", and are available in depths of 3½ in., 5 in. and 7 in.; and in widths of 30 in. and 42 in.; to handle 260 to 1710 cfm each. Used in offices, factories, apartments, homes, etc.



QUIET-FLOW Cooling Tower Silencers

Noise from Cooling Towers is controlled by "Quiet-FLOW" Silencers using special acoustic panels (and adjustable Resonators when necessary.) Quiet-FLOW Silencers for Cooling Towers are designed to assure proper attenuation of both intake and/or discharge. They are designed for minimum pressure drop to meet specifications.



FAN PLENUM ABSORPTION PANELS & SPLITTERS

Standard interchangeable acoustic panels & splitters available for hundreds of noise control applications! Available in many sizes and thicknesses, including panels with built-in resonators. Can be used to construct sound-proof enclosures, noisy machinery enclosures, shop offices, fan plenum absorbers, etc. All parts interlock and may be removed and reused again and again.

Technical Data regarding all phases of Noise Control is available by writing Industrial Acoustics Company, Inc., Air Conditioning Department.

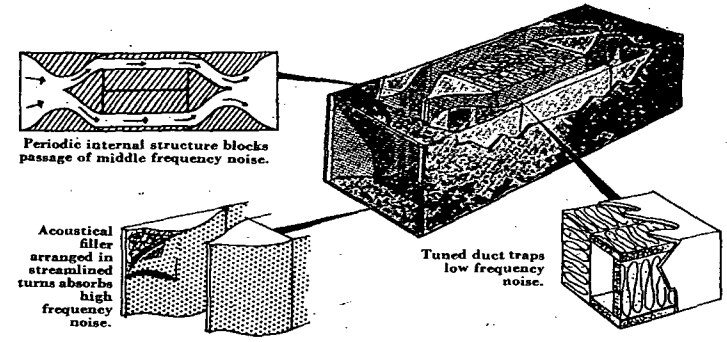
KOPPERS COMPANY, INC.

METAL PRODUCTS DIVISION
Sound Control Department • Baltimore 3, Maryland

Aircoustat

PACKAGED SOUND CONDITIONING

SOUND TRAPS FOR ALL AIR HANDLING SYSTEMS



This combination of acoustical structures in a single assembly is a unique and patented feature of *Aircoustat*. (*U.S. Pat. Nos. 2759554-2759556*)

Aircoustat sound traps are a fully effective means of silencing air conditioning and air handling system noises of all frequencies in the audio frequency range. They have been thoroughly laboratory and field tested and proved in thousands of installations throughout the United States and Canada. The units are simple to select and easy to install and no maintenance is ever required.

All **Aircoustat** cases and angles are constructed of 18 gage galvanized steel, assuring a structurally rigid, internally braced package. All units are internally damped to prevent noise leakage. All welds are continuous and all seams are mastic-sealed and guaranteed to be airtight when operating under an internal pressure of 8 in. of water. All acoustical materials are retained by a sheet of bonded matting plus a perforated galvanized face sheet. Units are rated as Class "A" fire resistant. Two inch metal extensions are standard on all units for making quick, easy, airtight connections to ductwork.

Aircoustats are available in over 60 models, with cross section sizes ranging from 6 in. x 14 in. to 36 in. x 28 in. Larger cross sections are achieved by assembling these small units into one large unit right at the job site, making it possible to accommodate unlimited airflow requirements and avoiding the necessity of handling lengths to meet all noise reduction criteria*.

All **Aircoustat** pressure loss and noise reduction values have been verified by qualified and independent laboratories—the United States Testing Company, Inc. and Bolt, Beranek and Newman, Inc., Acoustical Consultants. In every case, all acoustical and airflow data are clearly defined to the satisfaction of the architect and consulting engineer.

* See page 335 of this edition for further information and tables.

For Complete Information, Data Books and Other Free Literature, Contact: SOUND CONTROL DEPARTMENT, KOPPERS COMPANY, INC., 201 SCOTT STREET, BALTIMORE 3, MARYLAND

L·O·F GLASS FIBERS COMPANY

Dept. 19-118, 1810 Madison Ave., Toledo 1, Ohio

L·O·F GLASS FIBERS' DUCT INSULATIONS

LIGHTWEIGHT, FLEXIBLE, EFFICIENT

Properties: Made from fine, consistently uniform glass fibers bonded together in blanket form, L·O·F Glass Fibers' Duct Insulations resist the transfer of heat. They also resist fire, settling and the effects of humidity. Glass fibers are inorganic. They provide no sustenance for fungi, bacteria, rodents or insects.

Application: Lightweight, flexible and resilient, L·O·F Glass Fibers' Duct Insulations are easy to handle in large sections, easy to fasten securely by standard methods. They can be cut quickly with knife or shears; bent around curved surfaces; easily fitted into irregular areas.

DUCT LINERS

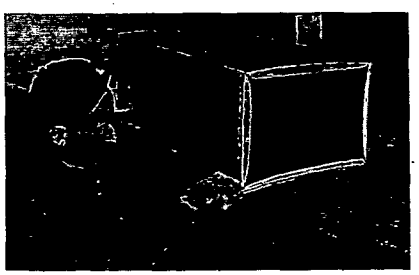
Microtex® Duct Liner effectively reduces mechanical noises, especially in the 250 to 4,000 cps range where most air-conditioning and heating system noises occur; retards heat transfer through duct walls. When lined with Microtex, the metal of the duct acts as a vapor barrier.

Product Data: Microtex is a semi-rigid, lightweight insulation made from fine glass fibers, reinforced with glass textile yarn, and bonded with a thermosetting resin. Also available: L·O·F Glass Fibers' Microlite and Super·Fine Duct Liner, non-rigid blankets made from extremely long, fine glass fibers without yarn reinforcement.

Noise Reduction Coefficient: Microtex Liner 1 in. thick has a coefficient of 0.70 in 1½ lb/cu ft density; 0.74 in 2 lb density and 0.75 in 3 lb density. No. 6 mounting used.

Packaging Data: Microtex Liner comes in densities of 1½, 2 and 3 lb/cu ft. Widths—36 in. and 48 in. Thicknesses—1 in. and 1½ in. Lengths—in rolls, 50 or 100 ft.

Coating: Duct Liners are coated on one side with black vinyl to resist air erosion even at peak operating velocities.



Microtex Duct Liner is an efficient acoustical and thermal insulation for air-supply ducts. *Lining is attached* to flat metal sheets with adhesive or mechanical clips; then formed in the brake with the metal.

DUCT WRAPS

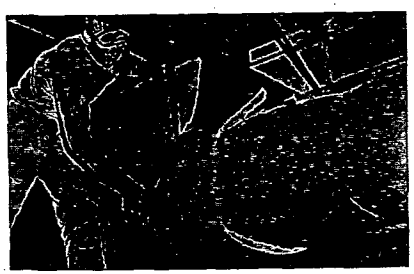
These low-density, resilient glass fiber insulating materials have high thermal efficiency and sound-absorption characteristics. Countless tiny air cells formed by millions of fine glass fibers effectively minimize heat loss or gain.

Product Data: Duct Wraps are available under the trade names Microlite®, Super·Fine and Microtex®. Microlite and Super·Fine are non-rigid blankets formed of extremely long, fine glass fibers. Microtex is a semi-rigid blanket of fine glass fibers, reinforced with glass textile yarn.

Thermal Performance: At 75 F mean temperature, Microlite or Super·Fine Duct Wrap of 1 lb/cu ft density has a k-factor of 0.24 (Microtex ¾ lb/cu ft density 0.29).

Packaging Data: Densities—½, 1 lb/cu ft (Microtex ¾ lb/cu ft). Width—18, 24, 36, 48 and 72 in. Thicknesses—1 in. and 2 in. Lengths—in rolls to 200 ft, depending on thickness and density desired.

Facings: All Duct Wraps are available plain, or with vapor-barrier facings such as reinforced paper and foil, vinyl film or aluminum foil applied. Where required, standard widths may have a 2 in. sealing tab on one or both sides.



L·O·F Glass Fibers' Duct Wraps can be fitted quickly around duct and suporting hangers. Light-gage wire ties complete the job. Workmen find these insulations soft and flexible; pleasant to handle without gloves.

L·O·F GLASS FIBERS COMPANY

SALES OFFICES: Atlanta, Baton Rouge, Boston, Charlotte, Chicago, Cincinnati, Cleveland, Dallas, Detroit, Houston, Jacksonville, Kansas City, Mo., Los Angeles, Milwaukee, New York, Philadelphia, Richmond, St. Louis, Syracuse, Toledo, Washington, D. C.
DISTRIBUTORS: (Located in principal cities)

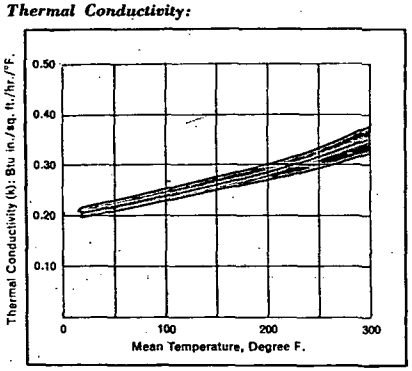
L·O·F GLASS FIBERS' MICRO-LoK PIPE INSULATION

MICRO-LoK* a resilient one-piece, molded glass fiber PIPE INSULATION with full-length "Spring-Hinge" action.

Description: Micro-LoK Pipe Insulation is made from long and fine glass fibers molded together into one-piece, three-foot sections for use in all standard iron pipe and copper tubing. Temperature limit is 350 F maximum down to −120 F. Primary uses are cold and hot water supply lines; heating supply and return lines, either steam or hot water; dual temperature lines; and chilled water, brine, refrigerant and special process piping.

Properties: Density—3.50 lb nominal. Moisture absorption—less than 0.2 per cent by volume. Alkalinity—less than 2 percent expressed as Na_2O. Corrosivity—(with steel, copper, aluminum) does not cause or accelerate. Specific Heat—0.20 Btu/lb/F. Thermal Diffusivity—@ 150F mean temperature 0.032 sq ft/hr. Micro-LoK is lightweight—a 3 ft section of 1 in. thick for fitting a 2 in. pipe weighs ¼ lb L.F. It does not break or crumble during shipment or application. Glass fibers are incombustible ... resistant to moisture ... and will not rot, shrink or swell.

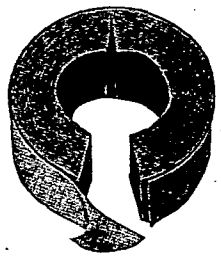
Thermal Conductivity:



Shaded area represents normal variation in conductivity due to manufacturing and raw material tolerances and to differences in testing equipment. Black curve shows average values and is used for heat loss calculations.

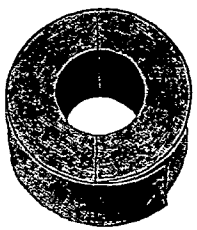
Jackets: Available in canvas, weather resistant, or a variety of vapor barrier jackets, such as general purpose, dual temperature and flame resistant. All jackets factory applied with a minimum 1½" longitudinal lap.

Application: Micro-LoK features an automatic "Spring-Hinge" opening and closing action that works like this:



Opens easily

Each section has a longitudinal cut completely through one wall and partially through on the other.



Closes Automatically

Locking action produces snug closure to insure minimum heat loss or gain.

The fiber lay-up of Micro-LoK permits easy cutting and fitting of the inside diameter to conform to irregularities common in pipe line systems, such as couplings, unions, etc. No tools—other than an ordinary knife or shears—are needed.

Packaging Data: Available in all standard wall thicknesses—especially ½ in. and ¾ in.—for all standard pipe and copper tubing sizes. Lengths—3 ft sections. Shipped in corrugated cartons.

For more information and samples on Micro-LoK Pipe Insulation, write or call the nearest *L·O·F Glass Fibers Company Sales Office, or Dept. 19-118, L·O·F Glass Fibers Co., 1810 Madison Ave., Toledo 1, Ohio.*

* Pat. Pending

OWENS-CORNING
FIBERGLAS

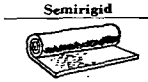
OWENS-CORNING INSULATION SELECTION GUIDE TOLEDO 1, OHIO

Owens-Corning Fiberglas Corporation has long been a supplier of quality insulation products for the heating, ventilating and air conditioning industry. The selection guide below illustrates the broad range of products designed specifically for nearly every mechanical equipment insulation requirement. Additional information and complete product specifications are available from any Owens-Corning Fiberglas Corporation branch office or authorized distributor.

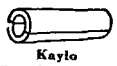
DUCT INSULATIONS

TYPE	USE	FACINGS FACTORY-APPLIED	OPERATING TEMPERATURE
 Semirigid	Semirigid board type duct insulations for all types of heating, ventilating, air conditioning and dual-temperature duct systems. Available with a variety of facings for appearance and for functional use.	Plain (no facing) for heating ducts	To 250 F
		Coated for heating ducts	To 250 F
		Vapor barrier for cooling or dual-temperature ducts with asphalt adhered kraft paper (SF) or fire-retardant embossed foil (FF)	SF to 150 F FF to 250 F
 Flexible	Resilient, flexible insulating blankets in rolls for insulating round and rectangular duct work and fittings. Available with a variety of facings for appearance and for functional use.	Plain (no facing) for heating ducts	
		Vapor barriers: Aluminum foil 0.001 in.; 0.002 in. thick; 0.004 in. grey vinyl film; 30-30-30 duplex kraft; reinforced foil and kraft laminate. Joint flange overlaps furnished.	To 250 F

DUCT LINERS

 Semirigid	Semirigid PF board type acoustical insulation liner for absorbing duct system noise. One or both sides are coated to reduce air friction loss and bind surface fibers. Also insulates thermally.	Fire-retardant	To 250 F
		Neoprene coating	Velocities to 6000 fpm
 Flexible	A sound-absorbent, mat-faced, flexible duct liner for rectangular or round duct systems. The smooth mat-faced surface reduces air flow resistance to a minimum and virtually eliminates the possibility of fiber air erosion.	Glass mat facing bonded to insulation	To 250 F

PIPE INSULATIONS

TYPE	USE	JACKET	OPERATING TEMPERATURE
 PF	With a variety of factory-applied jackets this universally used pipe insulation insulates all types of pipe and tubing from subzero to 450 F.	Universal Dual-Temperature Canvas Flame-Retardant Vapor Barrier Roofing Felt	−120 F to 450 F
 O-C Flexible Tubing Insulation	A molded foamed plastic insulation for iron pipe and copper tubing cooling and heating lines, chilled water and refrigerant lines.	None	0 to 200 F
 One-Piece	One-piece is a new addition to the Fiberglas line of pipe coverings. It has multipurpose use for all types of piping, is lightweight and easy to apply.	Same as PF	−30 F to 400 F
 Kaylo	Kaylo pipe and block insulation is a hydrous calcium silicate insulating material for heating lines and equipment operating at temperatures up to 1800 F	Canvas—Standard Weight	Standard Kaylo Up to 1200 F
		6 and 8 oz	Kaylo-20 Up to 1800 F

EQUIPMENT INSULATIONS

TYPE	TYPE	USE	FACING	OPERATING TEMPERATURE
 PF	Fiberglas PF equipment insulation is a highly versatile and efficient insulating material. Made in a wide range of thicknesses and densities to meet most requirements.	Heating and cooling equipment	Field-Applied	Up to 400 F
 Metal Mesh Blankets	Fiberglas metal mesh blankets are made with unbonded fibrous glass faced on one or both sides with wire mesh or metal lath. They conform readily to irregular shapes.	Heating equipment insulation	Available with a variety of meshor lath facings to receive surface finish.	Up to 1000 F

OWENS-ILLINOIS

General Offices, Toledo 1, Ohio

GLASS BLOCK AND TOPLITE PANELS

TWO (I) PRODUCTS

GLASS BLOCK PANELS. Owens-Illinois Glass Block are hollow, hermetically sealed glass units containing a 3/5 vacuum. They are laid in mortar to form light-transmitting panels in vertical walls. Owens-Illinois Glass Block provide quality-controlled daylighting by glass areas which reduce heat loss, surface condensation and infiltration in cold weather and heat gain in warm weather.

TOPLITE PANELS. Owens-Illinois Toplite Panels are made of hollow, partially evacuated and hermetically sealed glass units, prefabricated at the factory into panels of insulated aluminum-grid construction. Used horizontally in roofs, Toplite Panels offer controlled daylighting with reduced heat loss, surface condensation and solar heat gain.

THERMAL INSULATION. The coefficients of heat transmission 'U' for panels of O-I Glass Block and Toplite are as follows:

Size of Block		'U' (Btu/sq ft/hr/F Temp Difference)
5¾ in. sq		.60
7¾ in. sq		.56
7¾ in. sq	with Glass Fiber Insert	.48
11¾ in. sq		.52
11¾ in. sq	with Glass Fiber Insert	.44
Toplite Panels		.58

See page 123 in this GUIDE for additional U-Factors applicable to Owens-Illinois Glass Block.

SURFACE CONDENSATION. Because of the low over-all, air-to-air heat transfer, the exterior air temperatures which will produce condensation on panels of O-I Glass Block and Toplites are much lower than those for ordinary single glass. This permits higher humidities to be maintained where needed for winter air conditioning for comfort or for industrial processes.

INFILTRATION. Panels of glass block which are laid in mortar provide a barrier against infiltration. Dusts, drafts, and air and vapor leakage are minimized. Natural ventilation requirements can be met by installing windows in glass block panels.

SOLAR HEAT GAIN. Complete instantaneous solar heat gain data applicable to Owens-Illinois Glass Block designs for all exposures are given on Pages 190 through 194 of this Guide. The O-I design numbers which compare with the ASHAE type numbers are as follows:—

ASHAE Type No.	O-I Block Design Nos.
I	307, 316, 330, 331, 370
II	340
III	340 with Fiberglas Insert
IVA	365
V	363

New O-I glass block designs #80 solar-selecting and #80 solar-selecting green are available. They have a lower solar heat gain than any of those shown in THE GUIDE. These blocks have a variable rate of solar energy transmission depending on sun altitude. When the solar energy is maximum on the exterior of the wall, the #80 glass blocks have a minimum transmission. Calculated values of solar heat gain for these blocks are now available.

Solar heat gain through Toplite panels is much less than that through single glass or plastic skylights, due to their solar-selecting characteristics. They act as a mirror in reflecting a high percentage of the solar energy when the sun is at high altitudes as in the spring and summer months.

DESIGN AND SIZES. Owens-Illinois Glass Block are made in a variety of face designs and in three standard sizes (5¾ in. sq, 7¾ in. sq, and 11¾ in. sq—all three are 3⅞ in. thick). Mortar joints of ¼ in. thickness are recommended.

Prefabricated Toplite Panels are available in seven standard sizes: 2 x 2, 3 x 3, 4 x 4, 2 x 6, 3 x 6, 4 x 5, 5 x 5.—all 7 are 3 1/16 in. thick. Complete technical data, description and details will be gladly sent. Write: Kimble Glass Company, subsidiary of Owens-Illinois, Department HV, P. O. Box 1035, Toledo 1, Ohio.

PITTSBURGH PC CORNING

DEPT. S-9, ONE GATEWAY CENTER PITTSBURGH 22, PA.

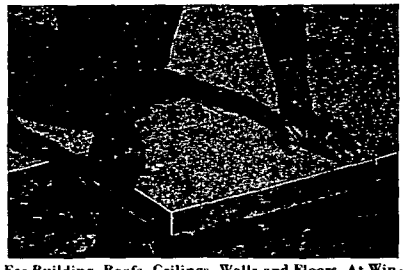
FOAMGLAS® . . . the cellular, stay-dry insulation

What FOAMGLAS® is . . . FOAMGLAS is a unique thermal insulation composed of closed cells of glass. It possesses a combination of physical characteristics which makes it virtually an ideal insulation.

It is completely inorganic and thus will not burn, rot, swell or slump. Its closed cell structure renders it impervious not only to moisture but to moisture vapor as well. Because it is water and vapor proof, it retains a constant insulating value (k) for the life of an installation. In addition, **FOAMGLAS** doesn't require a costly separate vapor barrier.

FOAMGLAS is unusually strong, having an average ultimate compressive strength of nearly 7 tons per square foot. But **FOAMGLAS** is light in weight, works easily with ordinary tools and therefore is very easily installed.

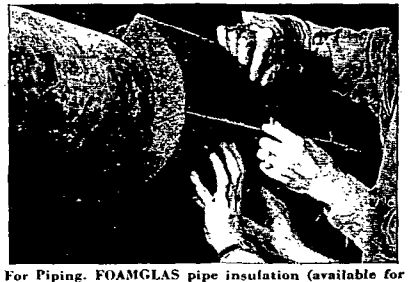
All of these characteristics have enabled **FOAMGLAS** users to benefit from its constantly-high insulating performance. That's why **FOAMGLAS** has gained recognition as an excellent insulation for roofs, ceilings, walls, and floors of all kinds of buildings as well as for hot, cold and dual temperature piping, tanks, and other equipment.



For Building, Roofs, Ceilings, Walls and Floors. At Winston-Salem, FOAMGLAS' high strength permitted insulating parking deck-roof of Sears Roebuck's air conditioned store. In your building, moisture-proof FOAMGLAS insulation can help maintain air conditioning system efficiency.



For Ducts. Light-weight blocks of FOAMGLAS were easily applied to these metal air conditioning ducts at the Moses H. Cone Memorial Hospital, Greensboro, North Carolina.



For Piping. FOAMGLAS pipe insulation (available for up to 36 in. pipe size) was applied on this welded pipe carrying cold water for the system air conditioning several new skyscrapers.

HOW FOAMGLAS can benefit you:

Properties		Benefits
Absorption	0	Moisture reduces insulation effectiveness and life. *FOAMGLAS* stays dry, maintaining constant insulating value.
Combustibility	Incombustible	Protects from fire hazards.
Compressive Strength	100 lbs/sq in.	Offers unusually high strength for varied structural and load bearing applications.
Hygroscopicity	No increase in weight in 365 days in air at 90 percent relative humidity	*FOAMGLAS is its own vapor barrier.*
Density	9 lbs/cu ft (ave)	Lightweight, easy to handle and install.
Thermal Conductivity (k) at 50°F	0.38 Btu/hr/sq ft/°F/in.	Excellent insulating value that *remains constant.*

FOAMGLAS is approved by *Underwriters' Laboratories* and is listed under Label Service.

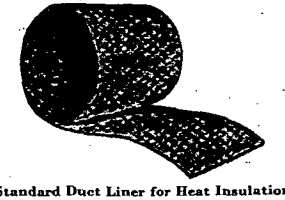
For full technical information, please write for our new catalogs indicating which of the above uses of FOAMGLAS you are most interested in. Address Dept. S-9, Pittsburgh Corning Corporation, One Gateway Center, Pittsburgh 22, Pa.

Pittsburgh Plate Glass Company

FIBER GLASS DIVISION

One Gateway Center Pittsburgh 22, Pennsylvania

FIBER PG GLASS



Standard Duct Liner for Heat Insulation



Neoprene Coated Duct Liner for Sound Absorption and Heat Insulation

Pittsburgh Superfine insulation is composed of extremely fine glass fibers, ranging upward in diameter from only 3/100,000 of an inch. It's an exceptionally light-weight, efficient thermal and acoustical insulating material. When specifications call for control of heat or cold—or you need sound absorbed or deadened—here are some of the advantages Pittsburgh Superfine Fiber Glass insulation provide:

High Thermal Efficiency—Thermal conductivity, k, at a mean temperature of 75 F is 0.22 to 0.29 Btu (sq ft) (hr) (deg F) (in.) thickness, depending on density. Effective for both high and low temperatures ranging up to 500 F.

Fine Acoustical Properties—Noise reduction coefficient of a 1 in. thickness is as high as 0.85 with the sound absorption curve showing high efficiencies in the ranges of the most irritating noise.

Light Weight, Easy to Handle—Added load in any application is virtually negligible. Readily cut with knife, shears, or die. Can easily be sewed and quilted, glued, tacked or taped into position.

Flexible, Resilient, Tough, and Soft—Made from long, fine glass fibers to attain superior qualities. Can be used on round or irregularly shaped objects and spaces as easily as in conventional layers. Fits closely into corners and tight spots quickly and easily, without damage to its structure.

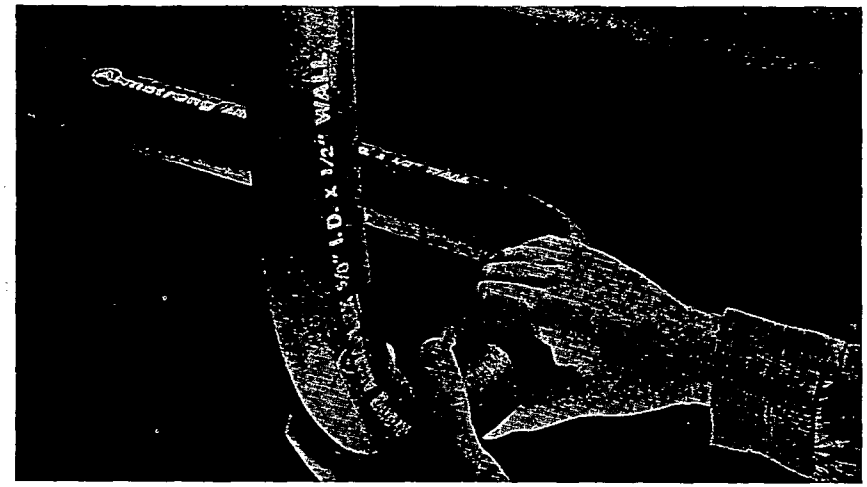
Saves Vital Space—In use: Higher insulating efficiency means less thickness of Superfine required, as compared to ordinary insulation. In shipment: Packaged compressed to reduce shipping and storage space.

STANDARD SPECIFICATIONS—Pittsburgh Superfine is available in standard roll widths of 18 in., 24 in., 36 in., 48 in., 54 in. and 72 in. It can be supplied in any desired width down to 2 in. Rolls are wound on 2 in. tubes and wrapped in heavy kraft paper. Superfine is also available in special lengths on request. It is furnished neoprene coated for *Duct Liner or with various paper, cloth or foil facings.

Type	Density (lbs per cu ft)	Thickness (inches)	Approx sq ft per lb	Typical Roll Length (feet)	Approx OD of Rolls (inches)
S-314	½	½	40	100	18
		¾	26.7	100	22
		1	20	100	26
		1½	13.3	100	30
		2	10	100	32
		3	6.7	100	38
S-315	¾	½	64	100	10
		½	32	100	16
		¾	21.3	100	20
		1	16	100	23
		1½	10.7	100	26
		2	8	100	28
		3	5.3	50	27
S-316	1	½	48	100	10
		½	24	100	14
		¾	16	100	17
		1	12	100	20
		1½	8	100	24
		2	6	100	34
		3	4	100	39
S-317	1¼*	½	32	100	13
		½	16	100	18
		¾	10.7	100	22
		1	8	100	25
		1½	5.3	100	31
S-318	2*	½	24	100	13
		½	12	100	23
		¾	8	100	27
		1	6	100	32
		1½	4	50	29
S-319	3*	½	16	100	16
		½	8	100	23
		¾	5.3		
		1	4		
S-105 (Aircraft)	½	½	40	50	14

* Duct Liner Densities

Armstrong
CORK COMPANY
Lancaster, Pennsylvania



ARMSTRONG ARMAFLEX PIPE INSULATION

for all air-conditioning, refrigeration, and plumbing and heating lines

Armstrong Armaflex is a flexible foamed plastic insulation in pipe covering and sheet form. Pipe covering is manufactured in nominal ⅜ in., ½ in., and ¾ in. thicknesses. These thicknesses are calculated to prevent condensation on cold lines within their individual recommended usage ranges at temperatures down to zero. Armaflex also can be used on heating lines, where it will withstand temperatures to 200° F. On outdoor installations, consideration should be given to weatherproofing. In sheet form, Armaflex is used for insulation of larger pipe sizes and for covering tanks and vessels of any size or shape.

The closed cell structure of Armaflex gives it a low k value of 0.28 at 75° F. mean temperature. Armaflex resists deterioration, so efficiency remains high in service.

The uniformly sized closed cells of Armaflex seal out air and moisture. The result is a vapor transmission rating of less than 0.1 perm.

Shrinkage of Armaflex when the inner surface is exposed to temperatures to 200° F. is negligible. Armaflex is only slightly affected by solvents and for all practical purposes is unaffected by oils, common corrosive chemicals, or fungus.

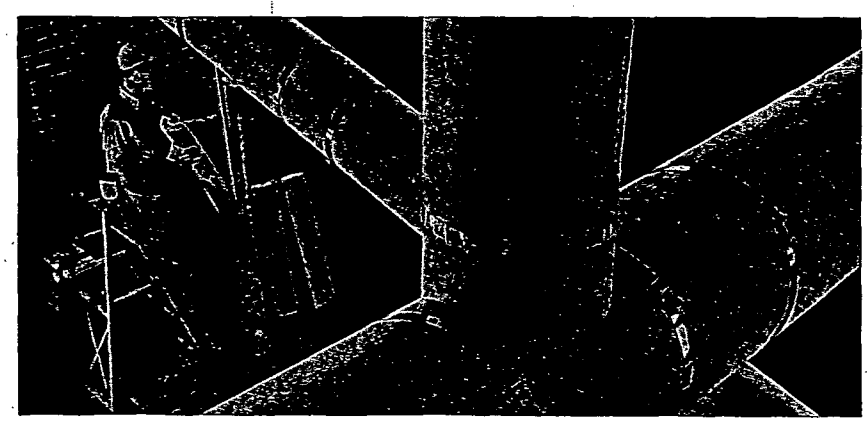
The extreme flexibility of Armaflex Pipe Covering and Sheets makes application very rapid. They can be applied in close quarters. Pipe covering can be slipped onto pipe and tubing prior to its installation or slit and snapped over piping that is already in place. The only sundry material required is Armstrong 520 Adhesive. Scotch Brand #472 Tape may be used to seal Armaflex joints on straight piping. Fittings are insulated with miter cut pieces of covering, quickly cut and assembled on the job, using 520 Adhesive to adhere joints.

Physical Properties of Armaflex

Pipe Covering	
Wall thickness—inches (nom)	⅜, ½, ¾
Inside diameter—inches	⅝ to 3⅛
Length of section—feet	6
Sheet Armaflex	
Size—inches	30 x 36
Thickness—inches	⅛, ¼, ⅜, ½, ⅝, ¾
Thermal conductivity	
Btu-in./sq ft-hr-°F-k_{75}	0.28
k_{125}	0.30
Water vapor transmission	
Perms (Grains-in./sq ft-hr-in.-Hg.)	less than 0.1
Air transmission—static pressure at 10 in. Hg. cu ft-in./hr-sq ft	less than 0.03
Water absorption—28 days submersion—percent by volume	3.9
Density—lbs-cu ft	
	pipe covering 7.5
	sheets 6.5
Fire rating	self-extinguishing

For complete application instructions and specifications, write today to Armstrong Cork Company, 5800 Preston Avenue, Lancaster, Pennsylvania.

THE DOW CHEMICAL COMPANY
Plastics Sales, Midland, Michigan



STYROFOAM® STOPS MOISTURE, REPELS HEAT

Advantages of Styrofoam* as a Low Temperature Pipe Covering

(1) Low "k" factor stays low because Styrofoam (a Dow plastic foam) resists water and water vapor. (2) Prevents condensation and dripping. (3) Light, clean, easy to handle, economical to apply. (4) Resists rot, mold, deterioration. (5) Will not split or crack nor is it subject to ice build-up around valves in extreme cold.

USES

Pipe covering • Wall, floor and ceiling insulation for low temperature work • Cavity and core walls • Insulation-plaster base • Refrigerated vehicles • Curtain wall panels • Perimeter and foundation insulation

ENGINEERING DATA

THERMAL PROPERTIES

Thermal Conductivity (k factor), Btu/(hr) (sq ft) (°F/in.)	Average less than 0.28 at mean temperature of 40°F.
Linear Thermal Coefficient of Expansion	.00003 to .00004 in./in.°F. between . . . 0°F. and 80°F.
Specific Heat	0.27 Btu/(lb.) (°F.) at 40°F.
Resistance to Heat (Maximum recommended temperature for continuous use)	Styrofoam 22—170°F. Styrofoam 33—160°F.

PHYSICAL PROPERTIES

		Styrofoam 22	Styrofoam 33
Density	(lb/cu ft)	1.6-2.0	1.7-2.3
Compressive Yield Strength	(psi)	16-32	16-38
Tensile Strength	(psi)	45-61	65-95
Shear Strength	(psi)	27-36	30-40
Flexural Strength	(psi)	42-61	48-99
Compressive Modulus	(psi)	1200-1700	1500-2000
Bending Modulus	(psi)	1000-1285	1250-1760
Modulus of Rigidity	(psi)	700-1600	1000-1300

WATER RESISTANCE PROPERTIES

Capillarity	None
Water absorption (When subjected to 90°F., 90 per cent relative humidity for 15 days)	Less than 0.03 per cent by volume
Water absorption (When completely submerged for one week)	Water pick-up only on surface cells less than 0.15 lb./sq. ft. of areas

VAPOR TRANSMISSION

When Styrofoam acts as a barrier between spaces having different atmospheric conditions	1.0-2.0 grains/(sq ft) (hr) (in. thickness) (in. of Hg. vapor pressure difference)

MANUFACTURERS

Styrofomics, Inc., 17 Hawkins Street, Somerville, Mass. • Glo-Brite Products Incorporated, 6415 North California Avenue, Chicago, Illinois • Tufflite, Inc., Ballston Spa, New York • Styro-Fabricators, 1401 Fairfax Trafficway, Kansas City, Kansas • Robinson Industries, 3051 Curtice Road, Coleman, Michigan • MMM, Incorporated, P.O. Box 968, Portsmouth, Ohio • Kennedy Industries, 4447 Alger Street, Los Angeles, California • Fastab Insulation, Inc., 304 Cleveland Transit Building, Cleveland 14, Ohio • Fibrous Glass Products, Inc., Hicksville, L. I., New York • Kransco Manufacturing Company, South San Francisco, California

For further information on Styrofoam in any application write to THE DOW CHEMICAL COMPANY, Midland, Michigan, Plastics Sales Department 1941R. We can also give you information on Scorbord*, the new easy-to-apply member of the Styrofoam family specifically designed for perimeter and cavity wall applications.

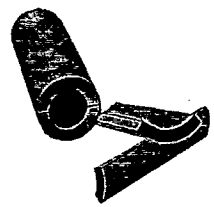
* STYROFOAM and SCORBORD are trademarks of The Dow Chemical Company

YOU CAN DEPEND ON DOW

THE PHILIP CAREY MFG. COMPANY

LOCKLAND, CINCINNATI 15, OHIO

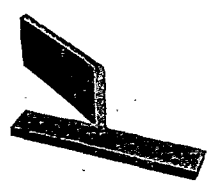
Carey PRODUCTS



ALLTEMP PIPE INSULATION

Uses: Alltemp pipe covering is an all-purpose insulation for temperatures from 100 F to 1600 F. It has an extremely low thermal conductivity, low internal shrinkage, and exceptional moisture resistance. Alltemp is particularly recommended for irregular surfaces, and for use around valves and fittings where it can be applied in single-layer, or broken-joint double-layer construction.

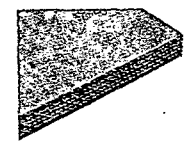
Description: Alltemp is a combination of special inorganic silicas and asbestos fibres. Every cubic foot is honeycombed with millions of tiny sealed air cells. Having excellent physical and chemical properties, Alltemp retains its high thermal efficiency and structural strength even when subjected to long and continuous exposure to high temperatures and moisture. Alltemp sectional pipe covering is available through 12 in.; tri-sections and quad-sections available in larger sizes. "Nesting OD" pipe covering sizes provide accurate tight fits over or into another size. This permits a wide range of assemblies and gives the advantage of building up many thicknesses from a limited number of prime units.



ALLTEMP BLOCK INSULATION

Uses: Alltemp Block is recommended for high temperature surfaces such as oil stills, ovens, boiler walls, breechings, furnaces and other equipment whose surfaces are between 100 F and 1600 F. It has an extremely low thermal conductivity, low internal shrinkage, and exceptional moisture resistance. Alltemp Block is available for flat and curved surfaces. It is particularly recommended for irregular surfaces which require an easy-to-apply insulation.

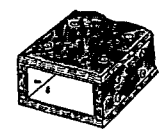
Description: Alltemp is a combination of special inorganic silicas and asbestos fibres. Every cubic foot is honeycombed with millions of tiny sealed air cells. Having excellent physical and chemical properties. Alltemp retains its high thermal efficiency and structural strength even when subjected to long and continuous exposure to high temperatures and moisture. It is available in blocks 1½ in., 2 in., 2½ in., 3 in., 3½ in. and 4 in. thick, measuring 6 in. and 12 in. wide and 36 in. long.



FIREFOIL BOARD

Uses: Carey Firefoil is recommended for insulated and acoustical duct construction in heating and air conditioning systems, industrial ovens, coil housings, plenum chambers, marine bulkheads, etc. Firefoil is recommended for interior use and under dry conditions.

Description: Firefoil Board is formed of laminated plies of corrugated and flat asbestos paper sheets, bonded with a special fire-resistant adhesive. Firefoil has approximately eight corrugated layers per inch of thickness. It is chemically treated and hardened to give it strength and resistance to both fire and water. Carey Firefoil is a most efficient and economical panel material for many industrial and building construction applications. Firefoil is recommended for use at temperatures up to 1000 F. Carey Firefoil is available in thicknesses from ½ in. to 3 in., in ½ in. multiples. Standard panel size is 48 in. by 96 in., but smaller sizes can be factory-cut to order.



CAREY SPUN WOOL DUCT INSULATION

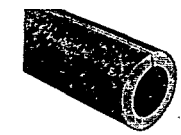
Uses: For insulating rectangular, metal ducts for ventilating, air-conditioning and cooling systems in schools, stores, office buildings, factories, apartments, etc., where non-combustible insulation is required.

Description: Made in semi-rigid panels, processed from long, fine fibers of spun mineral wool. Panels are available faced with 0.0025 embossed aluminum foil, securely attached with non-flammable adhesive; coated for interior duct sound-proofing or asphalt-kraft vapor barrier facing. Light in weight, with a density of approximately seven pounds per cu ft, this insulation is easy to work and handle, is quickly installed and provides a high degree of thermal efficiency. Foil faced duct insulation is non-corrosive and has been tested and approved by *Underwriters Laboratories*. Sheet size is 24 in. x 48 in. in 1 in., 1½ in. and 2 in. thicknesses.

THE PHILIP CAREY MFG. COMPANY

LOCKLAND, CINCINNATI 15, OHIO

Carey PRODUCTS

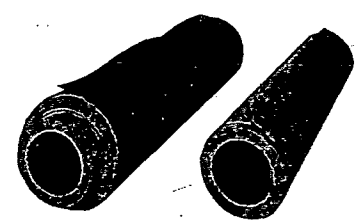


THERMOGLASS PIPE COVERING

Uses: Thermoglas pipe covering is used for pipe temperatures up to 350 F. It is recommended for hot water and low-temperature steam piping in industrial plants, commercial buildings and institutions. Thermoglas can be used as an outside layer over a high-temperature insulation.

Description: Thermoglas pipe covering is used on pipes having temperatures up to 350 F. It simply "snaps in place." Thermoglas is exceptionally lightweight insulation composed of long, fine, blown glass fibers bonded with thermal-setting phenolic resin. It is a strong, semi-rigid and resilient insulation with no coarse fibers to irritate hands or skin.

Thermoglas is available in one-piece sections made in 3-ft lengths for small sizes. Medium and large sections are furnished in 6-ft lengths. Canvas white and black dual-temperature and roofing felt jackets are available.



SUPERBO PIPE COVERING

Uses: Carey Superbo is a rigid dual temperature pipe insulation for chilled water piping or piping operating at alternating high and low temperature with a service range of from 35 F (lower limit) on the cooling cycle to 200 F (maximum) on the heating cycle. The integral vapor barrier jacket properly applied and sealed will protect chilled water lines from condensation or sweating. Thickness requirements vary with the pipe size and conditions of temperature and humidity. Where recommended thickness exceeds 1½ in. and the pipe size is 6 in. and over, double thick Superbo construction and broken joint application is required.

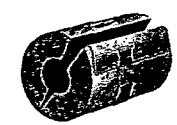
Description: Carey Superbo is a rigid pipe insulation made of multiple layers of "dry" insulating felt (wool felt) firmly cemented together. The inner surface is lined with asphalt saturated organic felt. A factory applied white vapor barrier jacket is furnished consisting of aluminum foil laminated between two layers of special creped Kraft paper, bonded with asphalt. This same vapor barrier jacket is also available with a black outer surface. Superbo is furnished with 4 in. wide collars of the same vapor barrier material as the jacket.



IMPERVO WOOL FELT PIPE COVERING

Uses: Impervo is used primarily for insulating chilled water and ice water pipe lines operating at temperatures between 40 F and 60 F. It is specially designed to keep water cold and prevent condensation causing damage from dripping. Recommended for cold service water piping under particularly high humidity conditions such as in laundries, kitchens, bakeries, textile and paper mills.

Description: Impervo is composed of multiple layers of insulating felt with water resistant felts on the inner and outer surface of each layer. It is furnished in sections three feet long with canvas jacket, for both iron and copper pipes from ¼ in. through 24 in., and is regularly made in Double ½ in. and Double 3.4 in. thickness. Other thicknesses can be made special to order. Four inch (4 in.) wide collars for sealing end laps, as well as longitudinal flaps on outer water resistant felt will be furnished special on request.



PERFECTO WOOL FELT PIPE COVERING

Uses: Perfecto is an efficient economical pipe insulation for temperatures between 60 F and 225, keeping hot water hot and cold water cold, and preventing sweating of cold water service lines. For service water temperatures between 45 F and 60 F, integral jackets with properly sealed joints give adequate service for intermittent operation. For continuous operation a water vapor barrier integral jacket should be used. Perfecto is not designed for dual temperature systems.

Description: Perfecto is a semi-rigid structure made of multiple layers of indented "dry" insulating felt (wool felt) firmly cemented together. The inner surface is lined with asphalt-saturated organic felt. Carey Perfecto covering is furnished with a white wrapper under the canvas.

Perfecto is finished with 10.2-yards-per-pound cotton cloth jackets. Heavier canvas jackets can be furnished. Perfecto can be finished with roofing jackets, such as the Carey recommended jacket of 45-lb Surety Roofing.

Descriptive Bulletins and Catalog Pages on Carey Insulation Products are available at Carey District Offices. Application details and specifications can be obtained by writing INSULATION DIVISION, The Philip Carey Mfg. Company, Plymouth Meeting, Pennsylvania.

B. F. Goodrich

The B. F. Goodrich Company
574 Derby Place, Shelton, Conn.

B. F. Goodrich Molded Cell-Tite® Tubing for moisture-proof insulation

B. F. Goodrich Molded Cell-Tite tubing is made of tough, flexible rubber with uniform non-connecting cell structure. Especially efficient and convenient for pipe insulation, it will not absorb moisture nor support combustion. Insulation value is excellent.

Easy application

Wherever used, B. F. Goodrich Molded Cell-Tite may be easily secured with air-drying adhesive. Short lengths of tubing may be slipped on; longer ones may be slit lengthwise, snapped over pipe and adhesive applied to cut edges. These edges will not absorb moisture.

Low water absorption

B. F. Goodrich Molded Cell-Tite meets ASTM specifications. Weight gain is under 10 percent after 24 hours immersion in water; under 5 percent after 3 minutes additional atmospheric pressure.

Efficient insulation

Thermal conductivity of B. F. Goodrich Molded Cell-Tite is 0.28 at 75F with no deterioration in service.

Low shrinkage and aging

B. F. Goodrich Molded Cell-Tite has no appreciable change after 7 days at 200F. Resistance to weather-aging is good. Linear shrinkage, when aged 4 weeks at 158F is under 10 percent, most shrinkage occurring in the first week.

Wide range of density and firmness

Standard grades of B. F. Goodrich Molded Cell-Tite are ASTM 41 to 43 (from 5–13 psi to compress a one square inch disc to 75 percent of its height). Can be made in practically any firmness. Density varies, depending upon size and firmness, averaging 10 lb per cu ft for soft, 15–20 lb per cu ft for medium, and 20–30 lb per cu ft for firm.

Many sizes

B. F. Goodrich Molded Cell-Tite tubing is made with standard outside diameters of ¾ in., 1 in., 1¼ in., 1½ in., and can be made 1⅝ in., 1¾ in., and 2 in. Almost any inside diameter is available up to ¼ in. of the outside diameter. Lengths up to 50 ft facilitate ease and simplicity of installation.

Variety of colors

Black is standard and can be furnished as non-staining to lacquer, enamel or plastic. (Tested for 24 hrs under ultra-violet light or for 10 days at 158F.) Any one color available for requirements of 7,500 ft or more. White, if specified, would be off-white.

Please write for more information

HAGAN Mfg. Company, DELPHOS, OHIO

Thermal and Acoustical CELLULOSE FIBER INSULATION

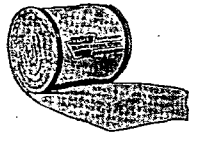
PREFERRED BY MANY in the ELECTRIC HEATING INDUSTRY

K-24 NATURE'S INSUL-FIBER



BLOWING WOOL

K-factor = 0.24 • Quality production control • Only first-grade cellulose material used • Pressure atomization process for consistent density and texture • Positive chemical control provides permanent protection • Permanent fire resistance, rodent and vermin resistant, harmless to handle • Expels moisture • Bonded material.



BLANKETS

K-factor = 0.24 • Light weight Resilient • Moisture Resistant Vermin Resistant • All std. sizes All std. moisture barrier configuration



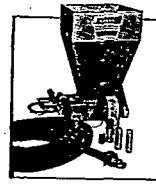
SILL PAD

Designed for easy application around high heat loss window and door framing—4 50 ft. rolls 6 in. wide to the carton.

WHY IT IS USED

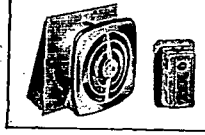
Insulated Surface	U Factor	Watts Factor
BLOWN MATERIAL		
Ceiling with 6"	.038	.011
Ceiling with 4"	.055	.016
Frame Walls with 3⅝"	.059	.017
BATT MATERIAL		
Knee Walls with 3⅝"	.063	.018
Double Wood Floor with 2"	.088	.027
Double Wood Floor with 3"	.065	.019

These coefficients, from HAGAN products, provide **LOWER** ELECTRIC HEAT OPERATING COSTS. Products proved in thousands of electrically heated homes and commercial buildings.



HAGAN MODEL J BLOWER — PORTABLE, 110 VOLT

- 1 H.P. Tornado Blower motor, ½ H.P. agitator motor, controlled air flow. Moves 25 bags per hour.
- Airtight shut-off nozzle.
- 20-yr. Hagan experience perfected this applicator package for cellulose material. Complete with hole saws, couplings, 75 ft. hose, nozzle, etc. Package complete.
- All wiring properly protected from damage due to rough handling.
- Operating instruction manual included.



RELATIVE HUMIDITY CONTROL SYSTEMS FOR IDEAL LIVING CLIMATE

1. Maintain safe, healthy humidity.
2. Prevent window steaming.
3. Prevent winter mildew.
4. Prevent wall and ceiling sweating.
5. Prevent paint peeling caused by inside moisture.
6. Allow the insulation to do a better job by keeping it dry.
7. The fan naturally expels cooking grease and holds down cleaning cost and effort.

JOHNS-MANVILLE

22 East 40th Street, New York 16, N. Y.

Offices in All Large Cities

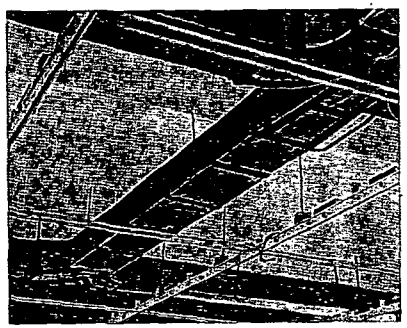
SPINTEX DUCT INSULATION

J-M Spintex®, a super-springy, spun fibre insulation, is an *excellent* insulation for air-conditioning ducts in offices, factories, institutions and commercial buildings of *all kinds*. Produced by a special Johns-Manville spinning process which assures complete uniformity, the fibres possess unusual fineness. This improved *fibrous structure* adds countless heat-blocking dead air spaces per cubic inch to help keep working temperatures precisely the same throughout the structure at *lower operating* cost.

Spintex will not burn or let fire spread along duct *systems*. The fibres are non-absorptive and treated to arrest capillary action, and will neither rot nor mildew. It is not corrosive to any common metal or *building* material.

Spintex can be installed quickly and easily because it is light and "friendly" to handle—simple to cut and fit, even when *ducts are curved* or in difficult-to-reach areas. It is available in rigid, semi-rigid or flexible form.

Spintex is regularly furnished in a variety of special facings to meet any vapor condition, incombustibility requirement or decorative *need*. It *provides* a good base for plaster finish . . . or where no facing is indicated, Spintex presents an attractive appearance with trim, tidy joints.



AIRACOUSTIC SHEETS

J-M Airacoustic® Sheets meet the exacting requirements of air duct sound conditioning. Since they absorb a large percentage of the sound that strikes them, they provide a highly effective means of reducing disturbing duct noises. Composed of mineral wool and a suitable binder, Airacoustic Sheets are completely non-combustible, highly sound-absorbent, moisture-repellent, odorless and vermin-proof. They are supplied 24 in. x 36 in. in ½ in. and 1 *in. thicknesses*.

SPINTEX HOME INSULATION



J-M Spintex® spun mineral fibre insulation is made by the revolutionary spinning process which produces finer fibres. Because of these finer fibres, *Spintex* has more heat-stopping dead air spaces per cubic inch. It is these dead air spaces and the thickness of the insulation which results in greater insulating efficiency.

Spintex will not rot or decay . . . is inherently fireproof. It will *not sag or settle* . . . the firmly felted fibres have the resilience to maintain a tight fit between framing members. It assures year-round comfort, protection and savings in fuel and air-conditioning costs.

J-M Spintex Batts and Blankets are fabricated in a variety of thicknesses to meet all requirements. They are light in weight, have greater rigidity and resilience for ease of handling and tight fit, cut clean and sharp *for better, faster, labor-saving* application. Each has an efficient vapor barrier and is carefully manufactured to meet *Federal Specification HH-I-521C*. Blankets are available either Kraft paper-wrapped or aluminum-wrapped.

J-M Spintex Home Insulation is also available in "blown" (nodulated) form for new and existing light construction.

Details on Request

For details on J-M Spintex Home Insulation or the complete line of J-M *Industrial Insulations*, *write* Johns-Manville, Box 14, New York 16, N. Y.

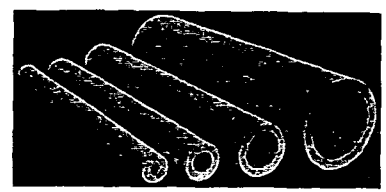
JOHNS-MANVILLE

22 East 40th Street, New York 16, N. Y.

Offices in All Large Cities

JOHNS-MANVILLE INSULATIONS
For Heating and Plumbing Service

FIBROCEL PIPE INSULATION
for service from 35F to 300F



Fibrocel® is a molded silica insulation developed by Johns-Manville for use on hot, cold and chilled water lines, low pressure steam pipes and dual service heating and cooling systems.

No Shrinkage The use of Fibrocel makes objectionable shrinkage negligible. It is not affected by changing atmospheric conditions. Joints stay tight . . . no unsightly gaps with accompanying heat leakage or costly repairs.

Firm, Strong Structure Fibrocel resists compression and abuse. It gives back-up protection against puncture of jacket or vapor seal. Its smooth hard surface takes abuse in rough *service*.

High Thermal Effectiveness Tiny particles of silica are exploded and expanded to form a cellular structure of heat-trapping dead *air spaces*.

Permanent and Sanitary Fibrocel is impervious to flame, rot, odor and vermin attack. It will not disintegrate in water; original properties are restored after drying.

Easy to Install Fibrocel is light and easy to handle, it readily cuts with a knife. Tight, snug-fitting edges provide better heat control, smooth appearance.

Jacket Styles Type VS—richly creped pure white kraft backed by ½ mil-foil. Type VB—outer surface has same rich creping, inner barrier is full mil thickness of aluminum foil; pipe facing is creped black kraft. Type C—canvas finish.

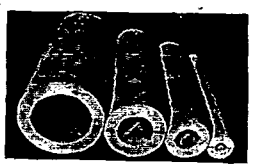
AEROTUBE PIPE INSULATION
for service from 32F to 200F

Aerotube® is a foamed plastic tubular insulation expressly recommended for heating, plumbing and *air-conditioning* service. Its closed cell structure acts as a "built-in vapor barrier" to stop the passage of moisture . . . obviates the necessity of applying a separate vapor barrier. Its extreme flexibility and resiliency permit quick, easy application, especially on bent tubing and fittings. Aerotube is clean, odorless, durable . . . does not carry flame and is self-extinguishing.

Aerotube is furnished in 6 ft lengths in the following iron pipe and copper tubing sizes: ⅜ in. thick—¼ in. through 2 in.; ½ in. thick—¼ in. through 3 in. It is also available in sheet form for flat surfaces and equipment not adaptable to pipe insulation.



J-M THERMOBESTOS®
For Service to 1200F



Thermobestos is a calcium silicate in*sulation developed for the most rugged industrial service. Strong and rigid,* it will withstand unusual service abuse. It offers maximum moisture-resistance and low thermal conductivity, will not burn or carry flame, and is light in weight for easy handling and fast application. *Pipe insulation furnished in 3 ft sections in nominal thicknesses of 1 in.* (up to 6 in. pipe size only), 1¼, 2, 2½, 3 and 3½ in. Block insulation in 36 in. lengths, 6 and 12 in. widths, 1 in. through 4 in. thicknesses.

Other J-M Insulations

Product	Description	Standard Sizes
*J-M 85 percent Magnesia For service to 600F	Made from hydrated basic carbonate of magnesia bonded with asbestos fibre. Light weight, high insulating value, easily worked.	Pipe ins.—3 ft sections or segments, nom. thicknesses of 1 in. through 3 in. Block—12 and 36 in. lengths; 3, 6, 9, 12 in. widths, 1 through 4 in. thicknesses.
*Asbestocel® For service to 300F	Cellular type insulation for use on pipes conveying low pressure steam and hot water.	3 ft sections in standard thicknesses of 2 to 8 plies, each ply approx ¼ in. thick.
*Rock Cork® For service from −300F to +200F	Moisture-resistant insulation for all refrigeration applications. Sanitary, odorless, fire-resistant.	Sheets: 18 in. x 36 in. thicknesses to 4 in. Pipe Ins.—3 ft sections in Ice Water, Brine, Heavy Brine thicknesses.
J-M No. 201 Insulating Cement	Hydraulic-setting, one-coat application cement for insulating small pipe fittings. Covering capacity, 36 bd ft per 100 lbs.	50-lb bags
J-M No. 450 Insulating Cement	Mineral wool cement with excellent adhesion to either hot or cold surfaces. Covering capacity, 50 lb ft per 100 lbs.	50-lb bags
J-M No. 352 Insulating Cement	Asbestos cement for insulating pipe fittings or for use as finish over other insulation. Covering capacity, 19 bd ft per 100 lbs.	50 and 100-lb bags

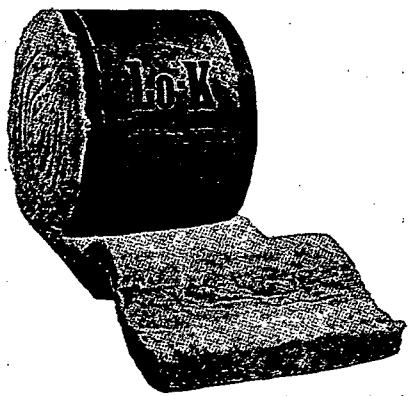
* Also available to fit straight runs of copper tubing with nominal diameters from ⅜ in. up.

Lockport Mills, Inc.

Dept. C Lockport, N.Y.

Thermal and Acoustical INSULATION

Lo-K"



Five featured types to meet every insulation need: (1) **OPEN BLANKET** backed by tough, waterproof, asphalt-coated kraft paper to form an effective vapor barrier. (2) **ENCLOSED BLANKET.** Insulation completely enclosed in envelope, asphalt-coated paper on one side, porous or "breather" type paper on other. (3) **OPEN ALUMINUM FOIL.** All the features of Open Type (1) plus the extra value of aluminum foil backing. Effective vapor barrier ... stops 90 percent of radiant heat. (4) **ENCLOSED ALUMINUM FOIL.** Superior in insulation plus values and thermal efficiency. (5) **DOUBLE FOIL,** aluminum foil on top and back, plus superior Lo-K Blanket efficiency for tops in insulating value.

Lockport's TEMP-LO

A New Type of *Semi-Rigid* Insulation

Improved processing makes possible a new and better semi-rigid blanket with greater thermal efficiency and ease of installation. It is light in weight, nonirritating and made in all standard sizes, styles and backings.

Distributing agencies now being established. Write for details.

Thermal Conductivity—The "k" value for cotton is 0.24 Btu/hr/sq ft/degree F/inch. (*See table.*)

Light Weight—Weight of 1 cu ft is ⅞ lb. (*See table.*)

Flame-Proofed—Withstands 1800F blow torch heat.

Moisture-Resistant—Chemical treatment, combined with natural protective coating on cotton fibres, enables cotton to effectively resist moisture. Prevents rot and mildew.

Smooth Texture—No sharp particles to irritate the skin.

Easy to Warehouse and Handle. Offers far more "compressibility." Requires one-third the trucking and warehouse space of ordinary insulation.

Simple and Economical to Install. Saves from 25 *to 40 percent in costs.*

Designed to Maintain Maximum Utility. Resists all types of deterioration. Won't sag or settle. Packaged in Rolls.

INSULATING VALUE OF VARIOUS INSULATORS[a]

The coefficients of conductivity (*k* value) are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. of thickness.

Type of Insulation	Wgt. per Cu Ft	k[b] Value
Cotton: Insulating Batt	.875	0.24
Rock Wool: Fibrous material made from rock	10.00	0.27
Mineral Wool: Fibrous material made from mineral slag	.....	0.27
Glass Wool: Fibrous material made from glass slag	1.50	0.27
Rigid Insulation made from sugar cane fibre	13.50	0.33
Chemically treated wood fibre between layers of paper	3.62	0.25
Eel grass between layers of paper	3.40	0.25
Stitched and creped expanding fibrous blanket	1.50	0.27
Shavings: Various from planer	8.80	0.41
Corkboard: No binder added	7.00	0.27
Rigid insulation made from wood fibre	15.90	0.33

[a] Compiled from Chapter 9, 1954 Edition.
[b] "*k*" indicates temperature conductivity.

BW ALFOL

Reflectal Corporation

A subsidiary of Borg-Warner Corporation
200 S. Michigan Ave., Chicago 4, Illinois

UNIQUE REFLECTIVE INSULATION

What ALFOL is: ALFOL is a reflective-type insulation consisting of multiple aluminum foil layers in handy blanket form. Designed to combine high thermal efficiency with positive vapor control, ALFOL is ideal for all types of residential, commercial and industrial buildings.

How ALFOL works: Utilizing the reflective insulating principle, the patented ALFOL blanket effectively resists all 3 forms of heat transfer. (1) Self-spacing sheets of pure aluminum foil reflect 95 percent of all radiant heat. (2) Convection is blocked by these same foil layers. (3) Multiple "captive" air spaces hold conduction to the minimum. The result is a combination of benefits that is totally unique.

PRINCIPAL ADVANTAGES OF ALFOL

High year-round efficiency: In resisting up-flow (winter) and horizontal heat transfer, ALFOL efficiency is comparable to that of normal bulk insulation. In resisting down-flow (summer) heat transfer, ALFOL is decidedly superior. The result is an overall efficiency profile that is truly outstanding. NOTE: See Chapter 9, Table 3 for characteristics of aluminum insulation.

Extra air-conditioning economy: Low heat storage capacity, combined with the decided summer advantage cited above affords substantial savings in air conditioning.

Reliable condensation control: A separate vapor barrier (consisting of polyethylene lining, foil or duplex-backed kraft) is a component feature of all ALFOL types. This positive, continuous vapor barrier guards against condensation.

Rapid, positive application: Self-spacing action of the unusual ALFOL blanket makes installation virtually fool-proof, and its speed of application means low installed cost.

Additional advantages: ALFOL is clean, permanent, odorless ... and completely uniform in quality. With 9 ALFOL Types to choose from, you specify what you need ... and get what you specify.

AN ALFOL TYPE FOR EVERY PURPOSE

(A) Standard ALFOL Types: for normal insulating requirements. Available in various combinations to provide 2, 3 or 4 reflective air spaces. Designations: Type 1, Type 1A, Type 2, Type 3, Type 4.

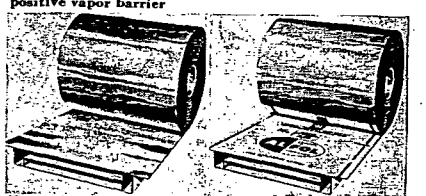
(B) Heavy-duty ALFOL Types: for utmost insulating performance. Similar to Standard Types, but with heavy reinforcing for extra rigidity and "box-type" end-design for full edge-to-edge coverage. Designations: Types 22F and 22 (with polyethylene backing), Types 44F and 44.

All ALFOL Types are available in widths to accommodate 12-in., 16-in., 20-in. and/or 24-in. framing centers. Blankets are packaged in continuous-length rolls of 500 or 250 sq ft.

FREE Data Book. For complete technical data and specifications, write for 28-page ALFOL Data Book. This colorful manual is available free to architects and engineers.



Standard ALFOL blanket **is available in 5 types for varying thermal needs. Type 2 (*shown*) has 2 aluminum foil sheets that expand automatically to deliver 3 reflective air spaces. Duplex-lined kraft backing provides positive vapor barrier**



Heavy-duty ALFOL **blanket is heavily reinforced, has "box-type" design for edge-to-edge fit. Type 44F (*left*) provides 4 reflective air spaces for maximum performance. Type 22F (*right*) features polyethylene lining for extra condensation safety**



Ideal for *sidewall* installations, ALFOL combines high thermal value with the vital protection of a positive, continuous vapor barrier.



Installed (*shown here*) in residential ceiling, ALFOL provides normal efficiency in winter, decidedly superior *efficiency in summer.*



Industrial "exposed ceiling" installation of ALFOL. Here, Type 44F (shown) provides full insulation plus a semi-finished ceiling.

American Flange & Manufacturing Co. Inc.

30 Rockefeller Plaza, New York 20, N. Y.

Plaza 7-2200

Ferro-Therm

Reg. U. S. Pat. Off.

METAL INSULATION

FULLY PROTECTED BY U. S. AND FOREIGN PATENTS



Ferro-Therm is stapled permanently in place

Ferro-Therm Reflects 90–95 percent of All Radiant Heat

Ferro-Therm Metal Insulation, available from sheets or coils of aluminum or steel with a special alloy coating, reflects 90 percent to 95 percent of all radiant heat. This high reflectivity, combined with extremely low heat storage capacity, provides maximum insulating efficiency in a minimum overall thickness.

Reflects Heat From Either Side

Ferro-Therm's high reflectivity enables it to resist the penetration of heat from either side. During the winter it reflects heat in; during the summer it reflects heat out—assuring year 'round comfort and a considerable saving in fuel.

Reduces Fuel Costs 25–30 percent

Installed in ceiling (or roof) and exterior walls, Ferro-Therm reduces fuel costs by an average of 25 percent to 30 percent. In ceiling (or roof) alone, Ferro-Therm will reduce fuel costs by 15 percent to 20 percent. During the summer, the installation of Ferro-Therm metal insulation will reduce temperatures in the house by 10 F. to 12 F.

Gives the Protection of Metal

Because Ferro-Therm is metal (1) It is not only noncombustible; it is an effective fire stop for wooden framework; (2) It prevents the penetration of termites, rodents and insects; (3) It does not absorb moisture or convey any wood-rotting moisture in framing members; (4) It does not settle or pack down, as the sheets are stapled permanently in place; (5) It is absolutely permanent.

Ferro-Therm Is Easy To Install— and Economical

Ferro-Therm takes up considerably less space than mass insulation of equivalent efficiency. Laboratory tests and thousands of applications have demonstrated that a wall of Ferro-Therm will provide insulating efficiency equivalent to mass insulation approximately twice as thick. And Ferro-Therm can be installed easily, quickly and economically. The sheets take up considerably less space than mass insulation. And they are light—easy to transport and handle. One man can install 1000 to 1200 sq ft of Ferro-Therm in a day.

INFRA INSULATION, INC.

525 Broadway, New York 12, N. Y.—WOrth 6-4314

This insulation has

7 3/4 INCHES non-metallic insulation value in **WINTER - U .04 - C .043 - R (1/c) 23.32 heat flow UP**

11 1/4 INCHES non-metallic insulation value in {SUMMER / WINTER} **- U .03 - C .029 - R (1/c) 34.16 heat flow DOWN**

9 Aluminum ¾" Heat-Resisting Spaces, Tough, Non-Perforated Aluminum Laminates Top and Bottom

TYPE 9-LL

Scientific Infra Insulation insulates against *Heat, Vapor, Dripping Condensation, Timber* Rot, Wet Plaster, Peeling Paint, Discomfort

Recent *research by the National Bureau of Standards* reveals how multiple layers of aluminum and air spaces drastically reduce Heat-Flow by Convection-Conduction in addition to Radiation.

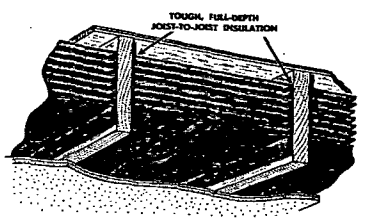


CHART OF SOME INFRA INSULATIONS—Performances and Prices

NO. OF *SPACES	TYPE	ILLUSTRATION	PRICE PER SQ. FT.	DIRECTION OF HEAT FLOW	U (Btu)	C (Btu)	†R (1/c)	Non-Metallic Insulation Equivalents in Inches‡
6	6-LL		7¢	UP	.06	.068	14.64	4½"
				DOWN	.03	.034	29.46	9¾"
4	4-LL		5¢	UP	.08	.105	9.52	3½"
				DOWN	.03	.038	26.18	8¾"
				WALL	.06	.068	14.65	4½"
3	3-LL		4¢	UP	.10	.143	7.00	2½"
				DOWN	.04	.046	21.79	7½"
				WALL	.08	.101	9.92	3¼"
8	8-LL		10¢	UP	.04	.049	20.47	6½"
				DOWN	.03	.030	32.99	10"
7	7-AL		8¢	UP	.05	.057	17.58	5½"
				DOWN	.03	.032	31.28	10½"
5	5-AL		6¢	UP	.07	.082	12.13	4"
				DOWN	.03	.034	29.00	9⅜"

‡Each space is reflective, & parallel, with at least 1 surface of aluminum foil of 3% emittance. †Calculated on basis of limiting thermal values cited in Fed. Specs. LLL-I-521b; HH-I-585; HH-I-521c; HH-I-551a. *Determined by method of National Bureau of Standards in H.H.F.A. Research Paper 32. ●Average installation rate is 2000 sq. ft. per day per man.

Scientific construction of multiple layers of aluminum, fiber and air spaces minimizes condensation on or within the insulation. There is slight capacity for absorbing water. These insulations, with a long and continuous (up to 500 ft.) non-porous vapor and water barrier on top as well as on bottom, made of genuine non-perforated metal of almost zero permeability, help protect against *dripping* condensation from above if any, timber rot, peeling paint, wet plaster, crumbling masonry, rust and insects.

For additional information and a free copy of Housing Research Paper 32, write to Infra Insulation Inc., Dept. G-9, 525 Broadway, N. Y. 12, N. Y.

Reflectal Corporation

A subsidiary of Borg-Warner Corporation
200 S. Michigan Ave., Chicago 4, Illinois

FOR UNDEVIATING CONTROL OF SOLAR HEAT

What KoolShade is: KoolShade Sunscreen is a sun control device consisting of tiny horizontal louvers permanently interwoven with tough vertical warp wires (see illustration A). Functional and attractive, the result is a durable solar screening that combines optimum shading efficiency . . . with minimum possible obstruction of light, air and view.

How KoolShade works: Operating like a tiny venetian blind installed outside the window, KoolShade intercepts the sun's rays *before* they reach the glass. Illustration B shows how its tiny louvers (spaced 17 or 23 to the inch, depending on the type) screen out up to 100 percent of the direct solar rays.

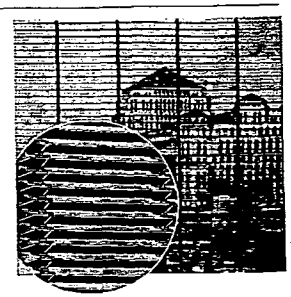
PRINCIPAL ADVANTAGES OF KOOLSHADE

Reduces air conditioning costs: The ideal adjunct to air conditioning, KoolShade can sharply reduce initial tonnage requirements. (By rule of thumb, each 100 sq ft of KoolShade replaces the need for about 1 ton of cooling capacity . . . at substantial savings in cost.) Also, KoolShade increases the efficiency of the cooling unit . . . hence cuts operating and maintenance costs.

Balances winter heating load: By eliminating solar "hot spots" (a major cause of system imbalance), KoolShade promotes more efficient winter heating. Cuts costs by reducing the need for excessive adjustment of thermostats.

Eliminates eye-straining glare: Widely used for daylighting control, KoolShade moderates sun and sky glare to improve lighting balance, minimize fall-off . . . guard against eyestrain. Yet the extra open area afforded by KoolShade's exclusive woven design admits ample light and air . . . and 83 percent clear outward visibility.

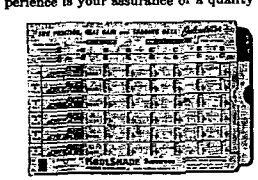
Weather resistant . . . never needs painting: Electrostatically coated with weather-proof nubelon enamel, KoolShade requires virtually no maintenance—less by far than ordinary insect controls. And it enhances architectural appearance.



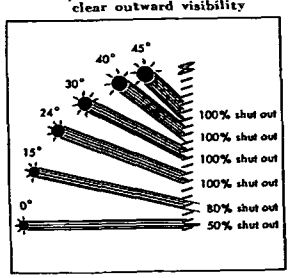
(A) Magnified view (*inset*) shows how KoolShade louvers are literally woven in place. Background photo (screen shown actual size) demonstrates 83 percent clear outward visibility

KOOLSHADE FRAMING AND INSTALLATION

For peak efficiency, durability and appearance, KoolShade is installed in KoolFrame aluminum extrusions designed exclusively for use with KoolShade. Custom-fabricated to accommodate fenestrations of any type or shape, KoolShade screens are sold and installed only by Franchised KoolShade Distributors. Their specialized experience is your assurance of a quality installation every time.



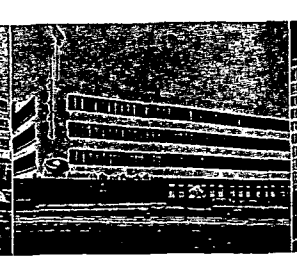
FREE Shading Data Calculator available on request to engineers and architects. Convenient slide chart—with handy inserts for 4 different latitudes—provides sun position, heat gain and shading data at a glance for any window orientation, any season. An invaluable tool for figuring solar heat loads



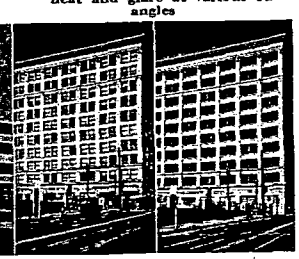
(B) Side view of louvers (Type LSA, magnified) illustrates KoolShade efficiency in screening out solar heat and glare at various sun angles



Air conditioned, Savannah (Ga.) Memorial Hospital reduced cooling load by 74 tons with KoolShade. Net savings in cooling tonnage alone amounted to $50,300



Installed after air conditioning, KoolShade was used at North American Life Building (Minneapolis) to eliminate solar "hot spots," cut operating costs



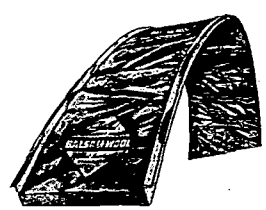
Used as adjunct to air conditioning at Los Angeles County Engineers Building, KoolShade reduced solar heat load 72 tons, saved $13,300 in equipment

WOOD CONVERSION COMPANY

Dept. 220-9, First National Bank Building, St. Paul 1, Minnesota

New York Chicago Boston Kansas City Detroit Atlanta
Buffalo St. Paul Minneapolis San Francisco

For many years a leader in the insulation field, Wood Conversion Company manufactures a complete line of flexible fiber and rigid insulation for industrial and domestic purposes. Basic product categories include Balsam-Wool blanket insulation, Nu-Wood structural insulation and Nu-Wood interior finish board. Balsam-Wool insulation is the product of scientific research, and is especially designed for insulating efficiency and positive application.



BALSAM-WOOL® BLANKET INSULATION

WITH REFLECTIVE LINERS—The wood fiber insulating mat of this newest Balsam-Wool insulation is bonded with asphalt to reflective liners. Both reflective surfaces reduce heat transfer by radiation across the wall, floor, or ceiling air space. Balsam-Wool thus reduces heat transmission three ways by conduction, convection and radiation. Application spacer flanges assure an effective insulating air space or reflective surface on each side of the Balsam-Wool blanket.

WITH ASPHALT LINERS—Regular Balsam-Wool is identical to the reflective liner type except that liners are formed of asphalt saturated and coated kraft. The warm side barrier protects the wood fiber insulating mat from condensing moisture; cold side liner reduces convection through the insulating mat and reduces air infiltration. Balsam-Wool insulating mat is uniform in thickness. Proper application is assured by built-in spacer flanges along blanket edge which are stapled to framing members.

PRODUCT ADVANTAGES

- Integral continuous vapor barrier.
- Provides air spaces on both sides of blanket.
- Fire-resistant insulating mat.
- Low thermal conductivity-value.
- Resists settling or shakedown.
- Blanket and air spaces correctly positioned by spacer flanges.
- Wind-resistant liners.
- Rugged product construction . . . stands up under rough handling.

NU-WOOD® INSULATING BOARD PRODUCTS

NU-WOOD SHEATHING—A strong, structural insulating sheathing in ½-in. and $^{25}/_{32}$-in. thicknesses; asphalt impregnated for moisture protection.

NU-WOOD ROOF INSULATION—Will fill all requirements of federal specification LLL-F-32 lb. Furnished in various thicknesses in plain and asphalt impregnated types.

SIZE DATA

BALSAM-WOOL	Reflective Liners—Standard (1 in.) and Double-Thick (2 in.)—16 in. and 24 in.
	Regular Liners—Standard (1 in.) and Double-Thick (2 in.)—12 in., 16 in., 20 in. and 24 in.
	Regular Liners—Type E (Economy) ($^{5}/_{8}$ in.)—12 in. and 16 in.
NU-WOOD SHEATHING	$^{25}/_{32}$ in. Thick—2 ft. x 8 ft. Shiplap Edge; 4 ft. x 8 ft., 9 ft. Square Edge
	½ in. Thick—2 ft. x 8 ft. "V" Joint Edge; 4 ft. x 8 ft., 9 ft. Square Edge
NU-WOOD ROOF INSULATION	½ in., 1 in., 1½ in. and 2 in. Thicknesses—24 in. x 48 in. or 23 in. x 47 in.

April Showers Company, Inc.

5980-88 Sligo Mill Road, N.E. **Washington 11, D. C.**

April Showers — LIGHT AS RAIN

PATENTED AUTOMATIC ROOF COOLING

(Trade Mark Reg. U. S. Pat. Off.)

Distributors and Dealers in Principal Cities

AUTOMATIC EVAPORATIVE ROOF COOLING

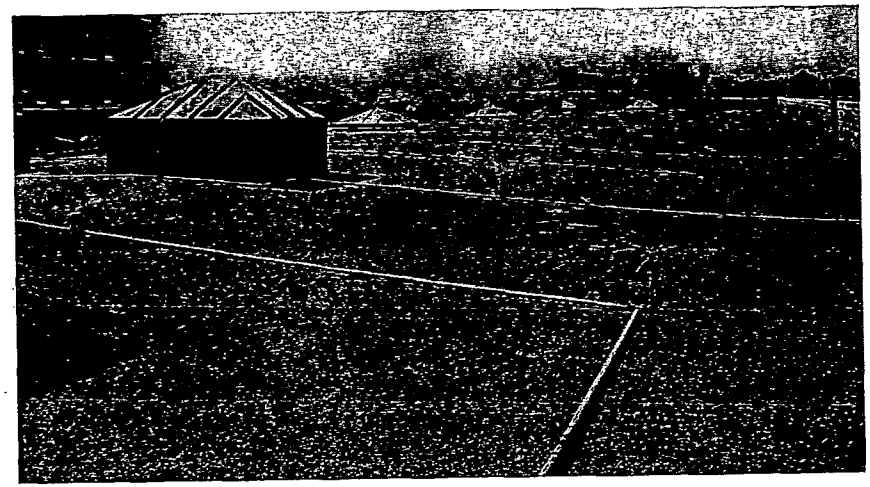


Illustration shows typical April Showers installation

AUTOMATIC EVAPORATIVE ROOF COOLING SYSTEMS

Spray Method

The April Showers Automatic Roof Cooling Systems are designed to stop solar heat at its source by evaporative cooling. They have been used extensively since 1934 for under-roof cooling of industrial and commercial buildings or to reduce initial and operating costs where a true air conditioning system is contemplated or in use. In the latter case, total tonage requirements are substantially reduced and operating costs cut an average of 25 percent.

April Showers Systems are completely automatic. The control thermostat is set to turn the system on at about 10 degrees above designed wet bulb temperature. A four degree drop shuts off the spray, holding the roof surface temperature at about 8 degrees above wet bulb temperature. This lowers under-roof temperatures 8 to 15 degrees. Not only is the roof temperature affected, but all supporting upright members and walls since the heavy, cool air spills down over them thus cooling them, too.

It is important to note that the April Showers Spray-head is scientifically designed to produce a fine DROP-LET—not a mist. This produces a fine film of moisture on the roof which permits excellent evaporation; the action that creates lowered temperatures.

April Showers operate on average city water pressure or utilize well or condenser waste water. Consumption is low, each Spray-head uses about 0.4 gpm and covers about 200 square feet. They operate an average of one minute in five; usually 15 or 20 seconds per spray period.

Today, there are over 25,000,000 square feet of roofs that are cooled by the April Showers Automatic Spray Method Systems. They are operating on many of the country's leading industrial plants and commercial buildings with gratifying results.

April Showers fit all shapes and sizes of roofs, help prevent roof fires, act as lightening arrestors, add longer life to composition roofs and there is no run-off. Estimates are made without charge; however, certain pertinent data is necessary to establish the factors that govern each individual installation. Ask or write for *Form E-102*. Our Engineering Staff stand ready to give you every assistance.

Patent Nos. 2,069,150—2,266,321—2,554,409. Other Pats. Pending.

FANJET DIV. - Muellermist Irrigation Co.

2616 S. Ninth Ave., Maywood, Illinois

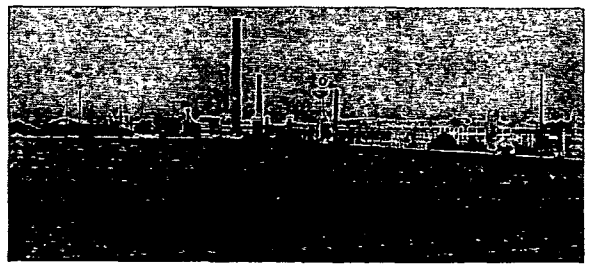
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FANJET "patented" Roof Cooling

Patent No. 2,834,635

ESTIMATES in 24 HOURS

PROPOSALS in 48 HOURS



PROTECT your CLIENTS with these GUARANTEES of the FANJET *system*

FANJET—The certified automatic Thermo-Controlled Non-Clogging Roof Cooling System

(Approved by Leading Roofing Companies)

—Delivered anywhere in the U. S. or abroad—

Reduces Inside Temp 8–15 Degrees.
Reduces Roof Temp 40–60 Degrees.
Reduces Air Cond Operating Cost.
Reduces Air Cond Initial Cost.
Reduces Personnel Turnover.
Reduces Production Rejects.
Increases Employee Efficiency.
Increases Life Of Air Cond. System.
Lessens Radiant Heat Rays.
Lessens Solar Heat.
Eliminates Pipe Clogging.
Increases Roof Life.
Lessens Roof Dust.
Provides Cooled Area Over Workers.
Acts As Lightning Arrester.
Prevents Thunderstorm Shock To Roof.
Promotes Good-Will Towards Management.
Absorbs Internal Heat Up Through Roof.

GUARANTEED Protection For The User.
GUARANTEED Spray Orifices (**NON-CLOG**).
GUARANTEED Workmanship (FIVE Years).
GUARANTEED Intermittent Operation.
GUARANTEED Materials (FIVE Years).
GUARANTEED **UNION** Manufactured.
GUARANTEED Non-Ferrous Materials.
GUARANTEED No Water Wastage Or Runoff.
GUARANTEED Spray Coverage.
GUARANTEED To Use **HARD** Water.
GUARANTEED Trouble-Free.
GUARANTEED To Operate On 20 psi.
GUARANTEED To Have Pipe Supports.
GUARANTEED Sweat Fitted.
GUARANTEED Line Drainage.
WARRANTY Of Manufacturers For Thermostats, Humidity And Timing Devices.

CERTIFIED PLAN with every PROPOSAL AND ESTIMATE

Some USERS: TUNG-SOL ELEC.—FEDERAL TELECOM. LAB.—LOCKFORMER CO.—MARCAL CO.—LAVEZZI MACH. WORKS—CARTER PRODUCTS CO.—SOLA ELECTRIC CO.—AMERICAN NAMEPLATE CO.—OKLAHOMA TIRE & SUPPLY CO.—KAILER YOUNGQUIST CO.

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THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

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To Get the Facts

about heating and fuel requirements, heating costs, operating and maintenance problems, the equipment situation, personnel, performance standards, fuel economics, sales and service—this is what we are trying to do, here at **COAL-HEAT:**

These are the fundamental factors that affect the market, the use and sale of fuel and equipment, one's job, or business.

So, it is our aim to get and print "the low-down" on these in each of the following markets:

1. **Apartment houses**
2. **Churches**
3. **Dept. & other stores**
4. **Dairies**
5. **Greenhouses**
6. **Hotels**
7. **Hospitals**
8. **Institutions**
9. **Laundries**
10. **Municipal & govt. bldgs.**
11. **Mfg. plants**
12. **Office bldgs.**
13. **Residential**
14. **Schools**
15. **Theaters & other bldgs.**

To find out what each of us concerned should know about the particular needs, problems and possibilities in each of these fields, and to help make more of the facts known where they will do the most good—this is our purpose!

Together, these groups represent some. 20-million plants, buildings, homes, 160-million tons of coal annually—the nation's largest fuel market, in either number of tons or customers. See "Heating-Fuel Markets"—$1.00 a copy.

Many of these plants and buildings are old, the equipment is in poor condition, operation leaves much to be desired, costs are excessive.

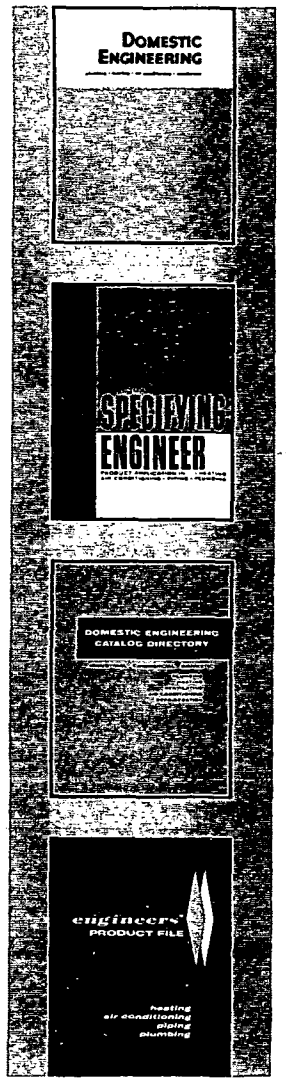
Since most janitors, building owners, members of school and hospital boards are not well informed about heating or the proper use of fuel, we face a tremendous educational job—a job that rests largely on the retail dealer and the salesmen because they alone come in periodic contact with these 20-million fuel users. To help provide the essential educational sales and service tools—that's what we're trying to do.

To help keep more customers happy, get into more of the new buildings, help develop new business—these are the objectives of **COAL-HEAT.** It can help you get more business!

FUEL AND HEATING EQUIPMENT—SALES AND SERVICE

DOMESTIC ENGINEERING COMPANY

1801 Prairie Avenue, Chicago 16, Illinois



DOMESTIC ENGINEERING MAGAZINE

Editorially keyed to the total business interests of qualified plumbing-heating-cooling contractor-dealers who specify, *sell, install and service. DE covers every* phase of the contractor-dealer's business: engineering, servicing, merchandising, research, financing, new products, legal, news, legislation. Winner of 16 out of the 17 national editorial awards presented to publications in its field by Industrial Marketing since 1939, DE holds the loyalty and confidence of "buying core" contractors who do more than two-thirds of the industry's total annual volume—loyalty and confidence that pays dividends to Domestic Engineering *advertisers. Member,* Audit Bureau of Circulations. Published monthly, $5.00 per year.

ACTUAL SPECIFYING ENGINEER

The only business publication written specifically for the engineer who actually specifies products into heating, piping, plumbing, ventilating and air conditioning systems. Editorial formula and performance are completely geared to the subject of specifications and product choices and to the application of products in various systems. Circulation is *made up of men who design systems* and specify products, and includes: independent consulting engineers specializing in mechanical systems; consulting engineers; engineers associated with architects; and specifying engineers associated with large industrial companies, institutions and commercial chains, big contractors and distributors, government (in all categories), utilities, etc. Published monthly, $5.00 per year.

DOMESTIC ENGINEERING CATALOG DIRECTORY

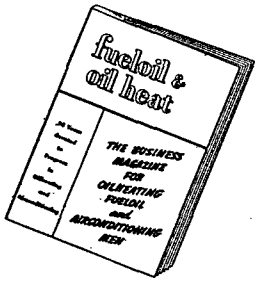
Published annually as a buyers' guide for large contractor-dealers and wholesalers. Contains 6 sections of MANUFACTURERS' CATALOGS on: Boilers, Burners, Furnaces, Controls, Air Conditioning, Refrigeration, Insulation, Pipe, Valves & Fittings, Pumps, Tanks, Appliances, Water Heaters & Softeners, Brass, Drainage Pipe, Tools, Compounds and Oil. PLUS . . . CLASSIFIED DIRECTORY OF PRODUCTS with 3,000 main classifications and 4,600 cross references. TRADE NAMES, an extensive list of over 10,000 with *corresponding manu*facturers' names. NAMES AND ADDRESSES of approximately 5,000 manufacturers in the Industry. TECHNICAL ENGINEERING DATA, complete array of easy-to-read, easy-to-use charts, graphs and tables. $12.50 per year. $25.00 for three years.

ENGINEERS' PRODUCT FILE

Published annually for engineers who *specify* Heating-Cooling-Plumbing. Contains 8 sections MANUFACTURERS' CATALOGS: Heating, Air Conditioning, Ventilating; Heating-Cooling Controls, Specialties; Tanks, Heat Transfer Equipment, Water Heaters, Water Treatment; Piping, Tubing, Valves, Fittings, Fire Protection Equipment, Pipe Hangers, Pipe Insulation; Pumps, Lawn Sprinkler Systems; Plumbing Fixtures, Accessories, Plumbers' Brass, Specialties; Plumbing Drainage Products. CLASSIFIED DIRECTORY of thousands of products, names, location of manufacturers. TRADE NAMES, lists thousands with product, manufacturer. MANUFACTURERS' NAMES, ADDRESSES, thousands whose products are specified. TECHNICAL, ENGINEERING DATA, 160 pages, basic tables, rules commonly used in systems layout, design.

Heating Publishers, Inc.

2 West 45th Street
New York 36, N. Y.



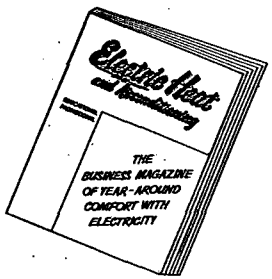
FUELOIL & OIL HEAT

Now in its 37th year of publication; is edited for all oilheating and airconditioning men. Covers installing and servicing of all oilheating and airconditioning equipment, selling and merchandising from the dealer level. Technical articles, exclusive market reports on oilheating, airconditioning and fuel oil. The regular monthly *Commercial-Industrial Section* covers installing and servicing of heavy oilheating equipment, boilers, large capacity oil-fired air movers and heavy fuel oil. Member Audit Bureau of Circulations and Business Publications Audit of Circulation. Published monthly. One year $3.00—3 years $5.00.

GAS HEAT AND COMFORT COOLING



The complete heating and cooling magazine covering the use of equipment which uses gas as a fuel exclusively. Covers all forms of comfort heating and airconditioning; commercial-industrial, residential, etc. Technical articles, sales and merchandising, exclusive monthly report on gas heating and airconditioning retail equipment sales. Covers all equipment using gas for a fuel, airconditioning, water heaters, incinerators, furnaces, boilers, conversion burners. Edited for the retailer-installer, serviceman, contractor. Many readers are utilization engineers with gas utilities. Now in its 9th year. Member of Business Publications Audit of Circulation. Published Monthly. One year $3.00—3 years $5.00.



ELECTRIC HEAT & AIRCONDITIONING

A new magazine devoted to a relatively new art—comfort heating with electricity. Covers both residential and commercial applications. Electric heating is coming fast, particularly in areas where the air conditioning load has given the electric utilities a very large, peak summer load. When this peak presents a non-profit imbalance, the utilities create comfort heating rates and push hard for electric heating to build a winter load. It has zone control possibilities. In some areas insulation is recommended to make electric heating competitive. **Electric Heat & Airconditioning** covers all phases of this growing industry including heat pumps. No engineer in heating and air conditioning should ignore this new art. This is the only publication 100 percent devoted to electric heating. Member of the Business Publications Audit of Circulation. One year $2.00—3 years $5.00.

BOOKS OF PARTICULAR VALUE:

"Fundamentals of Year-Round Air Conditioning Controls with Oil Heating" ($2.00 a copy).
"Summer-Winter Air Conditioning Controls for Gas Heat & Comfort Cooling ($2.00 a copy).

"Modern Gas Heating"

The first compilation of information devoted to gas heating installation and service work. Close to 100 pages with many illustrations—$3.00 a copy.

"Better Oil Heating"

A service guide for operation and maintenance of oil burners. Bound in attractive paper cover 8¼ × 11. Nearly 100 pages, 125 illustrations, tables and service hints. $2.00 a copy.

International Heating & Air-Conditioning Exposition

Permanent Address—480 Lexington Ave., New York 17, N. Y.

EXPOSITIONS HELD

Philadelphia, 1959; Chicago, 1957; Philadelphia, 1955; Chicago, 1953; Philadelphia, 1951; Chicago, 1949; New York, 1948; Cleveland, 1947–1940; New York, 1938; Chicago, 1936; New York, 1934; Cleveland, 1932; Philadelphia, 1930.

FUTURE SCHEDULE

2nd Southwest Heating & Air-Conditioning Exposition in Dallas—1960; scheduled for Chicago in 1961.

UNDER AUSPICES OF A.S.H.A.E.

These Expositions have been and will be held coincident with the annual meetings of American Society of Heating and Air-Conditioning Engineers, Inc., and under their auspices. Management is by International Exposition Company with permanent headquarters at 480 Lexington Ave., New York 17, N. Y.

EXHIBITORS

Comprise leading firms in each phase of the industry; number has varied from 150 to more than 400 exhibitors.

EXHIBITS

These range from and comprise all the types of articles discussed or advertised in this copy of THE ASHAE GUIDE.

1. *The* COMBUSTION *Group:*
Furnaces, burners (coal, oil and gas), grates, stokers, boilers, radiators (various types), refractories and auxiliaries.

2. *The* OIL BURNER *Group:*

3. *The* HYDRAULIC *Group:*
Water feeders, water heaters, pumps, traps, valves, piping, fittings, expansion joints, pipe hangers, etc.

4. *The* STEAM HEATING *Group:*
Vapor heating, steam specialties.

5. *The* HOT WATER HEATING *Group:*

6. *The* AIR *Group:*
Warm Air furnaces and stoves, registers and grilles, cooling towers, air filters, motors, fans, blowers, conditioning equipment, ventilators (room and industrial types), unit heaters, etc.

7. *The* AIR CONDITIONING *Group:*
Equipment which circulates and filters the air, in summer dehumidifies and cools; in winter heats and humidifies, and does all these in proper season for complete, all year-round air conditioning.

8. *The* CONTROL *Group:*
Instruments of precision for indicating, controlling or recording temperature, pressure, volume, time, flow, draft or any other function to be measured.

9. *The* REFRIGERATING *Group:*
Compressors, condensers, cooling apparatus, contingent apparatus and refrigerants.

10. *The* CENTRAL HEATING *Group:*
Apparatus and materials especially designed or adapted to the uses of central heating and central heating station supplies.

11. *The* INSULATING *Group:*
Structural insulators (refractory and cellulose materials), asbestos, magnesia clays and combinations thereof, pipe and conduit covering, etc., weather-stripping, adhesives etc.

12. *The* MISCELLANEOUS *Group:*
Electric Heaters, boiler and pipe repair alloys, liquids and compounds, tools of all kinds, and equipment not specifically included in the above groups, but related thereto.

13. *The* MACHINERY AND GENERAL EQUIPMENT *Group.*

14. BOOKS AND PUBLICATIONS

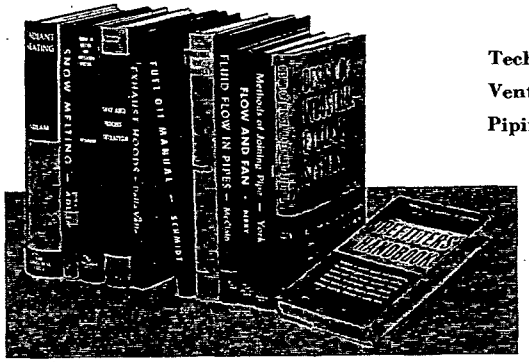
VISITOR ATTENDANCE

Attendance is by invitation and registration only, thereby presenting a selected audience. Included are contractors, dealers, jobbers, supply houses, home owners, industrial users, professional and service organizations, public utilities, real estate management concerns, etc. A detailed analysis of registered attendance is available on request.

Industrial Expositions in America lead the expositions of the world in style, business effectiveness, industrial influence and educational value. This Exposition stands among the leaders in Industrial Expositions in America. It is an educational institution which brings together the research developments and improvements in equipment and materials for use in heating, ventilating and air conditioning all types of buildings.

THE INDUSTRIAL PRESS

93 Worth Street, New York, 13, N. Y.



Technical Books on Heating, Ventilating, Air Conditioning, Piping and Plumbing

DESIGN OF AIR CONDITIONING SYSTEMS by F. W. Hutchinson
115 full-page charts solve problems of air conditioning system design involving cooling load, duct design, psychrometrics, solar energy, comfort conditioning, panel cooling, etc. For each chart the engineering and mathematical background plus design example and solution are given. 336 Pages, 118 Illus., $7.00.

DESIGN OF HEATING AND VENTILATING SYSTEMS
by F. W. Hutchinson
96 full-page charts solve problems of heating and ventilating involving load determination, duct design, panel heating, solar heating and combustion analysis. Companion book to Design of Air Conditioning Systems. Same methods to explain and solve problems are used. 320 Pages, 96 Charts, $7.00. By mail order, Canada and overseas, $8.50. SET OF BOTH, $12.50.

DESIGN OF INDUSTRIAL EXHAUST SYSTEMS by J. L. Alden
How to design, build or buy an exhaust system that will meet the requirements of law and industry. Covers flow of fluids—hood forms—air flow through hoods—pipe resistance—piping design—dust separators—low pressure conveyers—centrifugal exhaust fans—structural details—field measurements and their interpretation. 252 pages, 122 Illus., $3.50.

ELECTRICAL TESTING AND TROUBLESHOOTING by Philip Green
Simple, straightforward, time-tested procedures show how to locate and correct faults in circuits of all kinds including controllers, motors, transformers and transmission lines. Most tests described are performed with the equipment isolated from the power line. Written by an expert in electrical troubleshooting. 160 Pages, 34 Tables, $5.00.

EXHAUST HOODS by J. M. DallaValle
Practical hood design; latest principles of air handling, contaminant dilution and transport velocities; design of hoods for control of dust, mist, fumes, vapors, gases. Simple formulas and diagrams show exact procedures. 130 Pages, 127 Illus., $4.00.

FLOW AND FAN by C. H. Berry
Covers the flow of gas through ducts, and fan performance and control; moving air through ducts, fan selection and control, duct arrangement, system characteristics, flow analysis. Basic data and methods used to calculate system resistance. Practical information for selecting a fan for any duty. 232 Pages, 84 Illus., $4.00.

FLUID FLOW IN PIPES
by C. H. McClain
How to solve problems involving the flow of liquids and gases through pipes. How to handle viscosity, friction, heat, and other factors expressed in various dimensional systems. Worked-out problems show applications of principles. 124 Pages, 18 Illus., $4.00.

FUEL OIL MANUAL—2nd Edition by P. F. Schmidt
Covers characteristics and uses of every type and grade of fuel oil. Explains the meaning of each oil property, and shows how this information is applied in selecting, handling and burning fuel oil. A special chapter on fuel oil additives and another on troubles and remedies. 176 Pages, 34 Tables, $4.50.

HIGH TEMPERATURE WATER SYSTEMS by O. S. Lieberg
A thorough explanation of the principles, design, selection of equipment, operation and economics of high temperature water systems. Contains original design data used in actual systems designed by author. Features design of typical system based on principles set forth in this brand new book. 224 Pages, 109 Illus., Tables, $6.50.

HANDBOOK OF AIR CONDITIONING, HEATING AND VENTILATING
Edited by Clifford Strock
Completely new book containing thousands of facts, figures, data and principles on: Air Conditioning, Air Handling and Ventilation, Fuels and Combustion, Heating and Heat Transfer, Piping and Plumbing, Motors and Motor Starters, Mathematical Data and Drafting Room Symbols. Contains section on Terminology and complete cross-index. 1094 Pages, 7¾ in. x 10¾ in., 598 Charts, Maps, Illus, 518 Tables, $15.00.

INDUSTRIAL HEAT TRANSFER by F. W. Hutchinson
Provides 123 time-saving working graphs for the direct solution of most commonly encountered problems. Graphs are equal in accuracy to the equations from which they are derived and help eliminate errors in calculation. 336 Pages, 136 Charts and Illus., $7.00.

METHODS OF JOINING PIPE
by J. E. York
The most detailed information available on standard and special joints for all types of metallic, glass, tile, plastic and concrete pipe. Data on joints designed to take up movement due to expansion and contraction. 236 Pages, 249 Illus., $4.00.

PIPEFITTERS HANDBOOK—2nd Edition by F. R. Lindsey
Original tables enable pipefitters to solve problems of pipe bending, mitering, layout, threading, etc., either in the shop or in the field. Handy size, durable binding. 282 Pages, $6.00.

PLANT AND PROCESS VENTILATION by W. C. L. Hemeon
Design factors and data that can be applied to any situation. Principles are clear, logical. Shows how to estimate exhaust characteristics of any cold, multi-directional process; and obtain exhaust requirements for such a situation. 448 Pages, 172 Illus., $9.00.

RADIANT HEATING—2nd Edition by T. Napier Adlam
Latest developments, basic principles, facts, practical working data on applications of radiant energy for heating and cooling. Facts, figures, design data can be applied directly in designing and installing radiant heating systems without calculations. 504 Pages, 337 Illus., $6.00.

Prices slightly higher in Canada and overseas; write for catalog.

KEENEY PUBLISHING COMPANY

6 North Michigan Avenue, Chicago 2, Ill.





Heating, Piping & Air Conditioning is the one monthly publication that adequately serves the field of industry and large buildings. It is devoted to the design, installation, operation, and maintenance of heating, piping and air conditioning systems in plants, commercial, institutional and public buildings.

Each January issue includes a complete directory of commercial and industrial heating, piping, and air conditioning equipment, which lists all products, their trade names, and the manufacturers' addresses. It is the one established buying and specifying guide of the industry.

Heating, Piping & Air Conditioning is read by consulting engineers and architects . . . contractors . . . and engineers in charge of heating, piping and air conditioning in industrial plants, and other large buildings, federal, state, and city governments, school boards, and public utilities. Most of ASHAE members are subscribers.

Such coverage means, for the advertiser, consideration at all points in the selling of a heating, piping, or air conditioning product . . . consideration in its selection during the preparation of plans and specifications; in its actual purchase for installation; in its year-'round buying for operating and maintenance requirements. Without waste, the manufacturer of air conditioning products and equipment can reach through **Heating, Piping & Air Conditioning** those from whom he is seeking the necessary engineering acceptance.

SUBSCRIPTION PRICES—U. S. A. $3.00 a year.

AMERICAN ARTISAN is truly THE magazine of central residential air conditioning, warm air heating, and sheet metal contracting. Its readers are warm air heating and sheet metal contractors, dealers, jobbers, and manufacturers.

Special features of each issue have been devoted to air conditioning since 1932, when it first became apparent that air conditioning for homes was to be along the lines of the central forced warm air heating system. As a result of the ready adaptability of this type of heating system to all air conditioning factors, hundreds of thousands of homes today have winter air conditioning—supplied through forced warm air heating with air cleaning and humidification. Cooling apparatus can be attached to these systems readily whenever year-'round air conditioning is desired.

Each January issue includes the only complete directory of central residential air conditioning, warm air heating, and sheet metal products and equipment, which lists all products, their trade names, and the manufacturers' addresses.

The key man in the central residential air conditioning picture is the warm air heating and sheet metal contractor—the one man experienced in "treating air" at a central place and getting it properly distributed. And **American Artisan** is the key publication—because it reaches these key men with information that has made it the recognized authority on correct practice in residential air conditioning, warm air heating, and sheet metal contracting.

SUBSCRIPTION PRICES—U. S. A. $3.00 a year

Canada, $5.00 a year. Spain, Central and South America—$6.00 a year. Elsewhere $7.00 a year.
(Prices subject to change without notice due to impending increases in postal rates.)

MEMBER OF THE ASSOCIATED BUSINESS PUBLICATIONS AND THE AUDIT BUREAU OF CIRCULATIONS

Scott-Choate Publications

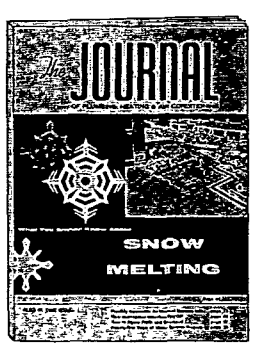
92 Martling Avenue, Tarrytown, New York



Subscription rates—$4.00 per year U. S., Canada and Pan America. Foreign, $6.00. Advertising rates on request.

HEATING & AIR CONDITIONING CONTRACTOR (formerly *Sheet Metal Worker*) is the leading magazine in the warm air heating, packaged air conditioning, ventilation, and sheet metal contracting industry. Its editorial scope includes not only residential heating and cooling work, but non-residential air conditioning and air distribution for comfort and process control.

HEATING & AIR CONDITIONING CONTRACTOR editorial content is directed primarily at the management and planning levels of the heating, air conditioning, and sheet metal contracting business. It covers not only the technical aspects of this field, but also marketing, merchandising, business management and legal problems. It is also the only publication in the field to devote regular monthly space to labor relations.



Subscription rates—$4.00 per year U. S., Canada and Pan America. Foreign, $6.00. Advertising rates on request.

THE JOURNAL OF PLUMBING, HEATING & AIR CONDITIONING is the recognized editorial leader in the plumbing and heating contracting industry. Through the use of intensive editorial research it has developed a unique formula for building effective readership. A key factor is consistent editorial balance among the various aspects of reader interest: technical, marketing, business management, overall industry issues.

In technical editorial, The **JOURNAL** has consistently been a pioneer. Among the subjects on which it has published the most comprehensive information ever produced are the following: baseboard radiation; submersible deep well pumps; hydronic air conditioning for residences and small non-residential structures; water conditioning for domestic and industrial use; etc.

The **JOURNAL** is also the only publication in its field with a regular monthly section devoted to the labor relations problems encountered by large mechanical contractors in their day to day operations.

Snips Magazine

5707 W. Lake St. Chicago 44, Ill.

PUTS A REAL PUSH BEHIND SALES

A friendly, close-to-the-reader periodical published for those contractors who handle Warm Air Heating, Air Conditioning, Sheet Metal Work, and Ventilation.



The Snips Magazine is preferred as an advertising medium by many notable industrial advertisers year after year, and many prominent jobbers and wholesalers who know their market well. It's a unique periodical which helps you put a real push behind sales.

LIVE STORIES IN SNIPS

SNIPS is packed each month, from cover to cover, with hundreds of live news stories and exclusive pictures including practical applications of latest advancements of the Sheet Metal, Heating, Air Conditioning and Ventilation trade. No long contributed stories. It's all field gathered material, secured the hard way, rubbing shoulders with the readers. Such work gets for the periodical a reader interest seldom found in a trade publication. You'll find this feature the basis of the sensational inquiry pulling power and sales producing value of advertising space in SNIPS.

MOST ADVERTISERS SELECT SNIPS FOR PAST 6 YEARS

When your copy appears in SNIPS, it is in the periodical which has carried more advertising accounts than any other magazine in this field for the past six years. This is in exclusion of the hundreds of listing ads carried each month. This continued preference for our magazine, as an advertising medium, is something of which we are extremely proud. The accomplishment challenges us to do a continually better publicity job to maintain this continued preference for our periodical as an advertising medium.

SNIPS HOLDS THE LINE ON RATES

To "hold the line" on our present advertising rates we did everything possible to reduce our costs. As a result there has been no rate increase since April, 1957. We were extremely happy that we wouldn't have to increase the rates as most trade papers have had to do, due to the recent postage increase and increases in the cost of paper, printing, cuts, etc. In fact, as a result of the drive made to get around these increases we came up with lower costs than before.

EXTRA COVERAGE AT NO EXTRA COST

We were able to put some of these savings into extra coverage. The final result was that while we promised a thirteen thousand coverage we have been able to give advertisers an extra thousand copies per month and still keep the rates the same.

TOP INDUSTRY COVERAGE TO CONTINUE

Even without this above mentioned coverage increase, we continued during the year past, to provide for you, the top coverage of the magazines serving the sheet metal, warm air heating, cooling and allied industries. No effort has been made to increase coverage promiscuously. Instead, every reader added has been a known, responsible buyer of the kind of goods you sell.

ABOUT SNIPS' MAILING LIST

It's truly a select group that gets SNIPS. They are the outstanding firms and individuals in the trade whom the better supply houses are selling or trying to sell. They are the contracting and installing concerns whom the principal jobbers and distributors of the industry—people who really know their territories best—consider worthy of cultivation by mail, promotion and salesmen's calls.

MUCH TO MERIT USE OF SPACE IN THIS PERIODICAL

There is much to merit your selection of the SNIPS Magazine as one of your advertising mediums in the coming year. It will be our constant endeavor to continue to give you the best trade paper advertising value available anywhere at any price.

Special Issues

January Annual & Winter Market Number—March Anniversary & Spring Market Issue—September Fall Market Issue—December Holiday Greeting Issue—Further information gladly sent on request.

ENGINEERS OF HUMAN COMFORT

The Heating, Ventilating and Air-Conditioning Engineers through their work and research bring to our homes, schools, offices, factories, theaters, hospitals and other public buildings in both summer and winter, that climate best suited to our comfort and health. These men realize the basic importance of heating, ventilating and air conditioning as a primary element in the well-being of civilized mankind, living and working mostly indoors. They are truly *Engineers of Human Comfort.*

MEN WITH VISION

The professional status of this form of engineering today can be largely attributed to the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS (ASHAE). Founded in 1894 by a handful of men determined to elevate the dignity of their profession and provide the public with more than perfunctory service, the Society represents an organization of more than 12,087 members. It includes 74 chapters throughout the United States and Canada plus members in many foreign countries.

On August 2, 1894, Hugh J. Barron, the acknowledged founder of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS (ASHVE), L. H. Hart, William M. Mackay, and 15 other interested men met in Mr. Hart's office in the World Building, New York City, to discuss the need for developing and disseminating technical advancements to heating and ventilating engineers. These men stimulated sufficient interest to obtain 75 charter members who met on September 10, 1894, in the Broadway Central Hotel, New York City, and adopted a name, by-laws, and constitution for the new Society.

Consistent with the increasing amount of activity by members in designing air-conditioning systems, in 1954 the Society changed its name to AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS.

SOCIETY SCOPE AND OPERATION

The three major activities of the Society are: Membership service, Publication, and Research, the records of its accomplishments being permanently recorded in the annual TRANSACTIONS.

The Society membership now includes engineers, educators, scientists, physicians, architects, contractors, and leaders of industry. Membership consists of Honorary, Life, Presidential, Fellow, Member, Associate, Affiliate and Student Grades.

The management of the Society is entrusted to 4 elected Officers and a Council of 13 elected members. Continuity of policy is insured by electing 4 men annually for a 3-year term and retaining the retiring president on the Council for 1 year.

Two Society meetings are held each year—the Annual Meeting during January or February, and the Semi-Annual Meeting usually in June or July. Every 2 years the *International Heating and Air-Conditioning Exposition* is held in conjunction with the Annual Meeting under the auspices of the Society.

The MEMBERSHIP STATUS as of January 26, 1959 was as follows:

Honorary	4
Presidential Members	19
Life Members	255
Fellows	9
Members	4,989
Associate Members over 30 yrs	3,712
Associate Members under 30 yrs	699
Affiliates	2,226
Students	174
TOTAL	12,087

HEADQUARTERS AND LABORATORY

The Society headquarters is maintained at 62 Worth St., New York 13, N. Y., and the ASHAE Research Laboratory, devoted to the study of fundamental principles of heating, ventilating and air conditioning, is located at 7218 Euclid Ave., Cleveland 3, Ohio.

ASHAE DEDICATED TO RESEARCH

From the day of its inception, a primary purpose of ASHAE was and still is research. For the first 25 years such research activity had been confined to individuals or committees utilizing available equipment or facilities.

In January 1919 the Society approved the establishment of a permanent research laboratory located in the U. S. Bureau of Mines, Pittsburgh, Pa. The Laboratory was moved in 1944 to limited quarters in Cleveland, Ohio and in 1946 the Society acquired its present large, permanent quarters in Cleveland. The Society is justly proud of the fact that it is the only professional engineering society that maintains and operates its own research laboratory.

All research activities are planned and supervised by the Committee on Research of 15 elected members, assisted by various Technical Advisory Committees of the Society. Since 1919, approximately $2,500,000 has been expended on research, which has helped guide the industry to better practice and equipment. In addition to work at the Society's laboratory, a substantial part of the research program has been carried on through the medium of cooperative agreements with leading educational institutions of the United States and Canada. Together with the cooperating institutions the Society has, during the last year, conducted 22 research projects including work on Air Distribution, Fan and Air Noise, Hot Water Heating, Industrial Ventilation, Insulation, Odor Control Comfort, and Physiological Factors. The research activities are financed from Society funds, of which a portion comes from membership dues and from its publications, and these funds are amplified by contributions from friends in the industries engaged in the general field of heating, ventilating and air conditioning.

SOCIETY PUBLICATIONS

TRANSACTIONS—The first Society publication established in 1895 provides a permanent record of Society activities published annually, containing scientific papers, proceedings of meetings, discussions, reports of meetings. Distributed to members as part of membership service; and to others for $6.00.

JOURNAL—Contains advance publication of technical papers, the reports of Society and Chapter meetings, and other items of general interest. The JOURNAL was published from April 1915 to October 1919 as a quarterly; 9 times a year from 1920 to 1924, then monthly from January 1924 to April 1929, and as a section of *Heating, Piping & Air Conditioning* magazine from May 1929 to December 1958.

THE GUIDE—A distinctly new service was inaugurated by the Society in 1922 when it established the HEATING VENTILATING AIR CONDITIONING GUIDE. Now in 1959, as the 37th Edition makes its appearance, it is notable that THE GUIDE has served effectively not only the membership but the entire profession and the allied industries, and has received world-wide recognition as a reliable and authoritative compendium of useful heating, ventilating and air-conditioning data.

Throughout the 37 years of its service THE GUIDE has become a reference book of unchallenged position in its special field of engineering. The intention of its founders, to provide an instrument of service containing reference material on the design and specification of heating, ventilating and air-conditioning systems and containing essential and reliable information concerning modern equipment, has been carefully safeguarded by those responsible for the compilation of each Edition.

THE GUIDE exerts today one of the most positive influences tending to elevate, improve and extend the whole Art of Heating, Ventilating and Air Conditioning. It is universally recognized as the most useful and authoritative work in its field, being used by practicing engineers, educators and manufacturers in all parts of the world, and as a text book by a growing number of the world's principal engineering institutions.

REGIONS AND CHAPTERS OF THE SOCIETY

Under the Regional Plan adopted in 1956 the 72 Chapters are divided into 7 geographical regions, with regions headed by a regional director who is chairman of the Chapters Regional Committee for his respective area. The regional directors are MEMBERS of the Council, and also comprise the Regions Central Committee of which the Second Vice President is chairman.

Each Chapters Regional Committee meets once a year between February and May to discuss Chapter problems and select a member to represent the region on the Society Nominating Committee.

Individual Chapters average eight meetings a year between September and May, which are distinct and apart from Society Annual and Semi-Annual Meetings.

REGION 1—Connecticut, Delaware, Maine, Massachusetts, New Hampshire, New Jersey, New York, Pennsylvania, Rhode Island, Vermont.

REGION 2*—Illinois, Indiana, Michigan, Ohio, Wisconsin.

REGION 3—Colorado, Iowa, Kansas, Minnesota, Missouri, Montana, Nebraska, North Dakota, South Dakota, Wyoming.

REGION 4—Arizona, California, Idaho, Nevada, Oregon, Utah, Washington.

REGION 5—Alabama, District of Columbia, Florida, Georgia, Kentucky, Louisiana, Maryland, Mississippi, North Carolina, South Carolina, Tennessee, Virginia, West Virginia.

REGION 6—Arkansas, New Mexico, Oklahoma, Texas, and Shreveport, La.

REGION 7—All Provinces of Canada.

*Includes Iowa-Illinois Chapter.

LOCAL ASHAE CHAPTERS

Arizona—Phoenix, Aris.
Arkansas—Little Rock, Ark.
Atlanta—Atlanta, Ga.
Austin—Austin, Tex.
Baltimore—Baltimore, Md.
Baton Rouge—Baton Rouge, La.
Bluegrass—Louisville, Ky.
British Columbia—Vancouver, B. C., Canada
Central New York—Syracuse, N. Y.
Central Ohio—Columbus, Ohio
Cincinnati—Cincinnati, Ohio
Connecticut—New Haven, Conn.
Delta—New Orleans, La.
El Paso—El Paso, Tex.
Empire State Capital—Albany, N. Y.
Fort Worth—Fort Worth, Tex.
Golden Gate—San Francisco, Calif.
Illinois—Chicago, Ill.
Illinois-Iowa—Moline, Ill.
Indiana—Indianapolis, Ind.
Inland Empire—Spokane, Wash.
Iowa—Des Moines, Ia.
Jacksonville—Jacksonville, Fla.
Johnstown—Johnstown, Pa.
Kansas—Wichita, Kan.
Kansas City—Kansas City, Mo.
Long Island—Garden City, N. Y.
Manitoba—Winnipeg, Man., Canada
Massachusetts—Boston, Mass.
Memphis—Memphis, Tenn.
Miami Valley—Dayton, Ohio
Michigan—Detroit, Mich.
Minnesota—Minneapolis, Minn.
Mississippi—Jackson, Miss.
Montreal—Montreal, Que., Canada
Nebraska—Omaha, Neb.
New Mexico—Albuquerque, N. M.
New York—New York, N. Y.
North Jersey—Newark, N. J.
North Texas—Dallas, Tex.
Northeastern Oklahoma—Tulsa, Okla.
Northern Alberta—Edmonton, Alta., Canada
Northern Ohio—Cleveland, Ohio
Northern Piedmont—Greensboro, N. C.
Oklahoma—Oklahoma City, Okla.
Ontario—Toronto, Ont., Canada
Oregon—Portland, Ore.
Ottawa Valley—Ottawa, Ont., Canada
Philadelphia—Philadelphia, Pa.
Pittsburgh—Pittsburgh, Pa.
Puget Sound—Seattle, Wash.
Chapitre de la Ville de Québec—Québec, P. Q., Canada
Rhode Island—Providence, R. I.
Rocky Mountain—Denver, Colo.
Sacramento Valley—Sacramento, Calif.
San Diego—San Diego, Calif.
Savannah—Savannah, Ga.
St. Louis—St. Louis, Mo.
Shreveport—Shreveport, La.
South Carolina—Columbia, S. C.
South Texas—Houston, Tex.
Southern Alberta—Calgary, Alta., Canada
Southern California—Los Angeles, Calif.
Southern Piedmont—Charlotte, N. C.
Southwest Texas—San Antonio, Tex.
Toledo—Toledo, Ohio
Utah—Salt Lake City, Utah
Virginia—Norfolk, Va.
Washington, D. C.—Washington, D. C.
West Texas—Lubbock, Tex.
Western Massachusetts—Springfield, Mass.
Western Michigan—Grand Rapids, Mich.
Western New York—Buffalo, N. Y.
Wisconsin—Milwaukee, Wis.

F. PAUL ANDERSON MEDAL

In 1930 an endowment fund for the award of The F. Paul Anderson Medal was created by the late Thornton Lewis and Rules of Award were prepared by a special committee appointed by the Council. This award is in recognition of outstanding work in the field of heating, ventilating, and air conditioning. Since 1932 there were 16 awards:

1932 Willis H. Carrier*
1936 Dr. Arthur Cutts Willard
1939 Prof. Frank B. Rowley
1942 Dr. Frederick E. Giesecke*
1944 Comm. Ferry C. Houghten, U.S.N.R.*
1946 Capt. Alfred E. Stacey, Jr., U.S.N.R.
1947 James H. Walker*
1950 Dr. Charles-Edward A. Winslow*
1951 Samuel R. Lewis
1952 Homer Addams*
1953 Everett N. McDonnell
1954 Walter L. Fleisher
1955 Charles S. Leopold
1956 Robert W. Keeton, M.D.*
1957 Prof. M. K. Fahnestock
1958 Prof. George L. Tuve
1959 Prof. Constantin P. Yaglou

*Deceased

PRESIDENTS OF THE SOCIETY

1959—A. J. HESS
1958—E. R. QUEER
1957—P. B. GORDON
1956—JOHN W. JAMES
1955—JOHN E. HAINES
1954—LOUIS N. HUNTER
1953—REG. F. TAYLOR
1952—ERNEST SZEKELY
1951—LAUREN E. SEELEY
1950—LESTER T. AVERY
1949—A. E. STACEY, JR.
1948—G. L. TUVE
1947—BALDWIN M. WOODS
1946—ALFRED J. OFFNER
1945—C.-E. A. WINSLOW
1944—S. H. DOWNS
1943—M. F. BLANKIN
1942—E. O. EASTWOOD
1941—W. L. FLEISHER
1940—F. E. GIESECKE
1939—J. F. MCINTIRE
1938—E. HOLT GURNEY
1937—D. S. BOYDEN
1936—G. L. LARSON
1935—JOHN HOWATT
1934—C. V. HAYNES
1933—W. T. JONES
1932—F. B. ROWLEY
1931—W. H. CARRIER
1930—L. A. HARDING
1929—THORNTON LEWIS
1928—A. C. WILLARD
1927—F. PAUL ANDERSON
1926—W. H. DRISCOLL
1925—S. E. DIBBLE
1924—HOMER ADDAMS
1923—H. P. GANT
1922—JAY R. MCCOLL
1921—CHAMPLAIN L. RILEY
1920—E. VERNON HILL
1919—WALTER S. TIMMIS
1918—F. R. STILL
1917—J. IRVINE LYLE
1916—HARRY M. HART
1915—DWIGHT D. KIMBALL
1914—SAMUEL R. LEWIS
1913—JOHN F. HALE
1912—JOHN R. ALLEN
1911—REGINALD PELHAM BOLTON
1910—JAMES D. HOFFMAN
1909—WILLIAM G. SNOW
1908—JAMES MACKAY
1907—C. B. J. SNYDER
1906—JOHN GORMLY
1905—WILLIAM KENT
1904—ANDREW HARVEY
1903—H. D. CRANE
1902—A. E. KENRICK
1901—J. H. KINEALY
1900—D. M. QUAY
1899—HENRY ADAMS
1898—WILTSIE F. WOLFE
1897—WM. M. MACKAY
1896—R. C. CARPENTER
1895—STEWART A. JELLETT
1894—EDWARD P. BATES

MEETINGS OF ASHAE

Year	Annual Meeting	Date	Place	Semi-Annual Meeting	Place
1895	1st	Jan. 21-23	New York, N. Y.	None	
1896	2nd	Jan. 21-23	New York, N. Y.	None	
1897	3rd	Jan. 26-28	New York, N. Y.	June 18	New York, N. Y.
1898	4th	Jan. 25-27	New York, N. Y.	July 15	Atlantic City, N. J.
1899	5th	Jan. 24-26	New York, N. Y.	None	
1900	6th	Jan. 23-25	New York, N. Y.	None	
1901	7th	Jan. 22-24	New York, N. Y.	July 12-13	Chicago, Ill.
1902	8th	Jan. 21-23	New York, N. Y.	June 16	Atlantic City, N. J.
1903	9th	Jan. 20-22	New York, N. Y.	July 17-18	Niagara Falls, N. Y.
1904	10th	Jan. 19-21	New York, N. Y.	July 15-16	Detroit, Mich.

Year	Annual Meeting	Date	Place	Semi-Annual Meeting	Place
1905	11th	Jan. 17–19	New York, N. Y.	July 7–8	Chicago, Ill.
1906	12th	Jan. 16–18	New York, N. Y.	July 19–20	Chicago, Ill.
1907	13th	Jan. 22–24	New York, N. Y.	July 18–19	Milwaukee, Wis.
1908	14th	Jan. 21–23	New York, N. Y.	July 24–25	Niagara Falls, N. Y.
1909	15th	Jan. 19–21	New York, N. Y.	July 15–16	Indianapolis, Ind.
1910	16th	Jan. 18–20	New York, N. Y.	June 30–July 1	St. Louis, Mo.
1911	17th	Jan. 24–26	New York, N. Y.	July 6–8	Chicago, Ill.
1912	18th	Jan. 23–25	New York, N. Y.	July 11–12	Detroit, Mich.
1913	19th	Jan. 21–23	New York, N. Y.	July 17–19	Buffalo, N. Y.
1914	20th	Jan. 20–23	New York, N. Y.	July 9–11	Cleveland, Ohio
1915	21st	Jan. 20–22	New York, N. Y.	Sept. 16–17	Atlantic City, N. J.
1916	22nd	Jan. 18–20	New York, N. Y.	July 19–21	Detroit, Mich.
1917	23rd	Jan. 16–18	New York, N. Y.	July 18–20	Chicago, Ill.
1918	24th	Jan. 22–24	New York, N. Y.	June 26–28	Buffalo, N. Y.
1919	25th	Jan. 28–30	New York, N. Y.	June 10–12	Pittsburgh, Pa.
1920	26th	Jan. 27–29	New York, N. Y.	May 26–28	St. Louis, Mo.
1921	27th	Jan. 26–28	Philadelphia, Pa.	June 14–17	Cleveland, Ohio
1922	28th	Jan. 24–26	New York, N. Y.	June 6–7 June 8–10	Buffalo, N. Y. Detroit, Mich.
1923	29th	Jan. 23–26	New York, N. Y. and Washington, D. C.	May 21–23	Chicago, Ill.
1924	30th	Jan. 22–25	New York, N. Y.	June 19–22	Kansas City, Mo.
1925	31st	Jan. 27–30	New York, N. Y. and Boston	June 15–17	Atlantic City, N. J.
1926	32nd	Jan. 26–29	Buffalo, N. Y.	May 26–28	Lexington, Ky.
1927	33rd	Jan. 26–28	St. Louis, Mo.	June 28–30	White Sulphur Springs, W. Va.
1928	34th	Jan. 23–27	New York, N. Y.	June 26–29	West Baden, Ind.
1929	35th	Jan. 28–31	Chicago, Ill.	June 26–28	Bigwin Inn, Ont.
1930	36th	Jan. 27–31	Philadelphia, Pa.	June 24–27	Minneapolis, Minn.
1931	37th	Jan. 26–29	Pittsburgh, Pa.	June 22–25	Swampscott, Mass.
1932	38th	Jan. 25–29	Cleveland, Ohio	June 27–29	Milwaukee, Wis.
1933	39th	Jan. 23–25	Cincinnati, Ohio	June 22–24	Detroit, Mich.
1934	40th	Feb. 5–9	New York, N. Y.	June 20–22	Buck Hill Falls, Pa.
1935	41st	Jan. 28–30	Buffalo, N. Y.	June 16–19	Toronto, Ont.
1936	42nd	Jan. 27–30	Chicago, Ill.	June 22–24	Buck Hill Falls, Pa.
1937	43rd	Jan. 25–27	St. Louis, Mo.	June 24–26	Swampscott, Mass.
1938	44th	Jan. 24–28	New York, N. Y.	June 20–23	Hot Springs, Va.
1939	45th	Jan. 23–26	Pittsburgh, Pa.	July 4–6 *Oct. 30–31	Mackinac Isl., Mich. Atlanta, Ga.
1940	46th	Jan. 23–26	Cleveland, Ohio	June 17–19 *Oct. 14–15	Washington, D. C. Houston, Tex.
1941	47th	Jan. 27–29	Kansas City, Mo.	June 17–19	San Francisco, Calif.
1942	48th	Jan. 26–28	Philadelphia, Pa.	June 15–17	St. Paul, Minn.
1943	49th	Jan. 25–27	Cincinnati, Ohio	June 7–8	Pittsburgh, Pa.
1944	50th	Jan. 31–Feb. 1–2	New York, N. Y.	June 19–20	Grand Rapids, Mich.
1945	51st	Jan. 22–24	Boston, Mass.	None	
1946	52nd	Jan. 27–30	New York, N. Y.	June 10–13	Montreal, Que. (cruise)
1947	53rd	Jan. 27–30	Cleveland, Ohio	June 1–4	Coronado, Calif.
1948	54th	Feb. 2–5	New York, N. Y.	June 20–23	Bretton Woods, N. H.
1949	55th	Jan. 24–27	Chicago, Ill.	June 20–22	Minneapolis, Minn.
1950	56th	Jan. 23–26	Dallas, Tex.	June 19–21	Muskoka Lakes, Ont.
1951	57th	Jan. 22–25	Philadelphia, Pa.	July 2–4	Portland, Ore.
1952	58th	Jan. 28–30	St. Louis, Mo.	June 16–18	Spring Lake, N. J.
1953	59th	Jan. 26–29	Chicago, Ill.	June 29–July 1	Denver, Colo.
1954	60th	Jan. 25–27	Houston, Tex.	June 21–23	Swampscott, Mass.
1955	61st	Jan. 24–27	Philadelphia, Pa.	**April 14–16 June 27–29	Oklahoma City, Okla. San Francisco, Calif.
1956	62nd	Jan. 23–25	Cincinnati, Ohio	June 18–20	Washington, D. C.
1957	63rd	Feb. 25–28	Chicago, Ill.	June 24–26	Murray Bay, Que.
1958	64th	Jan. 27–29	Pittsburgh, Pa.	June 23–25	Minneapolis, Minn.
1959	65th	Jan. 26–29	Philadelphia, Pa.		

* Fall Meeting

** Regional Meeting

ASHVE AND ASHAE MILESTONES

1894–1959

1894 Organization meeting September 10, at Broadway Central Hotel, New York, with 75 Charter Members.

1895 First Annual Meeting, January 22–24 at 12 West 31st Street, New York. Incorporation under laws of New York State, TRANSACTIONS established.

1896 Society Emblem adopted at 2nd Annual Meeting—Dr. J. S. Billings, First Honorary Member.

1897 First Semi-Annual Meeting held at the Windsor Hotel, New York, June 18.

1906 First Chapter organized in Chicago.

1907 Membership reached 301.

1911 Membership totaled 405. New York Chapter formed. Society established Headquarters in Engineering Societies Building, 29 West 39th Street, New York.

1914 Charter amended.

1915 Journal first published in April as a quarterly; later issued monthly.

1916 Full time Secretary employed.

1918 Members in Armed Services—64.

1919 After several years of consideration and planning the Society established its Research Laboratory, in the United States Bureau of Mines, at Pittsburgh, Pa., having determined a preliminary *modus operandi* and provided an operating fund. The Society is justly proud that it is the only professional engineering organization which maintains and operates its own Research Laboratory.

1922 The 1st edition of the ASHVE GUIDE published in September.

1923 Comfort Zone Established by ASHVE Research.

1925 Dues raised to $25.00 and 40 per cent of dues of Members and Associates allocated to Research Fund.

1927 Membership of Council increased from 12 to 17.

1928 Appointment of Technical Secretary and completion of Code for Minimum Requirements for Heating and Ventilation of Buildings. Benjamin Franklin honored as Patron Saint of Society.

1929 Journal incorporated as a Special Section of *Heating, Piping and Air Conditioning*. Society Headquarters moved to 51 Madison Ave., New York, N. Y.

1930 Endowment created by Thornton Lewis for the F. Paul Anderson Medal. First International Heating and Ventilating Exposition held in Philadelphia.

1932 First Award of F. Paul Anderson Medal to Willis Haviland Carrier.

1933 New Constitution and By-Laws adopted—Membership dues reduced.

1938 Membership passed 3000.

1941 Society requested to do Research for U. S. Navy.

1944 50th Anniversary celebrated in New York.

1945 Members serving in Armed Forces—416.

1946 New Charter approved.

1950 New By-Laws adopted January 26, 1950.

1954 The name of the Society was changed to American Society of Heating and Air-Conditioning Engineers by vote of the members at a Special Meeting in New York, N. Y., November 22, effective December 8.

1956 The Honorable Herbert Hoover, Dr. Milton S. Eisenhower and Dr. Charles F. Kettering* were elected Honorary Members.

1957 Dr. Arthur Cutts Willard elected an Honorary Member.

1958 Membership total of ASHAE passed 12,000.
On December 1 the members of ASHAE and the ASRE, at simultaneous meetings held respectively in Chicago, Ill., and at New Orleans, La., voted to approve an agreement for consolidation.

*Deceased.

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS

Headquarters: 62 Worth St., New York 13, N.Y.
Tel. BArclay 7-6262

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1959

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First Vice President.....WALTER A. GRANT
Second Vice President.....JOHN EVERETTS, JR.
Treasurer.....J. H. FOX
Executive Secretary.....A. V. HUTCHINSON
Three years: WILLIAM J. COLLINS, JR., HAROLD A. LOCKHART, GEORGE W. MYERS, ROLLAND S. STOVER
Two years: F. H. FAUST, FRED JANSSEN, J. W. MAY, G. B. PRIESTER
One year: W. G. HOLE, P. J. MARSCHALL, E. R. QUEER, W. O. STEWART, P. N. VINTHER

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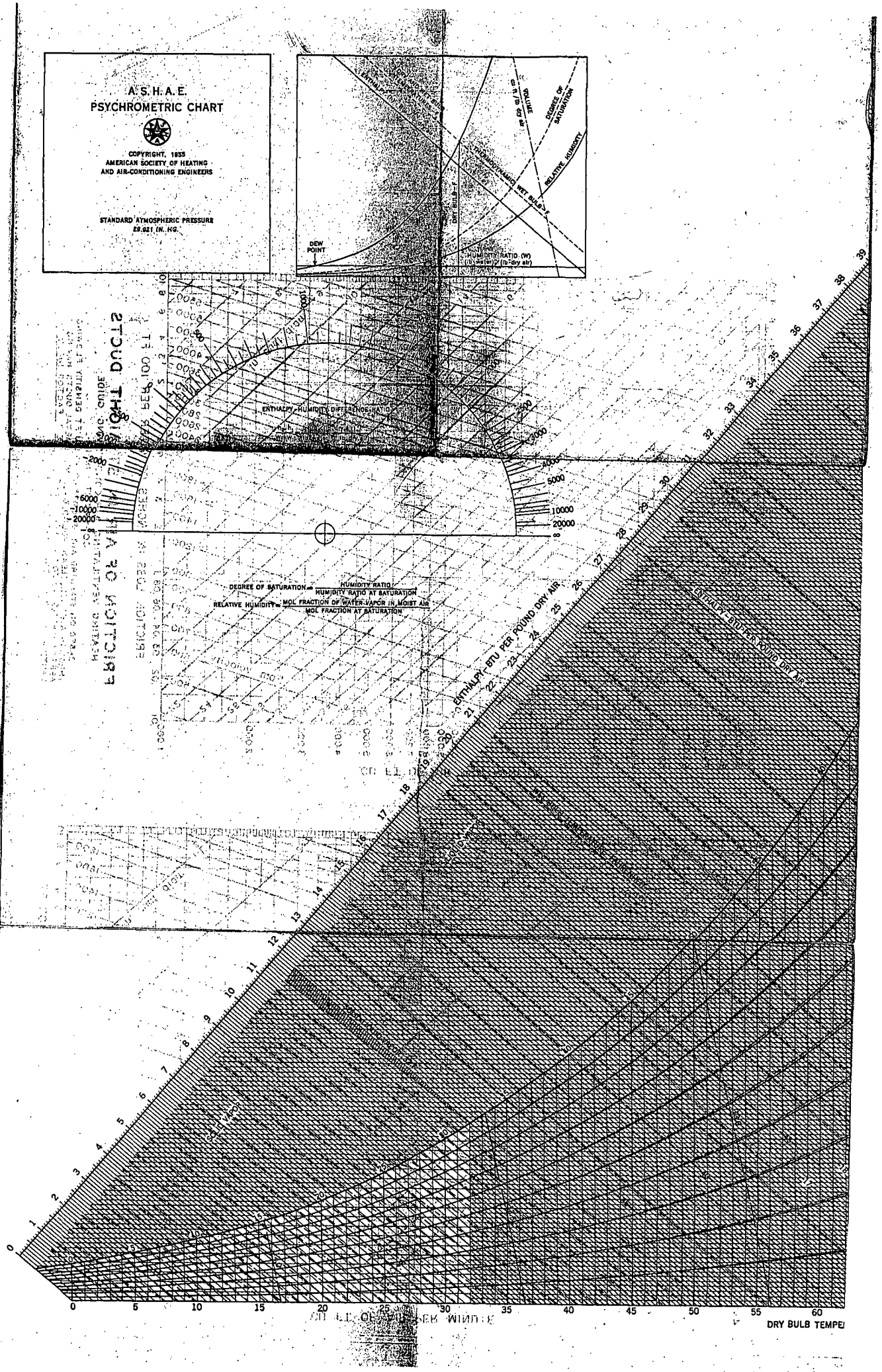
ASHAE Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio

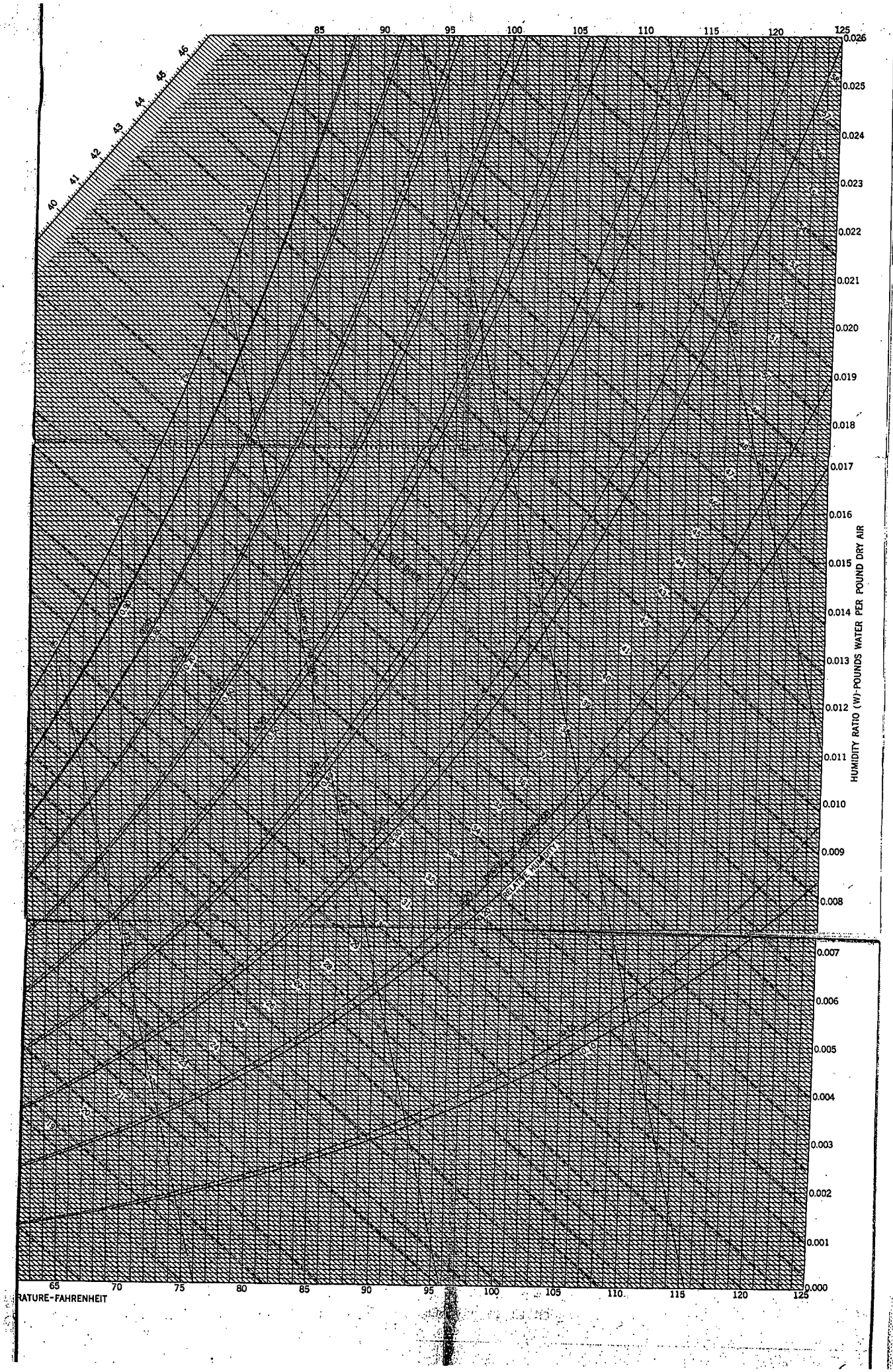
Director of Research.....B. H. JENNINGS
Assistant Director of Research.....C. M. HUMPHREYS
Assistant Director Industry Relations.....J. E. LOUCKS

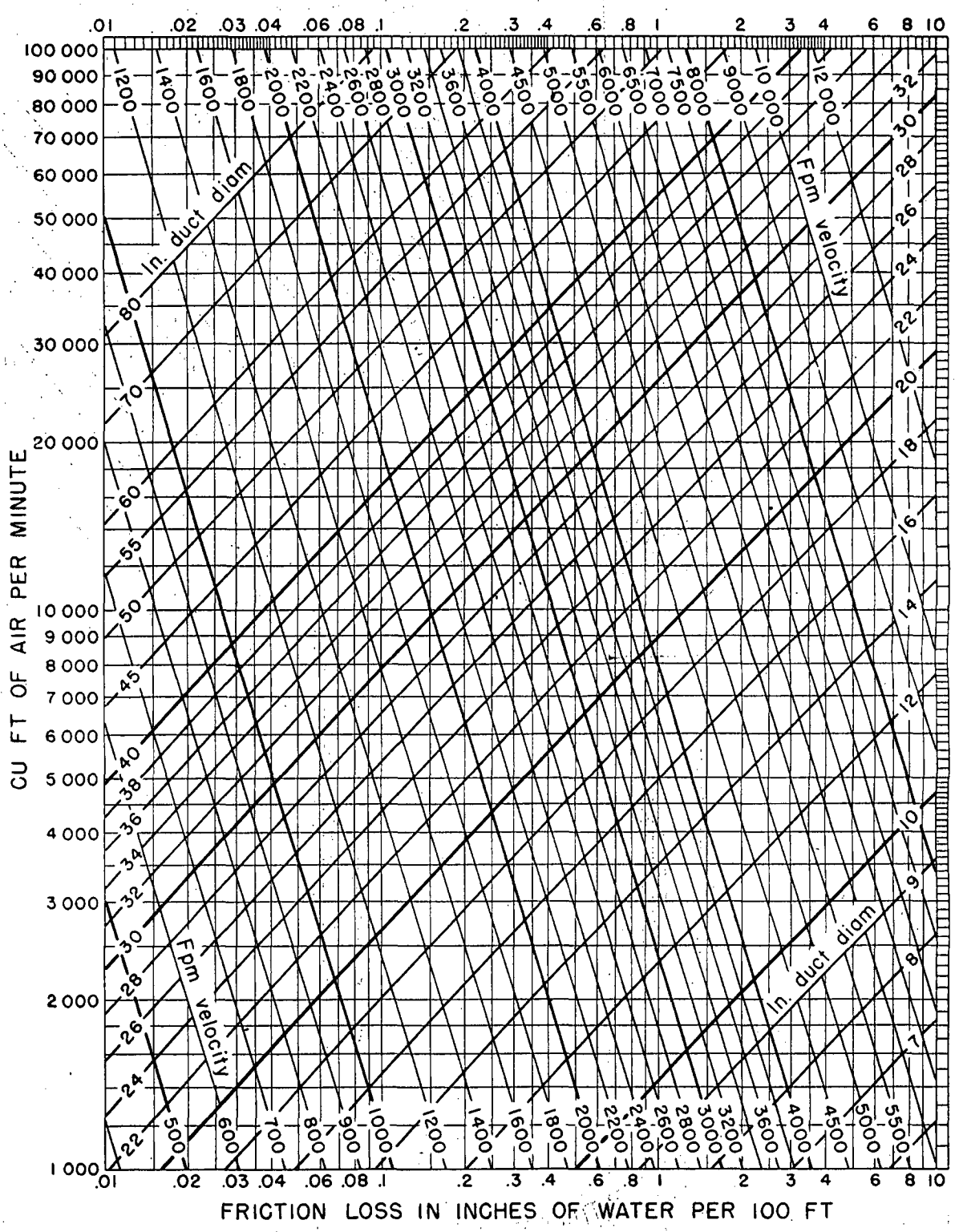
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STANDARD ATMOSPHERIC PRESSURE
29.921 IN. HG.
DEW POINT
VOLUME
DEGREE OF SATURATION
RELATIVE HUMIDITY
WET BULB
HUMIDITY RATIO (W)
ENTHALPY - HUMIDITY DIFFERENCE RATIO
DEGREE OF SATURATION = HUMIDITY RATIO / HUMIDITY RATIO AT SATURATION
RELATIVE HUMIDITY = MOL FRACTION OF WATER VAPOR IN MOIST AIR / MOL FRACTION AT SATURATION
ENTHALPY-BTU PER POUND DRY AIR
DRY BULB TEMPE


HUMIDITY RATIO (W)-POUNDS WATER PER POUND DRY AIR
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RATURE-FAHRENHEIT

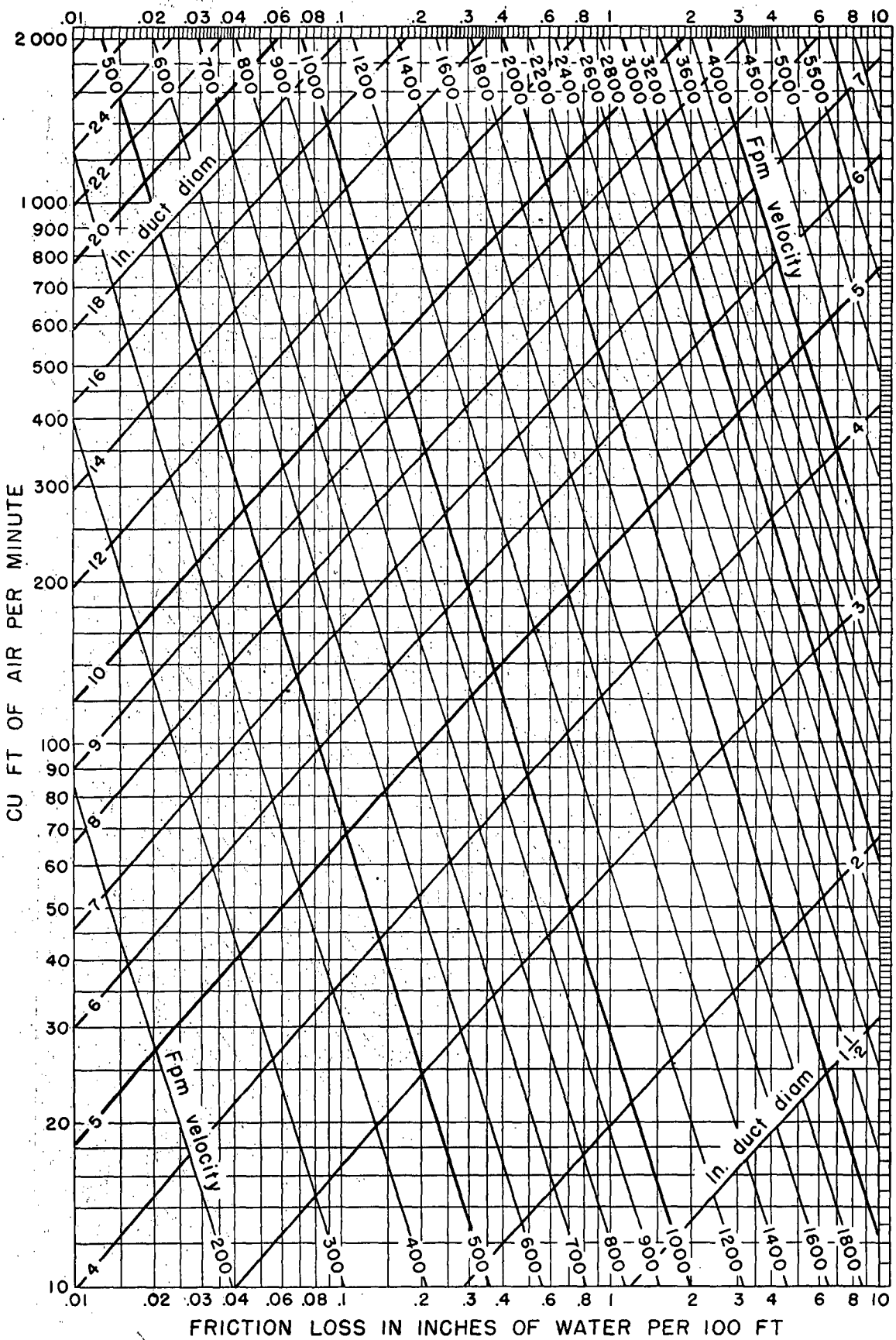


FRICTION OF AIR IN STRAIGHT DUCTS

HEATING VENTILATING AIR CONDITIONING GUIDE

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BASED ON STANDARD AIR OF 0.075 LB PER CU FT DENSITY FLOWING THROUGH AVERAGE, CLEAN, ROUND, GALVANIZED METAL DUCTS HAVING APPROXIMATELY 40 JOINTS PER 100 FT. NO SAFETY FACTOR INCLUDED.
CAUTION: DO NOT EXTRAPOLATE BELOW CHART.


CU FT OF AIR PER MINUTE
FRICTION LOSS IN INCHES OF WATER PER 100 FT
Fpm velocity
In. duct diam
Fpm velocity
In. duct diam